Modelling of Electrohydraulic Drive with a Valve Controlled by Synchronous Motor

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Abstract. The article describes modelling of electrohydraulic servo drive. In the drive a new type of proportional valve with a synchronous motor controlled by dedicated power electronics is used. The model of the electrohydraulic servo drive prepared in Matlab-Simulink is described. The study included the examination of the basic characteristics such as step response.

Keywords: electrohydraulic servo drive, proportional valve, synchronous motor.

1 Introduction

Electrohydraulic servo drives are commonly used in a large amount of practical applications. Wide industrial applicability of such drives is a result of their many advantages. They require small power input signals, to precisely control very big forces at output. Electrohydraulic servo drives are highly non-linear devices. Much of the current written publications related to the electrohydraulic servo drives focuses on improving the properties of existing drive, by implementation of modern forms of control, overseeing the work of these devices, as well as finding of new ways to provide movement of valve moving parts [3]. In proportional valves as the spool driving actuator most frequently proportional electromagnets are used. Following this path, it is advisable to search for set point actuators with better dynamic properties and positioning accuracy. In following article Permanent Magnet Synchronous Motors (PMSM) is proposed as control actuator for such task [3].

In the article the Authors describes the modelling of the electrohydraulic servo drive with proportional valve controlled by modern low power synchronous motor, which can be used in loop of the controller.

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2 Electrohydraulic Drive with Synchronous Motor – Simulation and Modeling

2.1 Proportional Valve with Synchronous Motor

Modern synchronous motors ensure high positioning accuracy with high dynamics. In the proposed proportional valve spool is actuated by a low-power Permanent Magnet Synchronous Motor (PMSM) [4].

In the literature many attempts to use of variable types of motors in hydraulic valves are described. Publication [2] described the use of a stepping motor in the valve, to obtain a very small velocity of electrohydraulic drive. Article [3] describes the use of servo motor for control of the proportional valve for high range of flow. In article [1, 6] Author presented the study of the use of a stepping motor in linear electrohydraulic drive.

2.2 Modeling of the Electrohydraulic Servo Drive

The motor (1) was connected to the spool *3* by flexible coupling bellow *2* (Fig. 1). Applying the electrical power to the motor causes rotation and simultaneously axial translation of the spool in the valve body *4*. Control edge openings x are proportional to the angular motor position and to the pitch of used thread *5*. Direction of rotation determines direction of spool translation and opening or closing of valve gaps x. The proportional valve *1* is connected to the hydraulic cylinder *2* as shown on Fig. 2. Servo drive is equipped with internal position sensor *3*, measuring the actual position of cylinder piston.

In the valve, Authors used PMSM motors type B&R 8LVA23. Its basic parameters are: the rated speed 3000 rev/min, rated current 2,9 A and stall torque 0,68 Nm. The motor was equipped with an absolute encoder type EnDat, providing a continuous information about the current position, even after a power failure, which assures a very high positioning accuracy and safe operation.

