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DEVELOPMENT OF THE UNIVERSAL MODEL OF MECHATRONIC SYSTEM WITH A HYDRAULIC DRIVE

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Зростаючі вимоги до продуктивності мехатронних систем з гідравлічним приводом активних робочих органів самохідних машин вимагають застосування нових підходів в процесі розробки та проектування. Функціональні параметри мехатронних систем залежать від раціонального вибору режимів роботи гідравлічної системи та конструктивного виконання мехатронних модулів цих систем. Якість мехатронної системи з гідравлічним приводом в більшій мірі визначається динамічними характеристиками. Для поліпшення динамічних характеристик запропонована універсальна модель, яка описує динамічні і статичні процеси, що відбуваються в елементах мехатронної системи. Насос, гідромотор, запобіжний клапан та робоча рідина розглянуті у взаємозв'язку, як єдине ціле. Універсальна модель враховує особливості функціонування і взаємний вплив всіх елементів мехатронної системи, а також особливості робочої рідини та може бути використана з будь-якими гідромашинами об'ємної дії. Дослідження динаміки зміни функціональних параметрів мехатронної системи з гідравлічним приводом здійснювалося на чотирьох етапах роботи: розгін гідроприводу (спрацювання запобіжного клапана); закриття клапана; завершення розгону; сталий режим роботи. Проведеними дослідженнями встановлено, що при пуску гідроприводу мехатронної системи з моменту спрацьовування запобіжного клапана і до його закриття умови експлуатації не впливають на зміну функціональних параметрів. При сталому режимі роботи спостерігаються пульсації, викликані нерівномірністю подачі насоса і коливаннями навантаження. Також необхідно відзначити, що мехатронна система з гідромотором, який має більший робочий об'єм, має кращі динамічні характеристики, ніж система з гідромотором меншого об'єму

Ключові слова: мехатронна система з гідравлічним приводом, універсальна модель, функціональні параметри, динамічні характеристики

1. Introduction

Current tendencies of self-propelled machinery hydrofication require the development of new and improvement of existing designs of mechatronic systems with a hydraulic drive for their movable operating elements. Functional parameters of the mechatronic systems with hydraulic drives depend on the rational choice of operating modes of the hydraulic systems and embodiment of their mechatronic modules. Quality of mechatronic systems is largely determined by their dynamic characteristics associated with such indicators as the time of regulation of transient processes, frequency and amplitude of fluctuations, etc. To study dynamic characteristics of mechatronic systems, it is necessary to develop mathematical models of the working processes occurring in the hydraulic drives of these systems.

The physical processes occurring in mechatronic systems with a hydraulic drive of movable operating elements are associated with movement of the working fluid through the pipelines and channels of hydraulic devices [1–5]. In addition to the main flows of the working fluid, which are necessary for the system functioning, there are additional flows in clearances between the parts of mechanisms and components of the hydraulic devices. When working out a mathematical model of the work processes occurring in a mechatronic system of the hydraulic type, various hydromechanical and dynamic phenomena must be considered. These phenomena characterize flows of the working fluid which are accompanied by appearance of pressure and flow rate fluctuations caused by compressibility of the working fluid, effect of fluid flows on the mechatronic system elements, etc. [5].

Mathematical description of hydromechanical processes is based on general equations of continuum motion known from the liquid and gas mechanics. Experimental values of the coefficients of hydraulic resistance, flow rates and hydrodynamic forces are used [5]. The use of general equations and dependencies of hydromechanics in the study of dynamic characteristics of mechatronic systems is determined by the principle of action, design and operating modes of these systems. The mechatronic systems feature dynamic processes in which motion of working fluids is non-stationary, that is, pressure, velocity and density of the working fluid depend on time in any point of the flow cross-section. In most cases, quasi-stationary values of the coefficients of amount of movement, kinetic energy, hydraulic resistance, and flow state are assumed in mathematical description of non-stationary flows of the working fluid.

Thus, the study of dynamics of the mechatronic system for actuators of movable operating elements of self-propelled machinery on the basis of a universal mathematical model including elements of the hydraulic drive and working fluid is a topical issue.

2. Literature review and problem statement

The growing requirements to performance of complex mechatronic systems used in self-propelled machines of an ever-increasing technological significance require application of new design methods in the process of their development. A mechatronic system with good dynamic characteristics ensures the possibility of estimation and optimization of dynamic characteristics of the entire system at the early design stages. An analysis of published studies has shown [6] that it is not enough to concentrate solely on optimization of subsystems. It is necessary to optimize the mechatronic system in general. Papers [6, 7] have considered the methods for optimal design of complex mechatronic systems. However, nothing was said about construction of physical and mathematical models of the processes occurring in the mechatronic systems.

Analysis of the evolutionary trend of the models used in the design and development of mechatronic products (robots) indicates the necessity of integration of the design characteristics and requirements in various fields of technology [8]. An integrated model for the design and development of mechatronic products in a context of flexible production systems was proposed but no consideration was given to the drive types.

Much attention is paid currently to development of dynamic models of manipulators [9–12]. Oscillation control algorithms were developed on the basis of dynamic models of mechatronic systems [9]. Numerical simulation has been carried out to study the effect of joint elasticity and perturbations as well as the parameters of structure and motion on variation of the oscillation power flow [10]. Dynamic effect of the drive motor on operation of manipulators was studied [11]. The problem of modeling dynamic mechatronic systems including industrial robots was presented. Analysis stages and basic mathematical interrelations were shown [12]. However, the issues related to the study of dynamic properties of the mechatronic systems used for actuators of movable operating elements of self-propelled machinery remain open so far.

