MODELING OF ELECTROHYDRAULIC DRIVE SYSTEM WITH AN INTEGRATED HEAT EXCHANGER IN DYNAMIC MODE

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ABSTRACT: The paper inhere describes the operation of an automated electrohydraulic drive system with a rotary actuator and an integrated "tube in tube" heat exchanger. A nonlinear mathematical model has been developed which reflects the dynamics of the system considering the alteration in temperature of the working fluid. The operation of the described automatic system is discussed in order to keep constant the temperature of the working fluid as a part of the electrohydraulic drive system. The developed mathematical model can be used for studying the dynamic processes in various system modes.

KEY WORDS: electrohydraulic system, automatic system, dynamic process, heat exchanger.

INTRODUCTION

In the determination of a number of practical problems associated with the use of electro–hydraulic drive systems it is assumed that the working fluid is incompressible and does not change its volume when the temperature and pressure change. During the operation of the system the working fluid is a subjected of heating, cooling and varying of pressure. This is related to changes in fluid's characteristics, resulting in changes of the resource, reliability and the efficiency of the system.

The process of system operation is associated also with friction forces between the working fluid and the components of the hydraulic drive, which increase the temperature of the working fluid. The temperature rise leads to reduced viscosity of the working fluid, and therefore increases the losses. Also, the heated to high temperatures working fluid causes excessed wear of parts of the system due to worsened conditions of working surfaces' lubrication.

To avoid excessive working fluid heating, there are used heat exchangers with a water or air cooling [1, 2, 3, 4]. Maintaining of better operating conditions of the hydraulic drive system involves the use of an automated cooling system for regulating the temperature of the working fluid, by altering the flow rate of the coolant.

Most developed to date mathematical models of electrohydraulic drive systems do not take into account the changing of the system parameters due to alteration of the working fluid's temperature. This prevents the adequate modeling of the processes occurring in the real system. In [5] is developed a nonlinear mathematical model of electrohydraulic drive system for reciprocating motion of the actuator, and the ongoing dynamic processes are studied in various modes. The issue of energy analysis and efficiency improvement of pumping systems for transport of fluids is discussed in [6].

The paper inhere presents a new developed nonlinear mathematical model describing the operation of electrical tracking system with a rotary actuator, reflecting the impact of working fluid's temperature on some parameters of the system. The compiled analogue model of the system allows studies in various operation modes.

MATHEMATICAL MODEL OF TRACKING ELECTROHYDRAU–LIC DRIVE SYSTEM

The setup of a tracking electrohydraulic drive system with a rotary actuator and an integrated heat exchanger is presented in Fig. 1. It consists of the following elements: 1 – volumetric pump; 2 – relief (safety) valve; 3 – high–pressure filter with a safety valve; 4 – servo valve; 5, 9 – pressure pipeline; 6 – hydraulic motor; 7 – object of regulation; 8 – speed sensor; 10 – setting device; 11 – summing device; 12 – electronic controller; 13 – manually operated hydraulic control valve (distributor); 14, 16, 17 – return valves; 18 – centrifugal pump; 19 – tank; 20 – low pressure filter.