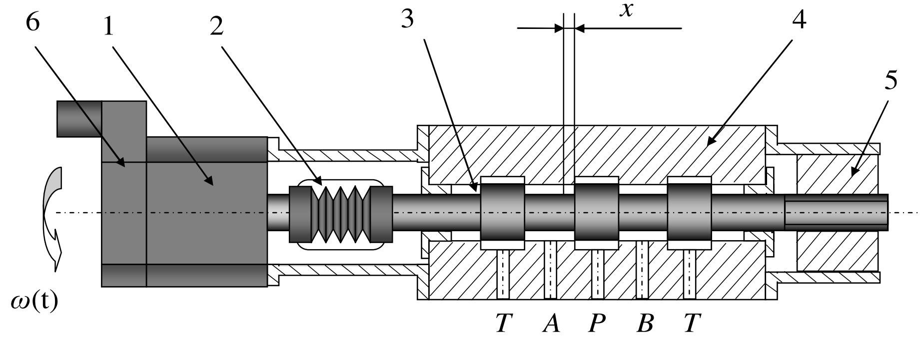

Fig. 1. Scheme of electrohydraulic proportional valve with PMSM motor

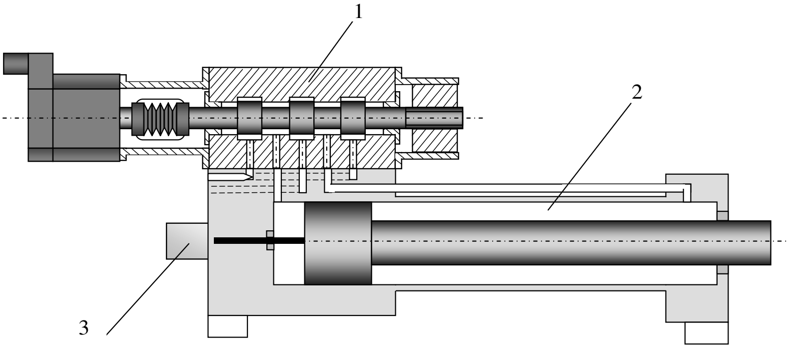

Fig. 2. Electrohydraulic servo drive with proportional valve with PMSM motor

Model of electrohydraulic servo drive (Fig. 3) was based on the graphical simplification of real system.

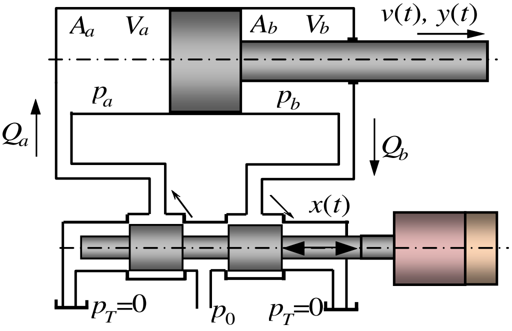

Fig. 3. Schema of the four edge electrohydraulic amplifier with PMSM motor and hydraulic actuator

The set of equations describing the electrohydraulic servo drive model can be specified as follows:

$$Q_a(t) = Q_{sa}(t) + Q_{ha}(t) + Q_v(t) \tag{1}$$

$$Q_b(t) = Q_{sb}(t) + Q_{hb}(t) + Q_v(t) - Q_{vb}(t) \tag{2}$$

$$Q_b(t) = K_{Qp}\, x(t) - K_l\, p_b(t) \tag{3}$$

$$Q_{ha}(t) = A\frac{dy(t)}{dt} \tag{4}$$

$$Q_{hb}(t) = aA\frac{dy(t)}{dt} \tag{5}$$

$$Q_{sa}(t) = \frac{V_a}{E_o}\frac{dp_a(t)}{dt} \tag{6}$$

$$Q_{sb}(t) = -\frac{V_b}{E_o}\frac{dp_b(t)}{dt} \tag{7}$$

$$Q_v(t) = K_v[p_a(t) - p_b(t)] \tag{8}$$

$$Q_{vb}(t) = K_{vb}p_b(t) \tag{9}$$

$$m\frac{d^2y(t)}{dt^2} + D\frac{dy(t)}{dt} = A[p_a(t) - ap_b(t)] \tag{10}$$

where: Q_a, Q_b – flow, Q_{ha}, Q_{hb} – absorption of the actuator chambers, Q_{sa}, Q_{sb} – flow of the covering losses due to compressibility, Q_{vb} – leakage flow on the piston rod, p_a, p_b – the pressure in the chambers of the actuator, A_a, A_b – active surfaces of the piston, V_a, V_b – the volume of liquid in the chambers of the actuator.

Unknown model parameters such as capacity of the hydraulic pipes and the flow rate has been identified with used of the Kalman filter.

The present set of equations (1–10) allows to determine transfer function, representing the change in velocity of the piston transforms Y(s) as a function of changes position of the slider amplifier transform X(s) (taking into account the zero loading force F_{obc}):

$$G(s) = \frac{sY(s)}{X(s)} = \frac{k_s\omega_s^2}{s^2 + 2\zeta_s\omega_s s + \omega_s^2} \tag{11}$$

where:

- gain:

$$k_s = \frac{K_{Qp}\left(\frac{1}{V_A} + \frac{a}{V_B}\right)}{A\left(\frac{1}{V_A} - \frac{a^2}{V_B}\right)}, \tag{12}$$

- pulsation:

$$\omega_s = \sqrt{\frac{c_s}{m}}, \tag{13}$$

- damping:

$$\zeta_s = \frac{D_w}{2m\sqrt{\frac{c_s}{m}}}, \tag{14}$$

- stiffness of the actuator:

$$c_s = E_0 A^2 \left(\frac{1}{V_A} - \frac{a^2}{V_B} \right). \tag{15}$$

3 Simulation

Simulation model was built based on the equations (1–10). In Fig. 4 the simulation model of the electrohydraulic servo drive with PMSM based valve is shown. Its structure is typical for single piston side cylinders, where non-symmetric designs of the cylinders requires separate block chains for each cylinder chamber [1, 4].