The developed mathematical model of a semiautomatic block with an amplifier based on a hydraulic accumulator [13] has enabled study of its static and dynamic characteristics. Mathematical models of the work processes occurring in pumps [14] and hydraulic motors [15] of the drive of movable operating elements of self-propelled machinery were offered but their interrelations and mutual influence have not been considered.

Mathematical models reflecting influence of the working fluid temperature on the moment of friction in gear pumps with internal gearing were proposed in [16]. Influence of pump operating modes on the leakage flow characteristics was investigated and a mathematical model for calculating leakage flow was proposed [17]. However, features of the working fluid were not considered.

The developed mathematical models of the work processes occurring in the safety valves of direct [18] and indirect [19] action have made it possible to study static and dynamic properties of safety valves. However, nothing was said on the use of these valves in hydraulic drives of mechatronic systems.

A design diagram and mathematical model of a hydrostatic transmission [20] including a pump, a valve and a hydraulic motor made as a single hydraulic block were considered. The working fluid was not considered as an element of the hydraulic system. The developed model was not intended to study output characteristics of each element of the hydraulic unit during operation of the hydrostatic transmission.

Analysis of published data has shown that the studies were conducted specifically for some element of the mechatronic system. Development of the mathematical models did not take into account features of functioning and mutual influence of all elements of the mechatronic system as well as the working fluid features. Also, hydraulic elements and working fluid were not considered as a single whole. Mutual influence of all hydraulic elements and working fluid on dynamic characteristics of the mechatronic system of the drive of movable operating elements of self-propelled machinery was not studied.

Thus, in order to improve dynamic characteristics of mechatronic systems with a hydraulic drive for movable operating elements, it is necessary to develop a universal model. This model should take into account features of functioning and mutual influence of all hydraulic elements and the working fluid.

3. The aim and objectives of the study

The study objective was to improve dynamic characteristics of the mechanically driven mechatronic system by studying dynamics of change of functional parameters based on the developed universal model. The proposed model should take into account features of operation and mutual influence of all elements of the system as well as the working fluid features.

To achieve the objective, the following tasks had to be solved:

– to develop a universal model which would make it possible to study dynamics of change in functional parameters of a mechanically driven mechatronic system;

– to study dynamics of change in functional parameters of a mechanically driven mechatronic system without taking into account the influence of operating conditions;

– to study dynamics of change in functional parameters of a mechanically driven mechatronic system in operating conditions.

4. Materials and methods used in the study of a mechanically driven mechatronic system

The following assumptions are usually made when constructing mathematical models [4, 5]:

– no working fluid leakage;

– zero dry friction;

– no wave processes in pipelines;

– constant temperature of the working fluid (that is, constant coefficients of kinematic viscosity and frictional forces);

– neglect of frictional losses;

– zero pressures in the exhaust, suction and drain lines;

– sharp working edges;

– large cross section and short length of the connected channels;

– constant elastic modulus of the working fluid.

To study dynamics of change in functional parameters of a mechanically driven mechatronic system of movable operating elements of self-propelled machinery, mathematical models of the work processes occurring in the pump, the hydraulic motor and the safety valve were considered. These models were constructed based on the equations of continuity, advance of movable parts of the mechatronic system elements and equations of flow in hydraulic devices [3, 4].

Fig. 1 shows the design diagram of the mechanically driven mechatronic system [4] representing a set of connected hydraulic devices, that is, the pump, the hydraulic motor and the safety valve. All elements of the mechatronic system are interconnected by the forms of interaction and interdependence with the help of the working fluid and form a single whole.

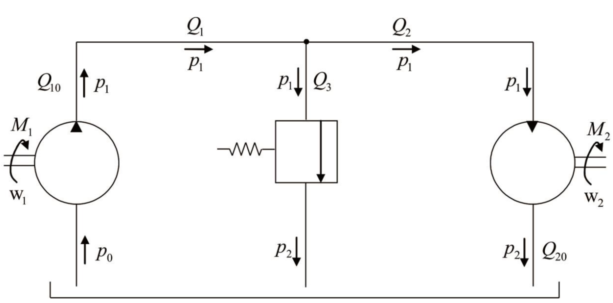

Fig. 1. Design diagram of the model of the mechanically driven mechatronic system: Q_1 is the pump feed; Q_2 is the flow rate of the working fluid fed to the hydraulic motor; Q_3 is the flow rate through the safety valve; Q_{10} is the geometric pump feed; Q_{20} is the geometric flow rate of the hydraulic motor; p_0 is pressure in the suction line, at the pump inlet; p_1 is pressure of the working fluid in the injection line of the mechatronic system; p_2 is pressure in the drain line of the mechatronic system; M_1 is torque on the pump shaft; M_2 is torque on the hydraulic motor shaft; ω_1 is angular velocity of the pump shaft; ω_2 is angular velocity of the hydraulic motor shaft

The proposed universal model of the work processes occurring in the mechanically driven mechatronic system features consideration of all its elements and the working fluid as a single whole. It is considered as the model enabling the study of dynamic characteristics of the mechanically driven mechatronic system and the use of various hydraulic machines of volumetric action. A mathematical description of the work processes occurring in the pump, the safety valve, the hydraulic motor and the working fluid based on the equations of flow rates, motion of elements and continuity is given in more detail in [4, 14, 17, 18].

