Simulation of the power unit of the automatic electrohydraulic drive with volume regulation

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Summary. The mathematical model of the dynamic characteristics of the power unit of the automatic electrohydraulic drive volume regulation is developed. A block diagram of the control signal transmission is represented. Transfer function of the drive is received.
Key words: pump, motor, pressure, flow, volume control, a block diagram, transfer function.

INTRODUCTION

Modern technologies of materials processing, plastic molding machine designs placing ever-increasing demands on the technical and functional characteristics of machine tools and special process equipment. The quality of the products for machining, forming, plastic molding depends on the feasibility of optimal laws of motion of blades, precision control of their movements, stability, speed with variable load. Therefore important the scientific and technical task is to improve the accuracy and extend the functionality of machines and equipment for processing materials.

Achievement of any of the kinematics of the working body, the possibility of program implementation optimal laws of motion provided by the use of automatic hydraulic transmission and, in particular, auto electrohydraulic drive (EHD) with volume control in equipment capacity exceeding 8 kW. However, there are currently no generic mathematical model of the work processes that take place in the drive, there is no generally accepted methods for calculating EHD adapted to drive machine tools and special equipment for material processing, enables the assessment and selection of drive components and devices, to predict its static and dynamic characteristics.

The power part of the electrohydraulic servo drive (EHD) includes a positive displacement pump with adjustable pitch, and accessories, the executive hydraulic motor displacement type [1, 2]. The most widely used in servo-hydraulic actuator has axial-piston pumps, which flow is regulated by a change in the angle of the cylinder, or changing the angle of inclination of the puck. As actuation hydraulic cylinders typically used with linear transmission output link, hydraulic cylinders with a rotary motion of the output link and axial piston and radial piston motors. Auxiliary devices include valves, filter, pump and tank system of feeding with work fluid of hydraulic power unit.

The aim of this work is to develop mathematical models of dynamic characteristics of the power unit of the automatic electrohydraulic drive with volume control and obtaining the transfer function of the drive by the control signal.

OBJECTS AND PROBLEMS

At fig. 1 a schematic diagram of a typical power unit of the drive with volume control containing two axial piston hydraulic machines: the main hydraulic pump 2 and 5 is presented. The pump shaft is driven by an asynchronous motor 1. Pump capacity is controlled by varying the angle of the cylinder block (or angle washer) with the 3, which can also be power, consisting of cylinder and valve. The pipe is connected to motor with two pipelines 4. Motor shaft via a gear 6 is connected to the governing board 7. For the leakages of fluid is an auxiliary pump (usually gear or plate) 13 driven in rotation by the shaft of the main pump. If the angle of the cylinder (tilt washer) of the ground the pump is controlled by steering, the auxiliary pump is also used to supply hydraulic fluid under pressure. The pressure in the pressure line of the auxiliary pump is supported overflow valve 10. This highway two make-up valve 9 is connected to a pipeline linking the main pump and motor. When the pressure in one of the pipes below the value of the corresponding make-up valve opens and passes the pressurized fluid from the pressure line of the auxiliary pump for as long as the line does not restore the required level of pressure. After that, make-up valve by the pressure in the pipe is closed. Make-up valves must maintain such a minimum pressure in the pipeline not to occur a cavitation in basic pump. To do this, set the required pressure in the pressure line of the auxiliary pump by adjusting the tension of the spring (set pressure) relief valve 10.

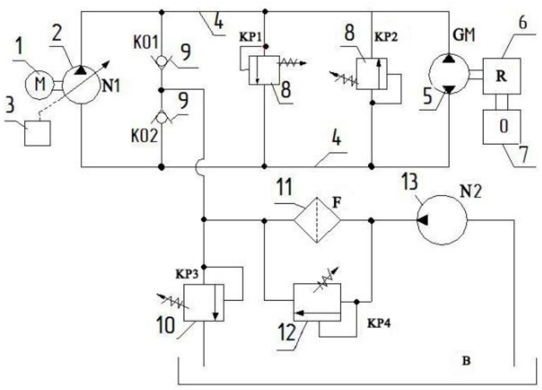

Fig. 1. The scheme of the power unit with interior

EHD regulation

From the occurrence of excessively high pressure hydraulic lines are protected by two safety valves 8. During excessive rise in pressure in one of the pipelines can open a safety valve to leak into another pipeline with low pressure. In the discharge of the auxiliary pump also has a pressure relief valve 12, which protects the pump from pressure build-up clogging the filter 11.