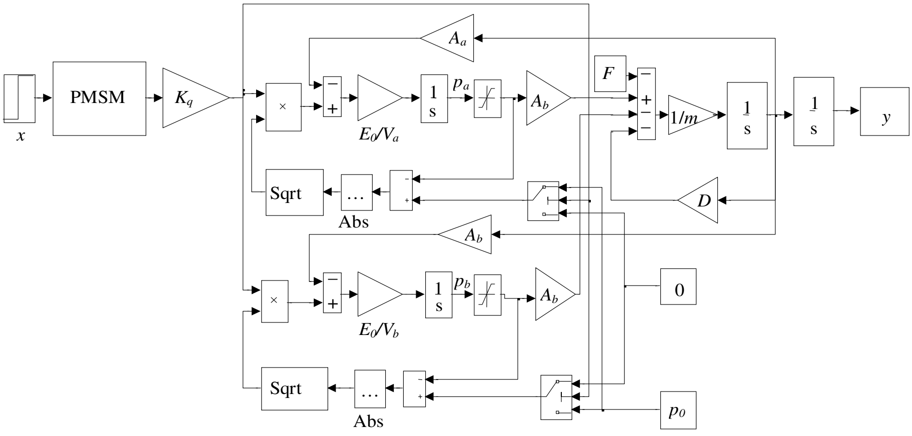

Fig. 4. Model of the electrohydraulic servo in Matlab Simulink software

The stroke of the hydraulic actuator was 200 mm. The diameters of the piston and the piston rod respectively, were A = 40 mm and Aa = 63 mm. The simulation includes linear stiffness and a linear coefficient of kinetic friction coulomb rate of $D = 29000$ [N·s/m]. The value of the reduced mass was m = 20.2 kg (mass of the piston and piston rod). The modulus of elasticity was $E_0 = 1.2 \cdot 10^9$ N/m^2. The values of the coefficients occurring in equations (6) (in the middle position of the piston) amounted to $E_0/V_a = 3.82 \cdot 10^{11}$ Pa/m^3 and $E_0/V_b = 5.96 \cdot 10^{11}$ Pa/m^3.

Model of the electrohydraulic servo drive was tested by used of the step response signals. Figures 5 shows the simulation results, with the displacement of the piston rod as the result of the valve opening. The simulation was performed for the supply pressure p_0 amounting to 5 MPa and 15 MPa.

Figure 6 shows the impact of the load on the drive system for a supply pressure p_0 = 15 MPa.

Figure 7 shows the displacement of the piston relative to the valve opening valve (with coincidence character).

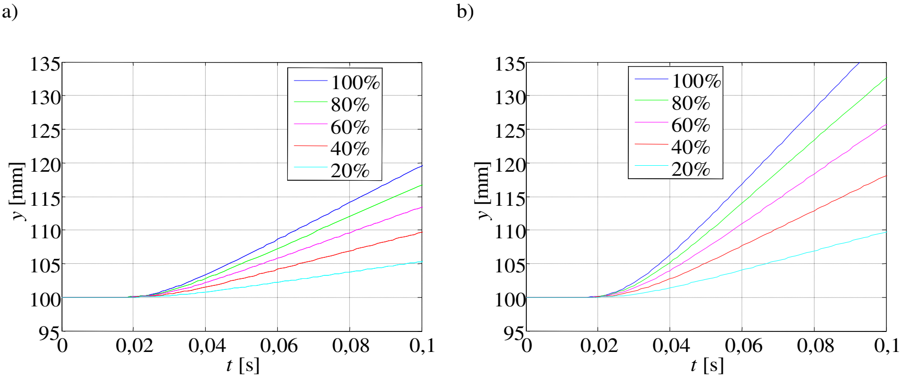

Fig. 5. Displacement of the piston rod for a supply pressure of 5 MPa (a) and 15 MPa (b)