The universal model of the work processes occurring in the mechanically driven mechatronic system including the pump, the hydraulic motor and the safety valve can be represented by a system of equations:

$$Q_2(t)=Q_1(t)-Q_3(t), \tag{1}$$

$$\begin{gathered}Q_{20}(t)+Q_{21}(t)+Q_{22}(t)+Q_{23}(t)+Q_{24}(t)=\\=Q_{10}(t)-Q_{11}(t)-Q_{12}(t)-Q_{13}(t)-Q_{14}(t)-Q_3(t),\end{gathered} \tag{2}$$

$$Q_3(t)=\mu\cdot\pi\cdot d\cdot E(t)\cdot\sqrt{\frac{2}{\rho}\left[p_1(t)-p_2\right]}, \tag{3}$$

$$\begin{gathered}\frac{V_2}{2\pi}\cdot\omega_2(t)+C_{21}\cdot p_1(t)+C_{22}\cdot\left[p_1(t)-p_2(t)\right]+\\+C_2\cdot\frac{\omega_2(t)}{E}\cdot\left[p_1(t)-p_2\right]+\frac{(V_2+V_{20})}{2\cdot E}\frac{dp_1(t)}{dt}=\\=\frac{V_1}{2\pi}\cdot\omega_1(t)\cdot e-C_{11}\cdot\left[p_1(t)-p_0'\right]-C_{12}\cdot\left[p_1(t)-p_0\right]-\\-C_1\cdot\frac{\omega_1(t)}{E}\cdot\left[p_1(t)-p_2\right]-\frac{(V_1+V_{10})}{2\cdot E}\frac{dp_1(t)}{dt}-Q_3(t),\end{gathered} \tag{4}$$

$$\begin{gathered}\frac{dp_1(t)}{dt}=\frac{E}{(V_{11}+V_{21}+V_3)}\cdot\left\{\frac{V_1}{2\pi}\cdot\omega_1(t)\cdot e-\frac{V_2}{2\pi}\cdot\omega_2(t)-\right.\\-C_{11}\cdot\left[p_1(t)-p_0'\right]-C_{12}\cdot\left[p_1(t)-p_0\right]-C_{21}\cdot p_1(t)-\\-C_{22}\cdot\left[p_1(t)-p_2\right]-\frac{1}{E}\cdot\left[C_1\cdot\omega_1(t)+C_2\cdot\omega_2(t)\right]\cdot\left[p_1(t)-p_2\right]-\\\left.-\mu\cdot\pi\cdot d\cdot x(t)\cdot\sqrt{\frac{2}{\rho}\left[p_1(t)-p_2\right]}-S\frac{dx}{dt}\right\},\end{gathered} \tag{5}$$

$$M_2=M_j+M_c, \tag{6}$$

$$M_2=\frac{V_2}{2\pi}\cdot\eta_m\cdot\left[p_1(t)-p_2\right],\quad M_j=J\cdot\frac{d\omega_2(t)}{dt}, \tag{7}$$

$$\frac{d\omega_2(t)}{dt}=\frac{1}{J}\cdot\left[\frac{V_2}{2\pi}\cdot\eta_m\cdot\left[p_1(t)-p_2\right]-M_c\right], \tag{8}$$

$$\begin{gathered}Q_{25}(t)=Q_1(t)-C_{21}\cdot p_1(t)-C_{22}\cdot\left[p_1(t)-p_2(t)\right]-\\-C_2\cdot\frac{\omega_2(t)}{E}\cdot\left[p_1(t)-p_2\right]-\frac{V_2}{2\cdot E}\frac{dp_1(t)}{dt},\end{gathered} \tag{9}$$

$$\begin{gathered}E=K(p_1+1)\cdot(Ap_1+B)\times\\\times\frac{(1-m_0)\cdot D_1+m_0\cdot D_2}{K(p_1+1)\cdot(1-m_0)\cdot D_1+m_0(Ap_1+B)\cdot D_2},\end{gathered} \tag{10}$$

where Q_1 is the amount of the working fluid coming from the pump to the hydraulic motor; Q_2 is the amount of the hydraulic fluid fed to the hydraulic motor; Q_3 is the rate of flow through the safety valve; Q_{10} is the geometric pump feed; Q_{11} is the leakage rate (into the drainage line) of the pump; Q_{12} is the flow-over rate (into the suction line) of the pump; Q_{13} is the pump flow rate caused by compression of the working fluid; Q_{14} is the deformation flow rate of the pump;Q_{20} is the geometric flow rate of the hydraulic motor;Q_{21} is the flow rate of hydraulic fluid leakages of the hydraulic motor (into the body);Q_{22} is flow-overs of the working fluid in the hydraulic motor;Q_{23} is the hydraulic motor flow caused by compression of the working fluid; Q_{24} is the deformation flow rate of the hydraulic motor; Q_{25} is the amount of the hydraulic fluid to the drain line from the hydraulic motor; C_1 is the coefficient of proportionality in the pump; C_{11}, C_{12} are the coefficients of leakages and flow-overs in the pump; C_2 is the coefficient of proportionality in the hydraulic motor; C_{21}, C_{22} are the coefficients of leakages and flow-overs in the hydraulic motor; V_1 is the working volume of the pump; V_{10} is the inactive volume of the pump; V_{11} is the volume of the working fluid in the injection cavity of the pump; V_2 is the working volume of the hydraulic motor; V_{20} is the inactive volume of the hydraulic motor; V_{21} is the volume of the working fluid in the injection cavity of the hydraulic motor; p_0 is the pressure in the suction line, at the pump inlet; p_1 is the pressure of the working fluid in the injection line of the mechatronic system (numerically equal to the pressure at the pump outlet and the pressure at the hydraulic motor inlet); p_2 is the pressure in the drain line of the mechatronic system; p_0' is the pressure in the drain line of the mechatronic system; J is the moment of inertia of the rotating masses; M_c is the moment of resistance; M_j is the moment of inertia; M_1 is the torque on the pump shaft; M_2 is the torque on the hydraulic motor shaft; ω_1 is the angular velocity of the pump shaft; ω_2 is the angular velocity of the hydraulic motor shaft; η_m is mechanical efficiency of the hydraulic motor; ρ is the density of the working fluid; μ is the flow coefficient; μ' is the coefficient of dynamic viscosity of the liquid; E is the modulus of elasticity of volume of a two-phase working fluid; d is the ram diameter; $x(t)$ is the ram displacement from its closed position; S is the effective area of the ram; K is the polytropic exponent; A and B are the parameters of the working fluid depending on the type of oil and operating temperature of the system; m_0 is the content of undissolved air in the working fluid, relative units.