Let`s note symbols of the hydraulic machines and hydraulic units in fig. 1. Here H1…H2 - pumps, GM - motor; KO1…KO2 - check valves; KR1...KR4 - safety valves, F - filter.

Before constructing a mathematical description of the power of the EHD surround regulation, form the design scheme, taking into account the following basic assumptions of:

1. Asynchronous motor 1 rotates shaft with angular velocity Ω_n, the value of which is independent power developed by the pump.
2. When operating hydraulic pressure in the pipelines 4 do not reach the values for opening the safety valves 8.
3. Pressure p_{nn} in the line before the make-up valve is kept constant.
4. Efforts, overcome by hydraulic motor 5 in the management of object 7, may be represented by the sum of points given to the shaft of the motor inertia loads, load positioning and hydraulic friction.
5. Pipelines will take as short to allow them to neglect the inertia of the fluid and pressure loss due to friction.

The design scheme is shown in fig. 2. In this circuit, the arrows show on the direction of fluid flow at the moment when the pressure is greater than the pressure p_1 and p_2.

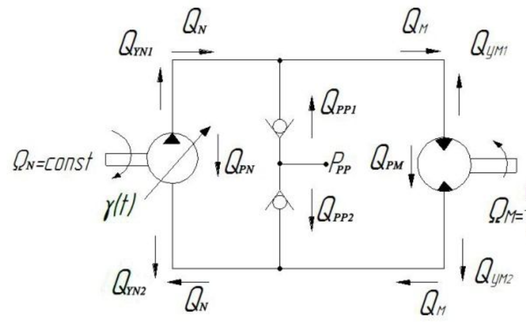

Fig. 2. The design scheme of the hydraulic power

With all the above assumptions made obtaining a linear mathematical model of the power of the EHD with interior control prevents one essentially nonlinear characteristic that determines the dependence of flow and recharge Q_{nn1} and Q_{nn2} through the make-up valve from the pressures p_1 and p_2 in the pipelines. If the pressure in the pipe is below the supply pressure p_{nn} to the make-up valves, small changes can apply pressure ratio

$$Q_{nn1} = k_{kl}\left(p_{nn} - p_1\right); \quad (1)$$

$$Q_{nn2} = k_{kl}\left(p_{nn} - p_2\right); \quad (2)$$

where: $k_{кл}$ - the conductivity of makeup valve [3, 4].

If the pressure in the pipeline exceeds the charge pressure p_{nn}, the

$$Q_{nn1}=Q_{nn2}=0, \quad (3)$$

as make-up valves are closed by the pressure in the pipelines.

At steady state hydraulic drive, for which the unloaded motor shaft does not rotate, the pressure in the pipes due to leakage of fluid from the pump and motor and final values of the conductivity of valves installed below the p_{nn}. With fluctuations in each pipeline after a certain make-up valve on one half cycle at low pressure enters the amount of fluid is compensated not only leaks, but the compressibility of the liquid. In the next half-cycle of the compression of large volume of fluid in the pipe happens, which leads to an increase in its pressure. Flow of fluid through the make-up in the hydraulic valves is accompanied by increase in the average for the period of pressure fluctuations in the system or in increased levels of pressure in them. The amount of fluid received in the pipelines for the period of oscillation depends on the amplitude of fluctuations in pressure p_1 and p_2, so the average for the period of the valve conductance amplitude pressure in the pipes.

In the study of the dynamics of the hydraulic drive with small deviations of the variables from the steady-state values the level of pressure in the lines can be adopted under p_{nn} pressure, so is permissible to use the linear relationship (1, 2). In the study of the dynamics of the hydraulic drive with large changes of variables have to take into account the non-linear characteristics of make-up valves.

