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DYNAMIC PARAMETERS DETERMINATION OF THE HYDRO-MECHANICAL SYSTEM ELECTRIC DRIVE CONTROL OBJECT

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The research objective is to develop a mathematical model of the the drive control object of lowering and lifting operations and feeding the rig drill, on the basis of which it is possible to quantitatively assess the influence of the elastic properties of hydraulic and cable-polyspast systems on the dynamic processes in the electro-hydro-mechanical system.

The research was carried out on the basis of the apparatus of mathematical analysis using ordinary differential equations and the Laplace transformation with the representation of dynamic links in the form of transfer functions, which allows to quantitatively determine the influence of the control object parameters on the quality of transient processes.

Dynamic characteristics of the hydraulic system are obtained, which is based on the basis of fundamental ratios: balances of lubricant consumption, torques and powers on the shaft of the hydro engine. A mathematical model of the control object was obtained and analyzed using standard dynamic links with known characteristics, which allow synthesizing the system of automatic control of the drive of lowering and lifting operations and the drilling rod supply.

From the obtained mathematical model analysis of the feed drive of the lowering and lifting operations control object, it was established that in the control object, in order to implement a rational method of controlling the drilling process with the support of the power flow in the zone of rock destruction at a constant level, it is necessary to control, in addition to traditional coordinates – current and motor speed, as well as technological coordinates – pressure in the hydraulic system, the speed of the hydro motor shaft and the linear speed of the position movement drilling rod, which requires the appropriate switching on of the control circuits.

When developing a system for regulating the feed drive and lowering and lifting operations, it is necessary to provide for the control of current and limit values of technological parameters in the process of moving the drilling rod: pressure in the hydraulic system, the frequency of rotation of the shaft of the hydro motor and the linear speed of the drilling rod movement, as well as the influence of additional fractional-rational functions on the quality of control of the drive system.

Key words: *balance of lubricant consumption, torgues balance, transfer functions, structural diagram.*

Хілов Віктор, Дифорт Віктор, Павлишина Олена. Визначення динамічних параметрів об’єкта керування електропривода гідромеханічної системи

Метою дослідження є розробка математичної моделі об’єкта керування приводу спуско-підйомних операцій і подачі поставу, на підставі якої можливо кількісно оцінити вплив пружних властивостей гідро-та канатно-поліспастої систем на динамічні процеси електрогідромеханічної системи.

Дослідження здійснено на підставі апарата математичного аналізу з використанням звичайних диференціальних рівнянь і перетворення Лапласу й уявленням динамічних ланок у вигляді передатних функцій, що дає змогу кількісно визначити вплив параметрів об’єкта керування на якість перехідних процесів.

Одержані динамічні характеристики гідросистеми, які ґрунтуються на базі фундаментальних співвідношень: балансів витрати мастила, моментів і потужностей на валу гідродвигуна. Отримана та проаналізована математична модель об’єкта керування з використанням стандартних динамічних ланок із відомими характерниками, які дають змогу синтезувати систему автоматичного керування приводом спуско-підйомних операцій та подачі поставу.

З аналізу одержаної математичної моделі об'єкта керування приводом подачі та спуско-підйомних операцій встановлено, що в об'єкті керування для реалізації раціонального способу керування процесом буріння з підтримкою потоку потужності в зоні руйнування породи на постійному рівні потрібно контролювати, крім традиційних координат – струму й частоти обертання двигуна, ще і технологічні координати – тиск у гідросистемі, частоту обертання вала гідродвигуна й лінійну швидкість переміщення поставу, що потребує відповідного вмикання контурів управління.

Під час розробки системи регулювання приводом подачі і спуско-підйомних операцій потрібно передбачити контролювання поточних і граничних значень технологічних параметрів у процесі переміщення бурового поставу: тиску в гідросистемі, частоти обертання вала гідродвигуна та лінійної швидкості переміщення поставу, а також вплив додаткових дробово-раціональних функцій на якість керування системи управління.

Ключові слова: *баланс витратив мастила, баланс моментів, передатні функції, структурна схема.*

Introduction. The machine for drilling explosive wells, developed by the Novo-Kramatorsk Machine-Building Plant, is equipped with a drive system for lowering and lifting operations (LLP), which includes a hydraulic pump and a hydraulic motor (HM). The hydraulic pump is driven by an adjustable electric drive using a frequency converter – induction motor system. The tension of the rope and cable polistpast system (CPS) is carried out by HM [1; 2].