The proposed universal model of the working processes occurring in the hydraulically driven mechatronic system describes dynamic and static processes occurring in its elements. The pump, the hydraulic motor, the safety valve and the working fluid are considered interconnected in a single whole. Moreover, this model takes into account operation features and mutual influence of all elements of the mechatronic system as well as the working fluid features and can be applied for any hydraulic device of volumetric action.

5. Analysis of the results obtained in simulation of operation of a mechanically driven mechatronic system

The mathematical model of the work processes occurring in the mechatronic system of hydraulic type was studied on a PC using the VisSim visual simulation package. This package makes it possible to integrate a system of nonlinear differential equations of high order at any given time under various conditions of operation of the mechanically driven mechatronic system. Therefore, it was used to determine the change in functional parameters of the mechatronic system taking into account interaction of the pump, the motor, the safety valve and the working fluid.

Let us assume the following initial conditions to simulate dynamics of changes in functional parameters of the mechanically driven mechatronic system of movable operating elements of self-propelled machines [21]:

– *the pump*: fixed displacement gear type with constant geometric feed Q_{10}=100 l/min; angular shaft velocity ω_1=125 s^{-1}; control parameter e=1; coefficient of proportionality C_1=8.5; coefficients of leakages and flow-overs C_{11}=1.5 and C_{12}=3.5, respectively;

– *the hydraulic motor*: planetary, PRG-22 model, working volume V_2=160, 200, 250, 320, 400, 500, 630 cm^3; constant moment of resistance M_c=360, 450, 560, 715, 900, 1,120, 1,430 N·m; inertia moment of rotating masses J=3.6; volumetric efficiency η_v=0.95, mechanical efficiency η_m=0.9; pressure at the outlet p_2=0; proportionality factor C_2=1; coefficients of leakages and flow-overs C_{21}=1.5 and C_{22}=12.99, respectively;

– *the valve*: safety type, spring stiffness C=200 kg/cm; preliminary spring compression x_0=0.125 cm; positive slot overlap x_z=0.53 cm;

– *the working fluid*: polytropic exponent K=1.2; parameters depending on the oil type and operating temperature of the hydraulic system A=12.62 and B=1740; content of undissolved air m_0=0.925 rel. un.

The initial parameters of the pump, the motor, the valve and the working fluid are set by *block* A (Fig. 2).

The change in amount of the working fluid fed to the hydraulic motor and passing through it which, as determined by expressions (1) and (9), is represented by *block* B (Fig. 3).

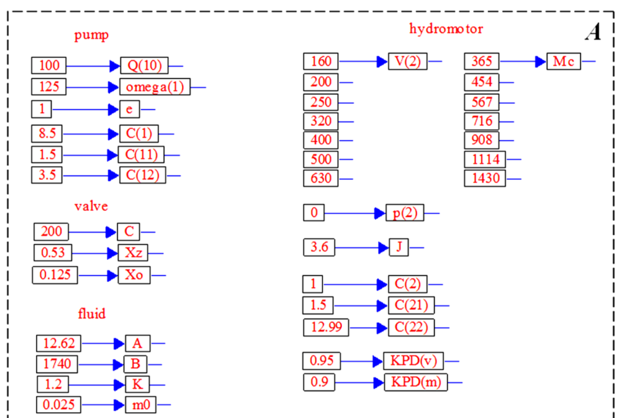

Fig. 2. Block of initial data of the hydraulic motor pump, the valve and the working fluid

Joint solution of the pump, the hydraulic motor and the safety valve flow equations (4) makes it possible to obtain a pressure change in the injection line of the mechatronic system determined from expression (5) and described by *block* C (Fig. 4).

The change in torque and inertia moments described by expressions (7) is represented by *block* D (Fig. 5).