For the time, when a small deviation of the cylinder (or the clone disk) from the equilibrium pump delivers fluid through the pipeline to the pressure p_1 and sucks fluid from the pipeline to the pressure p_2, the flow equation can be written using the D. Popov`s approach [4], as follows:

for pipeline pressure p_1 pump flow

$$Q_n = Q_m + Q_{nn} + Q_{nm} + Q_{yn1} + Q_{ym1} + Q_{cg1} - Q_{nn1}; \quad (4)$$

for a pipeline with pump flow pressure p_2

$$Q_n = Q_m + Q_{nn} + Q_{nm} - Q_{yn2} - Q_{ym2} - Q_{cg2} + Q_{nn2}. \quad (5)$$

In equations (4) and (5) the costs Q_{cg1} and Q_{cg2} are the components of the distribution moves that are associated with the compensation of compressibility. The remaining components designated in accordance with the design scheme (fig. 2). To simplify expressions, determining the coefficients in the following equations, we consider the pump and motor hydraulic machines of the same type, such as axial piston, differing only in the fact that the pump is regulated by the angle cylinder (or disk) is not regulated. In this case, you can take

$$Q_{nn} = Q_{nm} = Q_{per};$$
$$Q_{yn1} = Q_{ym1} = Q_{ym1};$$
$$Q_{yn2} = Q_{ym2} = Q_{ym2}.$$

Given these relations, we define the components of the pump flow in the form of

$$Q_m = \frac{q_m}{2\pi}\Omega_m = \frac{q_m}{2\pi}\frac{d\alpha}{dt}; \quad (6)$$

$$Q_{per} = k_{per}(p_1 - p_2); \quad (7)$$

$$Q_{ym1} = k_{ym} p_1; \quad (8)$$

$$Q_{ym2} = k_{ym} p_2; \quad (9)$$

where: q_m - working volume of the hydraulic motor; Ω_m - motor shaft angular velocity; α - the angle of rotation of the shaft motor; k_{per} - conductivity slits through which the pump and drive have a transmission of the fluid from the high pressure in the cavity with a low pressure; k_{ym} - the conductivity of the slits, which the leakage of fluid from the pump and motor.

Costs Q_{nn1} and Q_{nn2} determined by the ratio (1) and (2), and the costs Q_{cg1} and Q_{cg2} assuming the pipes is hard to find the following relationships [5]

$$Q_{cg1} = \frac{W_0}{E_g}\frac{dp_1}{dt}; \quad (10)$$

$$Q_{cg2} = \frac{W_0}{E_g}\frac{dp_2}{dt}; \quad (11)$$

where: W_0 - the internal volume of the pipeline with the connected volume of the cavity pump and motor;

E_g - bulk modulus of the working fluid.

Substitute the components of the costs according to (1, 2, 6-11) in equation (4, 5). Then add these equations and transform

$$Q_m = \frac{q_m}{2\pi}\frac{d\alpha}{dt} + 2k_{per}(p_1 - p_2) + k_{yt}(p_1 - p_2) + \\ + \frac{k_{kl}}{2}(p_1 - p_2) + \frac{W_0}{2E_g}\frac{d(p_1 - p_2)}{dt}. \quad (12)$$

Ideal pump flow Q_n in the form depending on the angle of inclination γ of cylinder blocks or angle swash plate pump

$$Q_n = \frac{q_n}{2\pi}\Omega_n, \tag{13}$$

where: $q_н$ - the pump displacement.

For axial piston pump

$$q_n = F_n z_n D_n tg\gamma, \tag{14}$$

where: F_n the working area of one piston (plunger) pump; z_n - the number of pistons; D_n - the diameter of the circle on which the axis of the piston pump.