Research material and method. A frequency converter, an asynchronous motor with an auto-regulation system, a hydraulic pump, a hydraulic pump, a hydraulic motor are included in the control object of the LLO drive. A new constructive link in the LLO drive is a hydraulic pump with a high pressure motor and a fully adjustable drive based on the frequency converter – asynchronous motor system. If the transfer functions of the electric drive are sufficiently well researched, then the transfer functions of the hydraulic system with the CPS in the form that is necessary for the construction of the control system are not given in the known literature [3; 4].

In the kinematic connection between the drive motor and the drilling projectile during the translational movement of the drilling rod, a hydraulic pump, a hydraulic pump with a gearbox and a hydraulic pump are included. An incompressible fluid (lubricant) is used in the hydraulic system. Ropes and pipelines are subject to elastic deformations. At the same time, the rotation frequency of the motor shaft and the linear speed of the drilling projectile reduced to the motor shaft in non-stationary modes are not equal. The drive system uses a high-speed electric drive with a bandwidth of the rotation frequency circuit up to 200 rad/s. At the same time, the frequency of the natural elastic oscillations of the CPS and the hydraulic drive falls within the bandwidth of the speed control loop. Failure to consider the flexibility of pipelines and ropes when analyzing the control object and synthesizing the control system can lead to a discrepancy between the expected and actual control quality.

Let's develop a mathematical model of the SPO drive control object. Fig. 1 shows the kinematic diagram of the electro-hydromechanical system of supply to the hole during drilling and LLO.

The rotor of the drive motor 1 with ω_1 angular frequency, rotates the hydraulic pump 2, creates pressure and lubricant flow in the hydraulic system (HS). Lubricant through the pipeline 3 enters the engine 4, the shaft of which rotates with ω_2 angular frequency. Reducer 5 reduces the rotation frequency ω_2 to the ω_3 value. Pulleys 6 are mounted on the output shafts of the gearbox, which, together with pulleys 7 and ropes 8, will form the CPS, which creates the force of translational movement with the linear speed V of the pressure traverse of the head of the drilling projectile 9, together with the drill rod 10 and the drill bit 11.

Dynamic processes are studied with the assumption that the engine and mechanism have lumped parameters:

J_1 – the total moment of inertia of the rotor of the engine 1 and the rotating parts of the hydraulic pump 2;

J_2 – reduced combined total moment of inertia of the rotating parts of the HM 4, reducer 5, pulleys 6, 7;

J_3 – the combined total moment of inertia of the moving ropes 8, the pressure traverse of the drill head 9, the drilling rod 10 and the drilling bit 11.

The system is subject to torques:

M – motor torque;

M_C – active resistance moment caused by the weight of the drill bit.

The feed mechanism is affected by dissipative forces caused by friction in the shaft bearings, internal friction forces in the hydraulic system working fluid (HS) and in the rope arising during its movement.

We apply a decomposition approach to the electro hydromechanical system under study, which assumes that there is a general (global) model that describes the basic properties of the system adequately to the goal of decision-making (or research). If such a model is available, the use