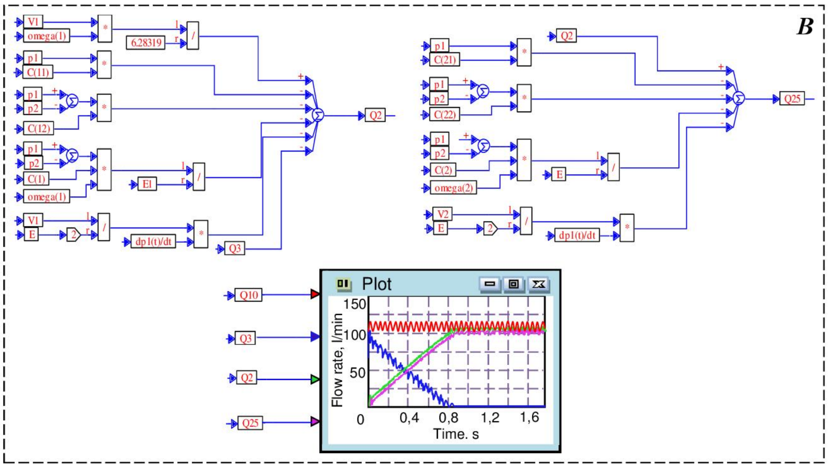

Fig. 3. Block of simulation of the change in the working fluid amount supplied to the hydraulic motor and passing through it

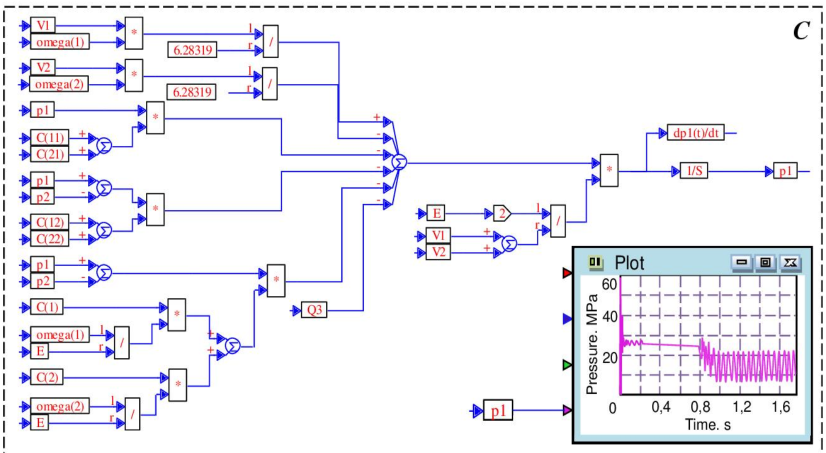

Fig. 4. Block of simulation of pressure change in the injection line of the mechatronic system

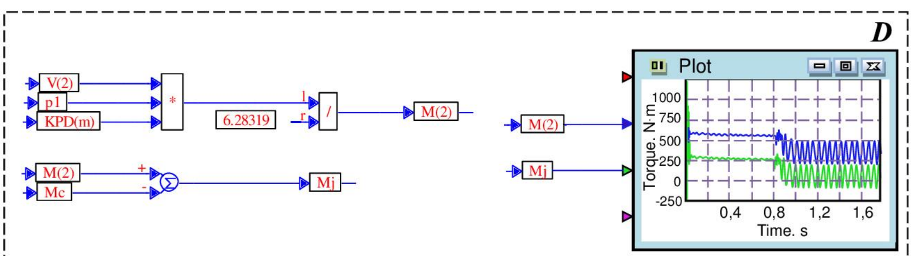

Fig. 5. Block of simulation of the change in torque and inertia moments

The change in the angular velocity (rotation frequency) of the hydraulic motor shaft determined from the equation of moments (8) is described by *the block* F (Fig. 6).

Operation of the mechanically driven mechatronic system was simulated on the example of two planetary hydraulic motors with working volumes V_2=160 and 630 cm^3, respectively. This simulation was made without taking into account the influence of operating conditions at constant pump feeds Q_{10}= =100 l/min (for both motors) and constant resistance moments M_c=365 and 1,430 N·m, respectively.

The results of simulation of dynamics of the changes in functional parameters of the mechatronic system without consideration of operating conditions are represented by corresponding dependences (Fig. 7–11). The obtained curves determine characteristics of the mechatronic system as a whole: dynamics of change in pressure in injection line of the mechatronic system, p_1 (Fig. 7), torque on the motor shaft, M_2 (Fig. 8), the motor shaft rotation frequency, n_2 (Fig. 9), and rates of the working fluid flow through the safety valve Q_3 (Fig. 10) and the hydraulic motor Q_2 (Fig. 11).

At actuation of the hydraulic drive of the mechatronic system (t=0...0.02 s), pressure in the injection line sharply increases (Fig. 7) with its peak p_1=84 MPa for a hydraulic motor with working volume V_2=160 cm^3 (curve 1) and p_1=63 MPa for a hydraulic motor with working volume V_2=630 cm^3 (curve 2). The pressure peaks exceed the nominal value (pnom=16 MPa) 5.2 times for a 160 cm^3 hydraulic motor and 3.9 times for a 630 cm^3 pump which is explained by the safety valve actuation lag.

With further acceleration (t=0.02... 0.78 *s*, the safety valve open), pressure in the injection line of the mechatronic system with a 160 cm^3 hydraulic motor practically does not change remaining at about 27 MPa (Fig. 7, curve 1). At the end of acceleration (t= =0.78...0.9 *s*), pressure gradually decreases and reaches its nominal value of 16 MPa (t>0.9 *s*, the safety valve closed). The mechatronic system with a 630 cm^3 hydraulic motor goes into a steady-state operation mode much earlier (Fig. 7, curve 2), over time period t>0.3 s.