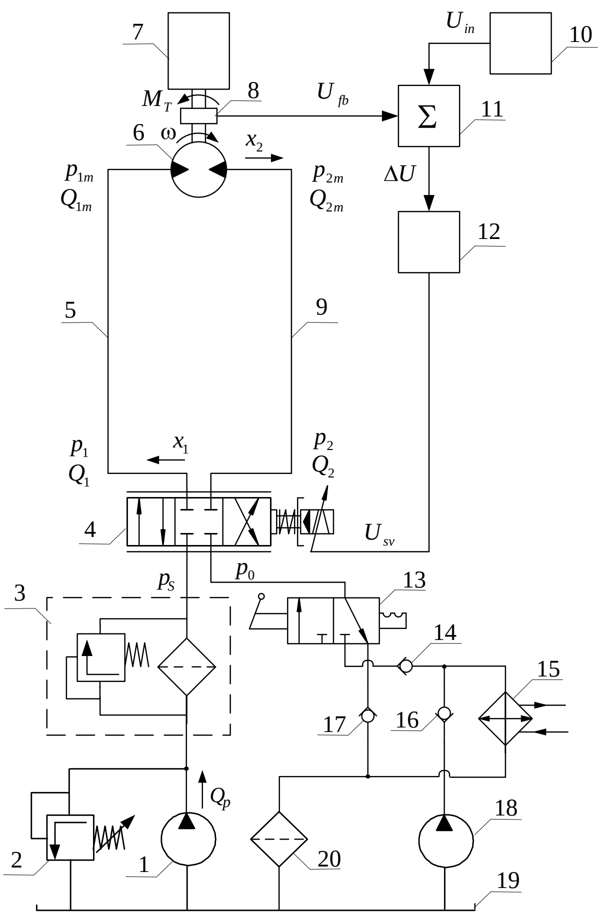

Figure.1. Setup of a tracking electrohydraulic drive system

Valve 13 and pump 18 can be used to implement several modes of cooling of the working fluid:

- Right section of the distributor 13 is working while the pump 18 is switched off. The working fluid passes through return valve 17, filter 20 and enters tank 19;
- Left section of the distributor 13 is working while the pump 18 is switched off. The working fluid passes through return valve 14, heat exchanger 15, filter 20 and enters tank 19;
- Right section of the distributor 13 is working while the pump 18 is switched on. The working fluid is sucked by pump 18 from the tank 19, passes through return valve 16, the heat exchanger 15 and together with the working fluid flowing through the right–hand section of the distributor 13 and the return valve 17, passes through filter 20 and enters the tank 19.

1. Equation for hydraulic motor's rotating speed:

$$J\frac{d\omega}{dt}+k_f\,\omega=w_m\left(p_{1m}-p_{2m}\right)-M_T \qquad (1)$$

where:

J – referred moment of inertia of hydraulic motor's shaft;

k_f – coefficient of hydraulic friction;

w_m – specific volume of the hydraulic motor;

ω – angular velocity of hydraulic motor's shaft;

p_{1m}, p_{2m} – pressures from the left (the right side) of the hydraulic motor.

2. Equation of the summing device:

$$\Delta U=U_{in}-U_{fb} \qquad (2)$$

where:

$$\Delta U_{fb}=k_{fb}\omega;$$

U_{in} – setting voltage;

k_{fb} – feedback coefficient;

ΔU – error.

3. Equation of the electronic PID–controller:

$$U_{sv}=k\left(\Delta U+\frac{1}{T_I}\int\Delta U\,dt+T_D\frac{d\Delta U}{dt}\right) \qquad (3)$$

where:

k – amplification factor of the controller;

T_I, T_D – time constants of integration and differentiation of the controller.

4. Equation of servo valve:

$$T_{sv}\frac{dx_{sv}}{dt}+x_{sv}=k_{sv}U_{sv} \tag{4}$$

where:

k_{sv}, T_{sv} – servo valve amplification factor and time constant;

x_{sv} – servo valve's plunger movement, $x_{min} \le x_{sv} \le x_{max}$.

5. Equations for the passing flow rates through the servo valve:

$$Q_1=\begin{cases}\mu_1\pi d\,x_{sv}\sqrt{\dfrac{2(p_S-p_1)}{\rho}}, & 0\le x_{sv}\le x_{max}\\[2ex] \mu_1\pi d\,x_{sv}\sqrt{\dfrac{2(p_1-p_0)}{\rho}}, & x_{min}\le x_{sv}<0\end{cases} \tag{5}$$

$$Q_2=\begin{cases}\mu_2\pi d\,x_{sv}\sqrt{\dfrac{2(p_2-p_0)}{\rho}}, & 0\le x_{sv}\le x_{max}\\[2ex] \mu_2\pi d\,x_{sv}\sqrt{\dfrac{2(p_S-p_2)}{\rho}}, & x_{min}\le x_{sv}<0\end{cases}$$

where:

Q_1, Q_2 – passing flow rates through the servo valve;

μ_1, μ_2 – flow rate factors;

ρ – hydraulic oil density;

d – diameter of servo valve's plunger;

p_S, p_0 – feeding and merging pressure;

p_1, p_2 – pressures at the beginning and the end of the pressure pipelines 5 and 9 (Fig. 1);

6. Equation of the passing flow rates *through the hydraulic motor:*

$$Q_1=w_m\omega+\frac{V_1}{B}\frac{dp_{1m}}{dt}sign(\omega) \tag{6}$$

$$Q_{2m}=w_m\omega-\frac{V_2}{B}\frac{dp_{2m}}{dt}sign(\omega)$$

where:

Q_1 , Q_2 – incoming and outgoing flow rates to (from) the hydraulic motor;

$V_1=V_2=q_m/2$;

q_m – hydraulic motor's working volume, $q_m=2\pi w_m$.

7.** Equation of liquid transport in pipeline 5 (Fig. 1)**:

$$\frac{\rho L_1}{A_{T1}}\frac{dQ_1}{dt}+\frac{f_1\rho L_1}{2\,d_1\,A_{T1}^{\,2}}|Q_1|Q_1=p_1-p_{1m} \tag{7}$$

where:

f_1 – hydraulic friction factor;

L_1, d_1 – length and internal diameter of the pipeline;

A_{T1} – cross–section area of the pipeline, $A_{T1}=\pi d_1^{\,2}/4$.

8. Equation for the passing flow rate through pipeline 9 (Fig. 1):

$$\frac{\rho L_2}{A_{T2}}\frac{dQ_2}{dt}+\frac{f_2\rho L_2}{2\,d_2\,A_{T2}^{\,2}}|Q_2|Q_2=p_{2m}-p_2 \tag{8}$$

where:

f_2 – hydraulic friction factor;

L_2, d_2 – length and internal diameter of the pipeline;

A_{T2} – cross–section area of the pipeline, $A_{T2}=\pi d_2^{\,2}/4$.

The change in temperature of the working fluid affects the values of the following parameters: density, dynamic viscosity, kinematic viscosity, elastic modulus and coefficient of hydraulic friction.

9. Equation of dynamic viscosity as a function of temperature and pressure.

Pressure in hydraulic systems normally is less than 40MPa, and therefore for the dynamic viscosity it is obtained [4]:

$$\mu(p,T)=a\,e^{\left[\frac{b}{(T+273.17)-c}\right]}e^{\left[\frac{p}{a_1+a_2T}\right]} \tag{9}$$

where:

a, b, c, a_1, a_2 – constants;

p, T – working fluid pressure and temperature.

10. Dependence of the volumetric modulus of elasticity on the working fluid temperature.

In general, the volumetric modulus of elasticity is a function of temperature and pressure of the working fluid $B=B(T,p)$. At pressures up to 30MPa the following dependence can be used [1]:

$$lg\frac{B_{T_1}}{B_{T_2}}=a(T_2-T_1) \tag{10}$$

where:

T_1, T_2 – different working fluid temperatures;

B_{T_1}, B_{T_2} – modules of volume elasticity respectively at temperatures T_1 and T_2;

$a = 2\times10^{-3}$.

11. Equation of the relationship between the density and the temperature of the working fluid [2]:

$$\rho = \rho_{15} - \rho_{15}\alpha_{p15}(T - 15) \qquad (11)$$

where:

ρ, ρ_{15} – working fluid density at a temperature T and at $T = 15^o C$;

$\alpha_{p15} = 0.0007$.

Kinematic viscosity $\upsilon = \frac{\mu}{\rho}$ of working fluid required for the calculation of the coefficients of linear resistance f_1 and f_2 in equations (7) and (8) are determined using the relationships (9) and (11).

The system of equations (1) through (11) describes the operation of the electro–tracking system of Fig. 1 in dynamic mode, taking into account the influence of the working fluid's temperature.