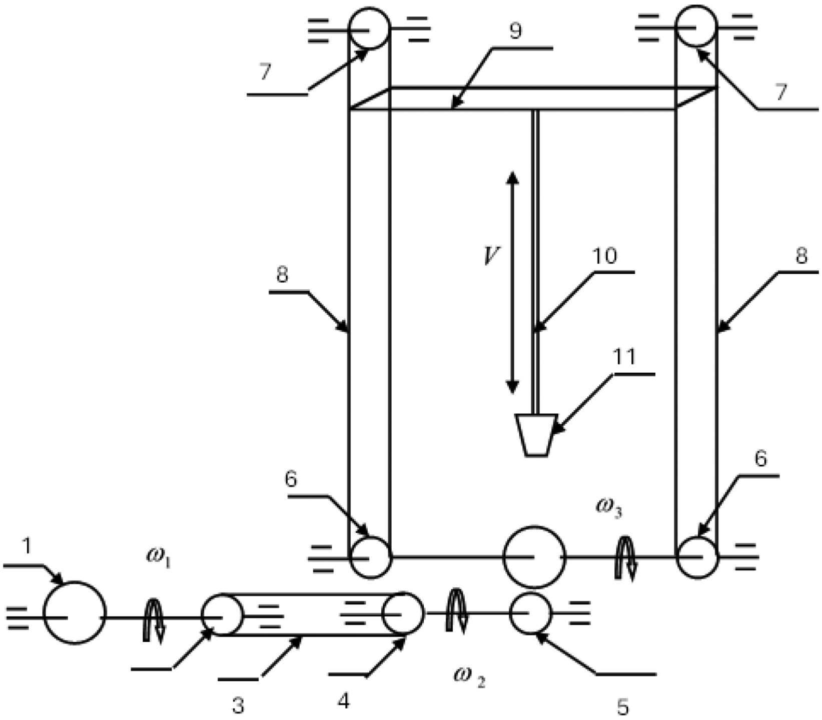

Fig. 1. Kinematic diagram of the LLO drive control object and supply supply: 1 – rotor of the drive motor; 2 – hydraulic pump; 3 – pipeline; 4 – HM; 5 – reducer; 6 – drive pulleys; 7 – guide pulleys; 8 – traction ropes; 9 – pressure traverse of the head of the drilling projectile; 10 – drilling rod; 11 – drilling bit

of the decomposition method can significantly reduce the dimensionality of the problem to be solved and reduce its solution to the sequential solution of intermediate (partial) problems of much smaller dimensionality. All models, constraints, and criteria of these tasks are directly derived only from a specially disaggregated global (general) model of the control object.

The basic architecture or model of the classical control theory in relation to drive systems is based on the description of the converter, motor, hydraulic system, and control system by a system of differential equations and the construction of a control system based on the accepted control concept. In particular, for the most critical drive systems, the methodology for building control systems with active sequential correction of the dynamic properties of the control object using loop tuning for modular or symmetric optimum is currently used as a technical standard.

Let's obtain a mathematical model in the form of a transfer function, according to which it is possible to synthesize a system of automatic control of the feed drive and the LLO of the drilling rod.

The operation of the hydraulic pump and the hydraulic pump is characterized by mutual influence due to the incompressible hydraulic fluid circulating in the pipeline, which is subject to elastic deformations. Therefore, in the mathematical description, we consider the operation of the hydraulic pump together with the hydraulic motor and the pipeline.

The input variables of the control object under investigation are the rotation frequency of the hydraulic pump shaft and its resistance moment. The output variables are the torque on the HM shaft (lubricant pressure) and the rotation frequency of its shaft (lubricant consumption).

The mathematical description is obtained under the following assumptions:

a) we do not take into account the mass of the lubricant circulating in the HS due to its insignificant value compared to other masses;

b) we ignore the change in the viscosity of the lubricant as a function of temperature, that is, we consider the processes to be isothermal;

c) leakage of lubricant from the HS is proportional to the pressure in the pipeline;

d) the change in the volume of lubricant in the pipeline is directly proportional to the volume of the supply pipeline and inversely proportional to the volume modulus of the pipeline material.

The tranfered function of the HS is based on two fundamental ratios: balances of lubricant consumption and torques on the shaft of the HM.

The oil displaced from the cylinders of the hydraulic pump rotates the HM, flows out of the HS and changes the volume of the pipeline (lubricant flow balance):

$$Q_S = Q_{HM} + Q_{LL} + Q_{PD}, \qquad (1)$$

where Q_S is the hydraulic pump supply;

Q_{HM} – consumption of lubricant through the HM;

Q_{LL} – consumption of lubricant for leaks from the HS;

Q_{PD} – lubricant consumption due to pipeline deformation.

The supply of the hydraulic pump and the flow rate of the hydraulic system are determined from the equations [5]:

$$Q_S = S_S \cdot R_S \cdot Z_S \cdot \omega_S = q_S \cdot \omega_S;$$
$$Q_{HM} = S_{HM} \cdot R_{HM} \cdot Z_{HM} \cdot \omega_{HM} = q_{HM} \cdot \omega_{HM},$$

where S_S, S_{HM}, are the cross sections of the cylinders;

R_S, R_{HM}, – the radii of the circles on which the centers of the cylinders are located;

Z_S, Z_{HM}, – number of pistons;

ω_S, ω_{HM}, – shaft rotation frequency;

q_S, q_{HM}, – total volume of cylinders, in liters (indices "s" – pump, "HM" – engine).

The consumption of lubricant leakage Q_{LL} from the HS is proportional to the pressure [6], i. e.

$$Q_{LL} = K_{LL} \cdot P,$$

where K_{LL} is the lubricant leakage coefficient, which depends on the state of the HS [7];

P – oil pressure.