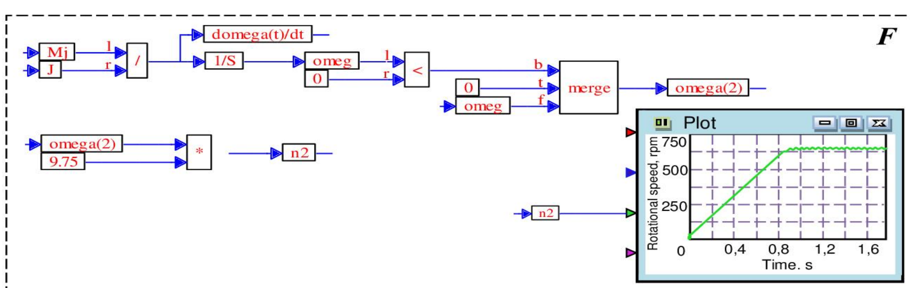

Fig. 6. Block of simulation of the change in angular speed (rotation frequency) of the motor shaft

The pattern of change in torque, M_2, on the shafts of hydraulic motors with working volumes of 160 cm^3 (curve 1) and 630 cm^3 (curve 2) completely repeats the pattern of pressure change in the injection line at all stages of operation (Fig. 8). In the steady-state operation of the mechatronic system, the torque values are 365 N·m and 1,430 N·m, respectively.

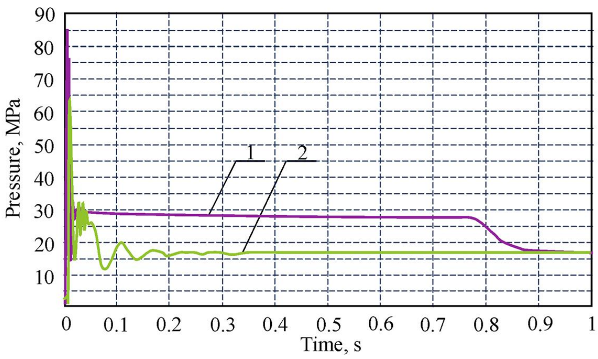

Fig. 7. Dynamics of pressure change in the mechatronic system with a 160 cm^3 hydraulic motor (1) and a 630 cm^3 hydraulic motor (2)

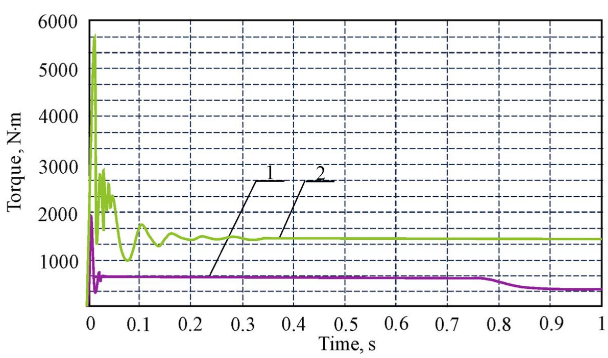

Fig. 8. Dynamics of the torque change on the shaft of the hydraulic motor operating in the mechatronic system: 160 cm^3 working volume (1); 630 cm^3 working volume (2)

Dynamics of change in rotation frequency of the 160 cm^3 hydraulic motor shaft (Fig. 9) shows that when the hydraulic drive accelerates (t=0...0.78 s) with the safety valve open, rotation frequency changes linearly (curve 1). In the period of acceleration completion (t=0.78...0.9 s), rotation frequency smoothly reaches its nominal value of 600 min^{-1} and a steady-state operation takes place at t>0.9 s. The mechatronic system with a 630 cm^3 hydraulic motor (curve 2) enters the steady-state operation mode in the period t=0...0.2 s and the value of rotation frequency reaches its nominal value of 150 min^{-1} at t>0.2 s.

An analysis of dynamics of the safety valve operation change (Fig. 10) shows that the safety valve in a mechatronic system with a 160 cm^3 hydraulic motor triggers at the initial moment and completely opens at t=0...0.02 s (curve 1). The rate of the working fluid flow through the valve reaches its maximum value Q_3=100 l/min. Further, the valve is smoothly closed during the period t=0.02...0.78 s and completely closed at t>0.78 s. When the mechatronic system with a 630 cm^3 hydraulic motor is operating, the safety valve is also actuated at the initial moment of time and is completely open at t=0...0.02 s (curve 2). At this moment of time, the working liquid flow through the valve is maximal: Q_3=100 l/min. Then the valve starts pulsating and closes in the period t=0.02...0.05 s and a steady-state operation of the mechatronic system takes place.

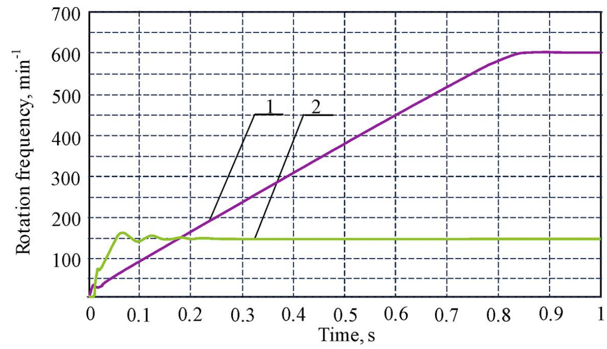

Fig. 9. Dynamics of change in hydraulic motor rotation frequency: 160 cm^3 working volume (1); 630 cm^3 working volume (2)

Dynamics of the change in the working fluid flow rate (Fig. 11) shows that filling of the 160 cm^3 hydraulic motor with a working fluid occurs at t=0...0.78 (curve 1) due to the opening of the safety valve. The flow rate smoothly reaches its nominal value and makes Q_2=98 l/min after closure of the safety valve (t=0.78...0.9 s). Filling of the 630 cm^3 hydraulic motor occurs at t=0...0.05 s (curve 2) due to activation of the safety valve at the moment of acceleration of the hydraulic drive. Further, the flow rate smoothly reaches its nominal value Q_2=98 l/min at t=0.05...0.2 and the steady-state operation of the mechatronic system is observed at t>0.2 s.