MATHEMATICAL MODEL OF AN AUTOMATED SYSTEM FOR CONTROL OF WORKING FLUID'S TEMPERATURE

Fig. 2 shows the principal scheme of an automated system for control of the temperature of the working fluid as a part of the hydraulic drive system.

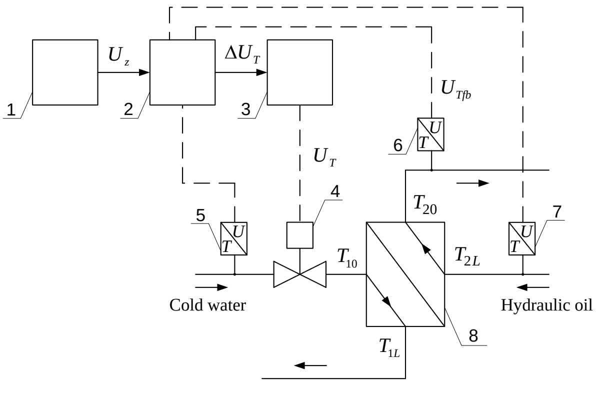

Figure 2. Scheme of the automatic system for regulating the temperature of the working fluid

The system consists of the following elements: 1 – setting device (ZU); 2 – summing device (PC); 3 – electronic controller (EP); 4 – valve with an actuator; 5, 6, 7 – temperature sensor; 8 – heat exchanger.

Heat exchanger

Fig. 3 shows a principle diagram of the relevant "tube in tube" type heat exchanger as an object with distributed parameters.

1. Equation of the heat flow in the tube (cold fluid):

$$\frac{\partial T_1(z,t)}{\partial t} + v_1\frac{\partial T_1(z,t)}{\partial z} = \beta_1[(T_{s1}(z,t) - T_1(z,t)] \qquad (12)$$

where:

v_1 – average speed of the cooling fluid;

T_1 – temperature of the cooling fluid;

$A_1 = \pi d_0^2/4$ – cross–section area of the pipeline;

d_0 – internal pipeline diameter;

$A_{s1}\Delta z = \pi d_0 \Delta z$ – heat exchange area;

($A_{s1} = \pi d_0$);

$\beta_1 = \frac{\alpha_1 A_{s1}}{\rho_1 A_1 c_{p1}}$ – heat exchanger parameter;

α_1 – heat exchange factor;

ρ_1, c_{p1} – density and specific heat capacity of the cooling fluid;

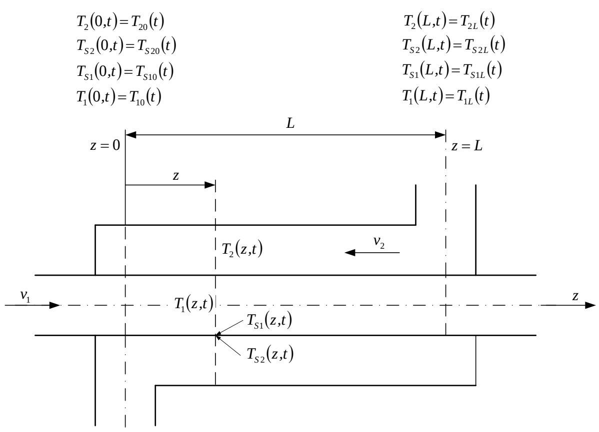

Figure 3. Heat exchanger principle diagram

2. Equation of the heat flow in the inter–tubular space (hot fluid):

$$\frac{\partial T_2(z,t)}{\partial t} + v_2\frac{\partial T_2(z,t)}{\partial z} = \beta_2[(T_{s2}(z,t) - T_2(z,t)] \qquad (13)$$

where:

v_2 – average speed of the cooling fluid in the inert–tubular space;

T_2 – temperature of the cooling fluid;

$A_2 = \pi(D^2 - d^2)/4$ – cross–section area of the channel;