Required lubricant consumption Q_{PD} for pipeline deformation [8]

$$Q_{PD} = dV/dt,$$

where dV – is the elementary change in the volume of the pipeline due to its deformation under pressure;

t – current time.

Under the accepted assumptions, the elementary amount of deformation of the pipeline [161]

$$dV = dP \cdot V/E,$$

where V – is the volume of hydraulic supply drives;

E – is the bulk modulus of the pipeline material [9].

According to Hooke's law, the difference in lubricant pressures at the inlet and outlet of the DH with the volume modulus of the pipeline are related by the ratio

$$dP = E/V \cdot dV.$$

Based on the power balance on the HM shaft, we find the required pressure in the supply HM:

$$P \cdot Q_{HM} = M_{HM} \cdot \omega_{HM};$$
$$P = M_{HM} \cdot \omega_{HM}/Q_{HM} = M_{HM}/q_{HM}.$$

Finally, the balance of lubricant consumption in the GS according to equation (1) takes the following form:

$$q_S \cdot \omega_S = q_{HM} \cdot \omega_{HM} + K_{LL} \cdot P + V/E \cdot dP/dt,$$

or considering the fact that the torque of the HM is proportional to the pressure in the supply pipeline and the total volume of the cylinders of the hydraulic pump ($M_{HM} = P \cdot q_{HM}$), we get

$$q_S \cdot \omega_S = q_{HM} \cdot \omega_{HM} + K_{LL} \cdot M_{HM}/q_{HM} + V/E/q_{HM} \cdot dM_{HM}/dt. \qquad (2)$$

The difference between the torque M_{HM} on the motor shaft and the value of the static moment of resistance M_C determines the dynamic torque of the |HM, which is balanced by the moment of inertia of the moving parts of the engine:

$$M_{HM} - M_C = J_3 \cdot d\omega_{HM}/dt, \qquad (3)$$

where J_3 is the moment of inertia of the rotating parts of the HM.

To obtain the classical transfer function, we move from the domain of the originals to the domain of images [6] with zero initial conditions. We apply the Laplace transform to equations (2), (3). Equation (2) is solved with respect to the torque of the HM, and equation (3) – with respect to the frequency of rotation of the HM shaft

$$\left.\begin{aligned} &M_{HM}(s) = (q_S \cdot \omega_S(s) - \\ &- q_{HM} \cdot \omega_{HM}(s)) \cdot (K_{LL}/q_{HM} + s \cdot V/E/q_{HM})^{-1} \\ &\omega_{HM}(s) = (M_{HM}(s) - M_C(s))/J_3/s, \end{aligned}\right\}$$

where $\omega_S(s)$, $\omega_{HP}(s)$ is the image of the frequency of rotation of the shafts of the HM;

$M_{HM}(s)$, $M_C(S)$ – the image of torque and static moments on the shaft of the HM.

For the last two equations in the field of images correspond to the structural diagram of the entire HS in the form convenient for the synthesis of the automatic control system of the feed drive and the LLO movement. This structural diagram is shown in Fig. 2.

Structurally, the HS is represented by two integrating links covered by two negative feedback loops: the internal one, which takes into account the lubricant consumption in the HS due to leaks, and the external one, which takes into account the lubricant consumption during rotation of the HM shaft. The resulting flow rate, which is the difference between the hydraulic pump supply and the total flow rate for the rotation of the hydraulic pump and leaks, determines the amount

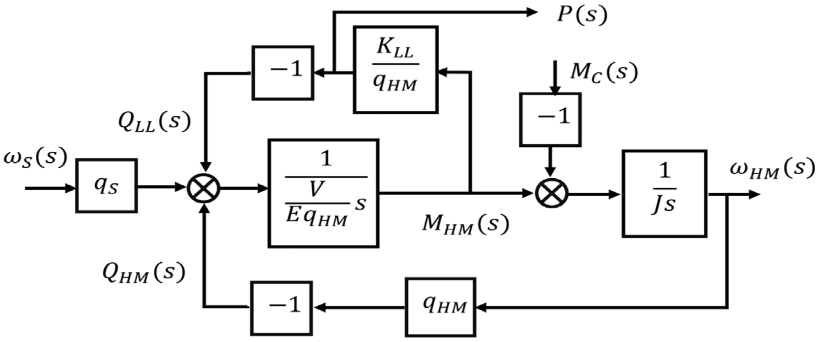

Fig. 2. Structural diagram of the HS of the feed drive and LLO

of deformation of the supply pipeline. The torque of the HM is proportional to the consumption and inversely proportional to the volume of the HM, the volumetric modulus of elasticity of the pipeline material and the DH capacity. The difference between the torque and static moments is applied to the mechanical parts of the engine, which leads to its acceleration if this difference is positive, and to braking in the opposite case.