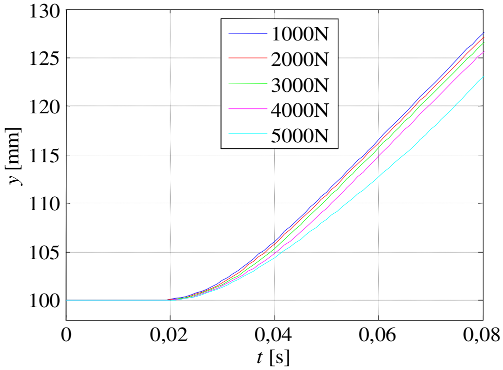

Fig. 6. Load impact on the electrohydraulic servo drive

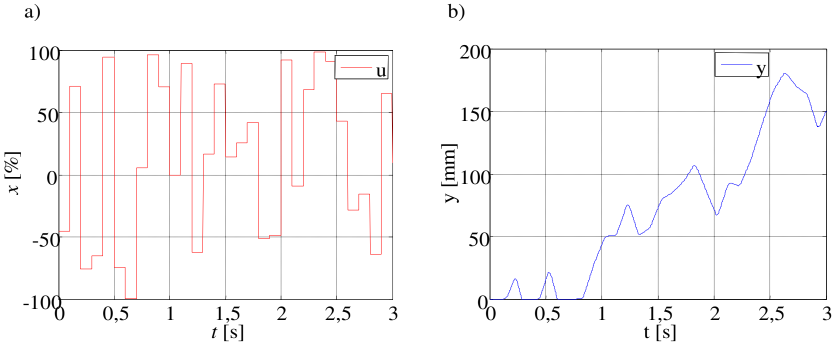

Fig. 7. Servo drive response (b) on given valve command signal (a)

4 Experimental Research

Modelling were allowed to the testing drive based on proposed here valve. The test stand (Fig. 8) consist of a hydraulic cylinder with stroke range 200 mm combined with a proportional valve controlled by synchronous motor. The cylinder was equipped with a magnetostrictive type of position sensor.

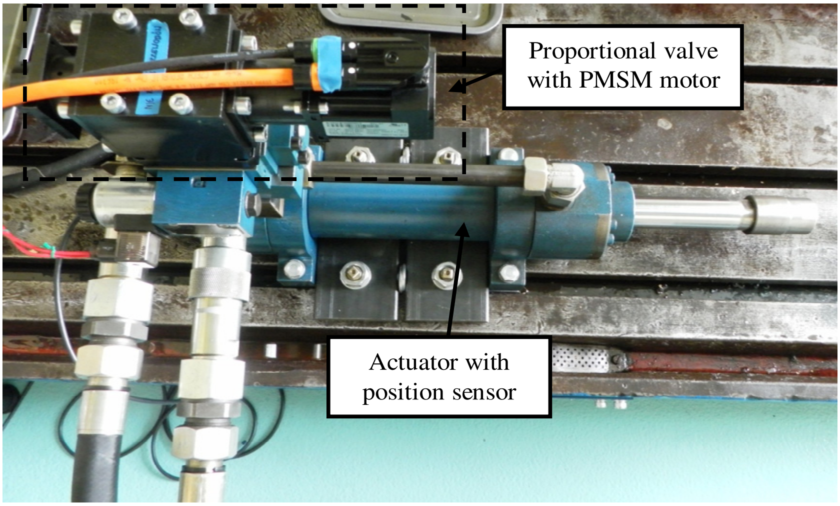

Fig. 8. Test stand

The drive was tested with step response signal. Based on the step response characteristic of the model and the real object can be state that the model accurately reflects the real object (Fig. 9). Position error between model and the real object, in extreme cases, was 2 mm.

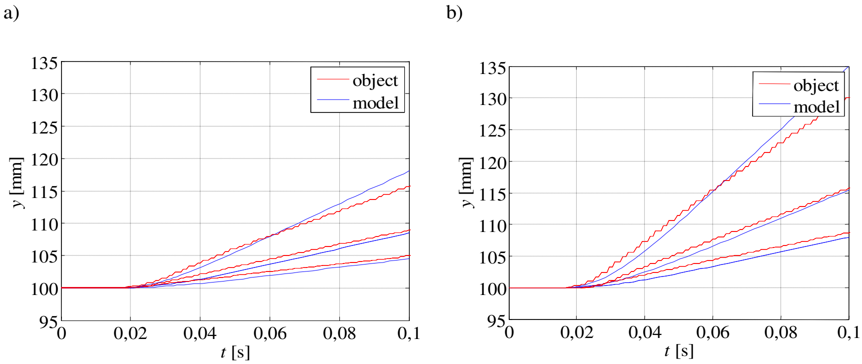

Fig. 9. Displacement of the piston rod for model and the object, for selected supply pressures: a) 5 MPa, b) 15 MPa

5 Conclusion

Article presents the basic equations describing the electrohydraulic drive. On the basis of this equations, simulation model was built. The electrohydraulic drive is equipped with the proportional valve controlled by PMSM motor. To compare the results of simulation with the real object, dedicated test stand was built. Based on these results can be state that the model accurately reflects the real object.

Performed drive model can be used as a reference model in adaptive control systems. The currently ongoing studies are aimed at implement it in the Model Following Control system.

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