$A_{s2}\Delta z = \pi(d + \delta)\Delta z$ – heat–exchanging surface;

$A_{s2} = \pi(d + \delta)$;

δ –Wall thickness;

D – Heat exchanger external diameter;

$\beta_2 = \dfrac{\alpha_2 A_{s2}}{\rho_2 A_2 c_{p2}}$ – heat exchanger parameter;

α_2 – heat exchange factor;

ρ_2, c_{p2} – density and specific heat capacity of the cooling fluid

3. Equation of the tube wall

$$\frac{\partial T_s(z,t)}{\partial t} = \frac{\alpha_2 A_2}{k}[(T_2(z,t) - T_s(z,t)] + \frac{\alpha_1 A_1}{k}[(T_s(z,t) - T_1(z,t)] \quad (4)$$

where:

T_s – average temperature of the pipe wall;

k – coefficient.

4. Equation for the average (for the tube thickness) temperature:

$$T_s = \frac{T_{s1} + T_{s2}}{2} \quad (15)$$

Automatic controller

During the compiling of the mathematical model of the system for controlling the temperature in Figure 1 it is assumed that to the summing device 2 is provided a signal only from the setting device 1 and from the temperature sensor 6.

5. Equation of the summing device:

$$\Delta U_T = U_z - U_{Tfb} \quad (16)$$

where:

U_z – setting voltage;

ΔU_T – voltage at the output of the summing device;

U_{Tfb} – feedback voltage, $U_{Tfb} = k_{fb} U_{st}$;

U_{st} – voltage from the temperature sensor.

6. Equation of the electronic PID controller:

$$U_T = k_T \left(\Delta U_T + \frac{1}{T_{TI}} \int \Delta U_T \, dt + T_{TD} \frac{d\,\Delta U_T}{dt} \right) \quad (17)$$

where:

U_T – voltage at the output of the electronic controller;

k_T, T_{TI}, T_{TD} – amplification factor, time constants of integration and differentiation of the electronic controller.

7. Equation for the motion of the actuator (DC motor)

$$T_\gamma \frac{d\gamma}{dt} + \gamma = \int k_\gamma U \quad (18)$$

where:

γ – angle of rotation of the shaft of the motor;

k_γ, T_γ – amplification factor and time constant of the DC motor.

8. Equation for the passing flow through the valve:

$$Q = \mu\, S(\gamma) \sqrt{\frac{2\,\Delta p}{\rho_1}} \quad (19)$$

where:

Q – passing flow rate through the valve, $Q = v_1 A_1$;

μ – Flow factor;

$S(\gamma)$ – internal cross–section area of the valve, $S(\gamma) = S_0 + k_s\,\gamma$;

S_0 – internal cross–section area of the valve in steady state mode;

k_s – coefficient;

Δp – Pressure drop, created by the valve.

9. Equation of the temperature sensor:

$$T_{st} \frac{dU_{st}(t)}{dt} + U_{st}(t) = k_{st}\, T_{20}(t - \tau) \quad (20)$$

where:

k_{st}, T_{st} – amplification factor and time constant of the temperature sensor;

τ – time delay of the temperature sensor.

CONCLUSIONS

The developed nonlinear mathematical model can be used for studying the dynamic processes in an electrical tracking system (Fig. 1) and respectively in automated systems for control of the temperature using a "tube in tube" type heat exchanger (Fig. 2).

By presenting the mathematical model in non–dimensional form it is possible to create an analogue model of the system, on the basis of which to be performed simulations in different modes and temperatures of the working fluid. This will be a subject of further studies on the topic.

In order to maintain a constant temperature of the working fluid or its change in a very narrow range it is necessary to carry out structural and parametric optimization of the system by pre–defined quality criteria. For better matching of the results from the numerical and the physical experiments it is necessary prior to be defined certain experimental dependences as a function of the working fluid's temperature for the following parameters: density, dynamic viscosity, kinematic viscosity, modulus of volume elasticity and coefficient of hydraulic friction.

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