As can be seen, the function is nonlinear. For small deviations of cylinder blocks (spacers) of the pump from the neutral position of the specified function can be linearized, and the equation (14) is written in the form

$$Q_n = k_{Q\gamma}\gamma, \tag{15}$$

where:

$$k_{Q\gamma} = \frac{\partial Q_n}{\partial \gamma}.$$

For axial piston pump

$$k_{Q\gamma} = \frac{F_n z_n D_n \Omega_n}{2\pi}.$$

Applying (15), we reduce (12) to the form

$$\frac{q_m}{2\pi k_{Q\gamma}}\frac{d\alpha}{dt} + \frac{W_0}{2E_g k_{Q\gamma}}\frac{dp_m}{dt} + \frac{k_\Sigma}{k_{Q\gamma}}p_m = \gamma, \tag{16}$$

where:

$$k_\Sigma = k_{yt} + 2k_{per} + \frac{k_{kl}}{2};$$

$$p_m = p_1 - p_2. \tag{17}$$

In equation (16), except for the input and output values γ and α, found of changing the time differential pressure p_m, which is dependent on hydro-motor overcomes the load. When the inertial load value p_m determined torque M_m, which is included in the equation of the rotational movement of the shaft motor

$$M_M - M_{mr1} - M_{mr2} - M_{poz} = J\frac{d^2\alpha}{dt^2}, \tag{18}$$

where: J - moment of inertia of the rotating parts of the motor to the shaft (given moment of inertia of the load and the rotor motor). In the future, this value will be called reduced moment of inertia motor.

Torque for the bulk hydraulic [6-8] is defined by the relation

$$M_m = \frac{q_м}{2\pi}p_m. \tag{19}$$

Friction torque M_{mp1} is created by friction in the motor. In general, the friction in the hydraulic motor can be mixed. To simplify the mathematical model of hydro-drive we will consider only the hydraulic friction [9, 10], setting

$$M_{mr1} = k_{mr1}\frac{d\alpha}{dt}, \tag{20}$$

where: k_{mp1} is calculated from the slope of the approximating performance $M_{mr1} = M_{mr1}(\Omega_m)$.

Friction torque M_{mr2} arising due to the friction of the load, consider the analogous dependence

$$M_{mr2} = k_{mr2}\frac{d\alpha}{dt}. \tag{21}$$

Moment due to the positional load

$$M_{poz} = k_{poz}\alpha, \tag{22}$$

where: k_{poz} - the rigidity of positional load.

Using (19-22), from (18) that

$$\frac{2\pi J}{q_m}\frac{d^2\alpha}{dt^2} + \frac{2\pi k_{mr}}{q_m}\frac{d\alpha}{dt} + \frac{2\pi k_{poz}}{q_m}\alpha = p_m, \tag{23}$$

where: $k_{mr} = k_{mr1} + k_{mr2}$.

Considering simultaneously the equations (16) and (23) we find

$$\frac{\pi J W_0}{E_g q_m k_{Q\gamma}}\frac{d^3\alpha}{dt^3} + \left(\frac{\pi k_{mr}W_0}{E_g q_m k_{Q\gamma}} + \frac{2\pi k_\Sigma J}{q_m k_{Q\gamma}}\right)\frac{d^2\alpha}{dt^2} + \\ + \frac{q_m}{2\pi k_{Q\gamma}}\left(1 + \frac{2\pi^2 k_{poz}W_0}{E_g q_m^2} + \frac{4\pi^2 k_\Sigma k_{mr}}{q_m^2}\right)\frac{d\alpha}{dt} + \frac{2\pi k_\Sigma k_{poz}}{q_m k_{Q\gamma}}\alpha = \gamma. \tag{24}$$

In real hydrodrives usually

$$1 + \frac{2\pi^2 k_{poz}W_0}{E_g q_m^2} + \frac{4\pi^2 k_\Sigma k_{mr}}{q_m^2} \approx 1. \tag{25}$$

Therefore, instead of (24) we have

$$\frac{\pi J W_0}{E_g q_m k_{Q\gamma}}\frac{d^3\alpha}{dt^3} + \left(\frac{\pi k_{mr}W_0}{E_g q_m k_{Q\gamma}} + \frac{2\pi k_\Sigma J}{q_м k_{Q\gamma}}\right)\frac{d^2\alpha}{dt^2} + . \\ + \frac{q_m}{2\pi k_{Q\gamma}}\frac{d\alpha}{dt} = \gamma - \frac{2\pi k_\Sigma k_{poz}}{q_m k_{Q\gamma}}\alpha. \tag{26}$$