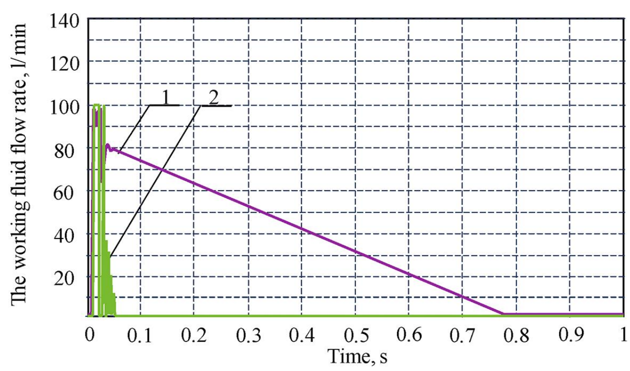

Fig. 10. Dynamics of the change in the rate of flow through the safety valve operating in the mechatronic system with the hydraulic motor working volumes of 160 cm^3 (1) and 630 cm^3 (2)

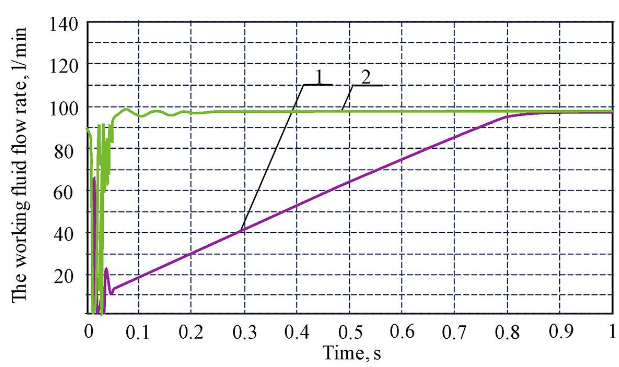

Fig. 11. Dynamics of the change of flow rate of the hydraulic motor operating in the mechatronic system: 160 cm^3 working volume (1); 630 cm^3 working volume (2)

Previously, ideal operation conditions for the mechanically driven mechatronic system were taken, that is, constant pump feed and load. However, the pump feed and load are uneven during operation of the mechatronic system. Unevenness of the pump feed described by the *block G* (Fig. 12) was simulated by summing up the half-sinusoids with a time shift [5], i. e.

$$Q_1(t) = Q_\omega \cdot \sin \omega t + Q_\omega \cdot \sin \omega (t - \tau), \tag{11}$$

where τ is the time shift, τ=0.05 s, and the load change described by *block H* (Fig. 13) was simulated through the moment of the resistance, Mc, which varies according to exponential and sinusoidal laws with a lag of t=0.05 s:

$$M_c = M_{c0} \cdot \left(1 - e^{-\frac{t}{T}}\right) + M_\omega \cdot \sin \omega t. \tag{12}$$

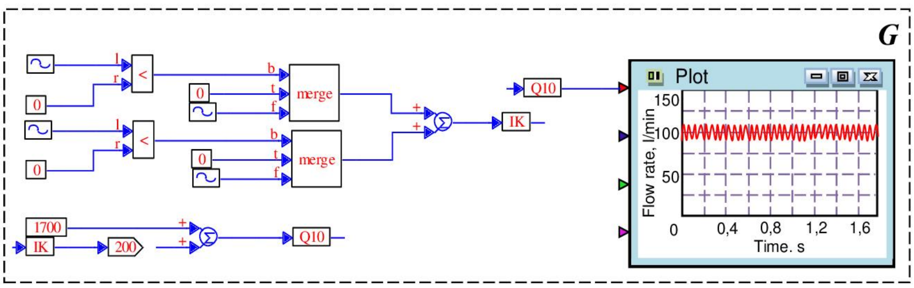

Fig. 12. Block of simulation of the pump feed unevenness

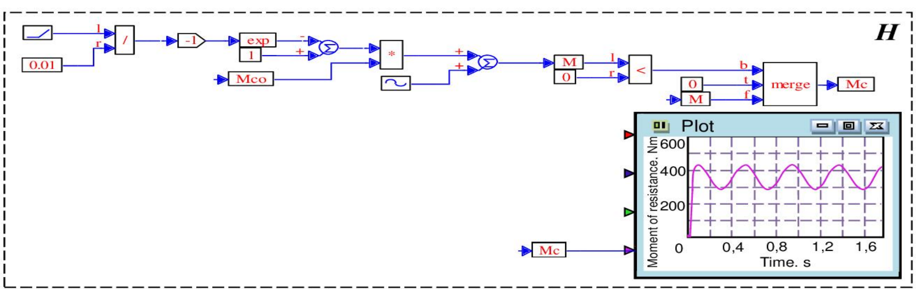

Fig. 13. Block of load change simulation

Dynamics of the change in functional parameters of the mechanically driven mechatronic system taking into account operating conditions (simultaneous simulation of the change in pump feed (Fig. 12) and load (Fig. 13)) is presented in Figs. 14–18.