In the structural diagram, there are two energy accumulators due to the presence of mechanical inertia of the rotating parts of the HV and the elastic properties of the HV pipeline. The presence of two inertias covered by common negative feedback leads to the ambiguity of transient processes in the HS.

The specific type of transition process depends on the structural parameters of the HS, and which can be estimated according to the obtained structural diagram. Transfer function of the entire system under control.

$$\frac{\omega_{HM}(s)}{\omega_s(s)} = \frac{q_S}{q_{HM}} \cdot \frac{1}{s^2 \cdot \dfrac{V \cdot J_3}{E \cdot q_{HM}^2} + s \cdot \dfrac{K_{LL} \cdot J_3}{q_{HM}^2} + 1} = \frac{q_S}{q_{HM}} \cdot \frac{1}{T_H^2 \cdot s^2 + 2 \cdot \xi_H \cdot T_H \cdot p + 1}, \quad (4)$$

where T_H, ξ_H – the time constant and the damping coefficient of the HS, moreover

$$T_H = \sqrt{V \cdot J_3 / E \cdot q_{HM}^2}; \quad \xi_H = 0{,}5 \cdot K_{LL}/q_{HM} \cdot \sqrt{E \cdot J_3 / V}.$$

The damping coefficient increases non-linearly with a decrease in the volume of the HS and an increase in the volume modulus of the pipeline material and the moment of inertia of the mechanical part of the HS. Damping is directly proportional to the leakage factor and inversely proportional to the internal volume of the HM.

The nature of transient processes depends, on the one hand, on the product of the volume of the HS by the square of the voluem, and on the other hand, on the product of the moment of inertia of the rotating parts of the HM by the modulus of volume elasticity of the pipeline material and the square of the leakage coefficient, which is determined from the characteristic equation (4).

If the inequality holds

$$4 \cdot V \cdot q_{HM} > E \cdot J_3 \cdot K_{LL}, \quad (5)$$

then the roots of the characteristic equation (4) are real, different and negative, the transition process is the sum of two exponents (aperiodic process). In the opposite case, when the inequality (5) is not fulfilled, the roots of the characteristic equation (4) will be complex-coupled and the curve of the transition process is a sinusoid, which is modulated by the amplitude of the exponent (oscillatory convergent transition process). In a separate case, when inequality (5) degenerates into equality, the roots of the characteristic equation will be multiple, and the transition process will be boudary aperiodic

Results. Thus, the obtained mathematical model reflects all the essential properties of the HS and allows determining its dynamic and static modes of operation. The static transfer function is found from the dynamic (4) by substituting the value $s = 0$. The structural diagram makes it possible to evaluate the impact on the dynamic properties of the design parameters of the HS, and, if necessary, to reduce the fluctuation of the HS in the feed drive, because otherwise, for ensure the stability of the entire control system, it is necessary to reduce its speed, which will negatively affect the quality of regulation. As can be seen from the found relation (5), excessive fluctuation of the HS can be eliminated by using a pipeline with an increased modulus of bulk elasticity, or by reducing the volume of the HS by reducing the length of the pipelines between the hydraulic pump and the hydraulic motor.

Conclusions. The following conclusions can be made as a result of research conducted on the electro-hydromechanical transmission of the feed drive and LLO as an object of automatic control:

1. The mathematical model of the electro-hydromechanical system of the LLO drive and power supply was developed, which allows for a quantitative assessment of the influence of the elastic properties of the HS and CPS on the dynamic processes in the transmission.

2. The obtained and analyzed mathematical model of the control object in the form of transfer functions, which allows synthesizing the system of automatic control of the LLO drive and the supply of drilling rod.

3. From the analysis of the obtained mathematical model of the object of control of the feed drive and SPOP, it turns out that in the object of control, in order to implement a rational method of control with the support of a constant flow of power in the breakdown zone, it is necessary to control, in addition to the traditional coordinates - current and engine speed, also and technological coordinates – the pressure in the hydraulic system, the frequency of rotation of the GM shaft and the linear speed of the position movement, which requires the appropriate switching on of the control circuits.

4. The presence of elastic connections in the transmission leads to the appearance of additional fractional-rational function, the zeros and poles of which move from the low- to high-frequency region when the number of rods in the posture decreases.

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