We consider the following parameters:

time constant hydraulic

$$T_{gp} = \frac{q_m}{2\pi k_{Q\gamma}}. \tag{27}$$

time constant motor

$$T_m = \sqrt{\frac{2\pi^2 JW_0}{q_m^2 E_g}}. \qquad (28)$$

the relative damping motor

$$\zeta = \frac{\pi\left(2JE_g k_\Sigma + k_{mr} W_0\right)}{\sqrt{2JW_0 E_g q_m^2}}. \qquad (29)$$

factor own feedback

$$k_{coc} = \frac{2\pi k_\Sigma k_{poz}}{q_m k_{Q\gamma}}. \qquad (30)$$

As can be seen from this expression, the coefficient of its own feedback part of the hydraulic power unit due to the combined action of hydraulic positional load and no-leak of hydraulic machines.

Given the input parameters (26) the Laplace transform [11, 12] to the form

$$T_{gp} s\left(T_m^2 s^2 + 2\zeta_m T_m s + 1\right)\alpha(s) = \gamma(s) - k_{coc}\alpha(s). \qquad (31)$$

Block diagram of the power part of EHD according to equation (31) are shown in fig. 3.

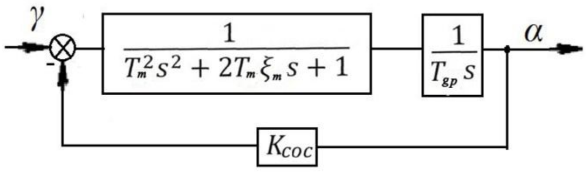

Fig. 3. Block diagram of the power part of EGP

In the absence of positional load (22) or seals for hydraulic, or insignificant moment of the forces positional load and high hydraulic leak (which is usually the case in practice) coefficient of the own feedback is negligible. It should be noted that this ratio in the event of significant influence can be included in the external feedback EHD.

Then, a block diagram of the power of the drive will be as shown in fig. 4. This scheme is, in fact, reflects the transfer of the control signal - the angle to clone the cylinder (or puck) γ - and its influence on the output signal - the angle of rotation of the shaft motor α.

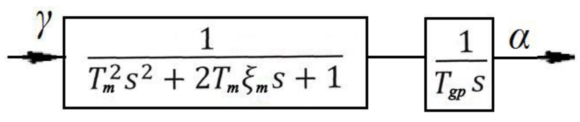

Fig. 4. The block diagram of the control signal EHD in the power section with interior control

In the future, consider the block diagram of the power part of EHD with the volume control as shown in fig. 4. According to this block diagram is obtained for the transfer function of the angle of rotation of the drive motor shaft angle α on the cylinder (or puck) γ

$$W_{\alpha\gamma}(s) = \frac{\alpha(s)}{\gamma(s)} = \frac{1}{T_{gp} s\left(T_m^2 s^2 + 2\zeta_m T_m s + 1\right)}. \qquad (32)$$

CONCLUSIONS

Thus, the mathematical model of the dynamic characteristics of the power unit automatic electrohydraulic drive volume regulation is developed. The block diagram of the control signal is presented. Transfer functions for the drive output shaft angle motor with control signal (the slope of the cylinder or disk) is received.

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МОДЕЛИРОВАНИЕ СИЛОВОЙ ЧАСТИ АВТОМАТИЧЕСКОГО ЭЛЕКТРОГИДРАВЛИЧЕСКОГО ПРИВОДА С ОБЪЕМНЫМ РЕГУЛИРОВАНИЕМ

Владимир Соколов, Наталья Азаренко, Яна Соколова

Аннотация. Разработана математическая модель динамических характеристик силовой части автоматического электрогидравлического привода с объемным регулированием. Представлена структурная схема передачи управляющего сигнала. Получена передаточная функция привода.

Ключевые слова: насос, гидромотор, давление, расход, объемное регулирование, структурная схема, передаточная функция.