When the hydraulic drive of the mechatronic system was actuated (t=0...0.02 s), an "overshoot" of pressure p1 in the injection line of the mechatronic system was observed because of the lag of the safety valve actuation (Fig. 14). The peak pressure of 88 MPa (curve 1) for the mechatronic system with a 160 cm^3 planetary hydraulic motor and 57 MPa with a 630 cm^3 hydraulic motor (curve 2) was observed. That is, the pressure peak increased by 4.5 % with a 150 cm^3 hydraulic motor and decreased by 9.5 % with a 630 cm^3 hydraulic motor compared to the ideal simulation conditions (Fig. 7).

With further acceleration (t=0.02...0.78 s) with open safety valve, pressure in the injection line of the mechatronic system with a 160 cm^3 hydraulic motor practically did not change remaining at about 27 MPa with minor variations (Fig. 14, curve 1). After the valve closure (t=0.78 s), pressure started to be significantly fluctuating (the amplitude up to 15 MPa) with the fluctuation frequency equal to the set fluctuation of the pump feed Q_{10} (Fig. 12). At the same time, the mean pressure value changed according to the sinusoidal law brought about by fluctuation of the moment of resistance, Mc (Fig. 13). Pressure in the injection line of the mechatronic system with a 630 cm^3 hydraulic motor was significantly fluctuating (Fig. 14, curve 2) after the valve closure (t=0.11 s) with fluctuation amplitude up to 15 MPa like in the system with a 150 cm^3 hydraulic motor.

The curves of torque M^2 change (Fig. 15) on the shafts of hydraulic motors with working volumes of 160 cm^3 (curve 1) and 630 cm^3 (curve 2) fully repeated the pattern of pressure change in the injection line at all operating stages. Moreover, the steady-state torque (t>0.78 s) for the mechatronic system with a 160 cm^3 hydraulic motor was characterized by a sinusoidal curve fluctuating with an amplitude of about 300 N·m (curve 1). For a 630 cm^3 hydraulic motor, the steady-state torque (t>0.11 s) on the shaft was characterized by a sinusoidal curve fluctuating with an amplitude of about 1000 N·m (curve 2). In this case, the average torque values on the shafts of hydraulic motors were about 370 N·m and 1450 N·m, respectively. The torque fluctuation frequency was determined by the set value of fluctuation of the pump feed Q_{10} (Fig. 12) and the sinusoidal fluctuations were determined by fluctuation of the resistance moment, M_c (Fig. 13).

Dynamics of the change in rotation frequency, n_2, on the shafts of the hydraulic motors (Fig. 16) with working volumes of 160 cm^3 (curve 1) and 630 cm^3 (curve 2) repeated the pattern of change in the rotation frequency, n^2, without taking into account the operating conditions (Fig. 9). However, the steady-state operation was characterized by fluctuation of the rotation frequency of the hydraulic motor shaft with an amplitude up to 30 min^{-1} for the 160 cm^3 hydraulic motor and up to 50 min^{-1} for the 630 cm^3 hydraulic motor.

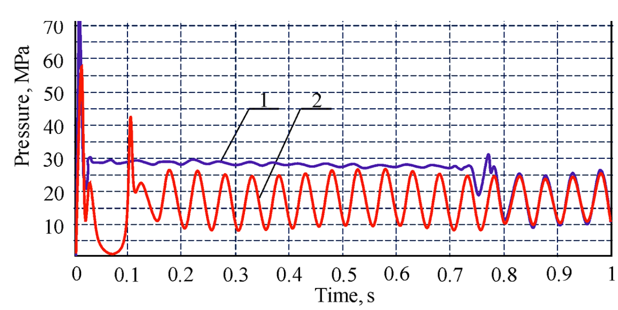

Fig. 14. Dynamics of pressure change in the mechatronic system with a hydraulic motor working volume of 160 cm^3 (1) and 630 cm^3 (2)

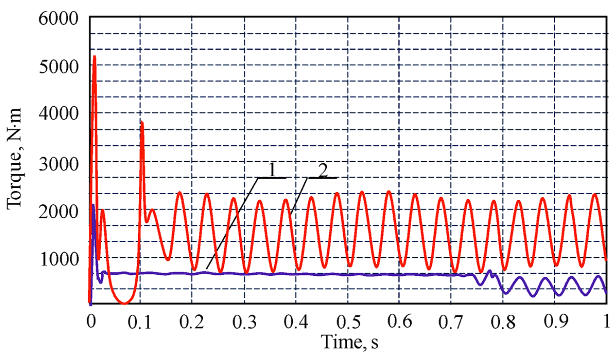

Fig. 15. Dynamics of the change in torque on the shafts of the hydraulic motors operating in the mechatronic system with working volumes of 160 cm^3 (1) and 630 cm^3 (2)

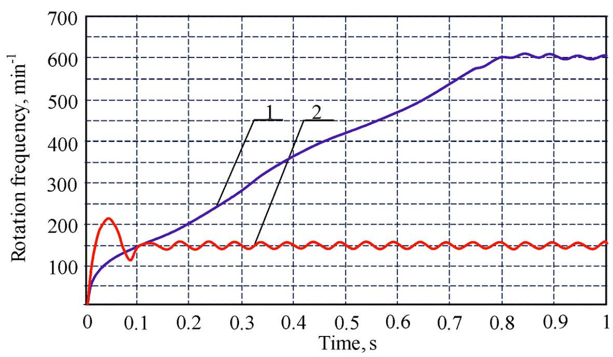

Fig. 16. Dynamics of the change in rotation frequency of shaft of the planetary hydraulic motor of PRG-22 series operating in the mechatronic system: 160 cm^3 working volume (1); 630 cm^3 working volume (2)

Analysis of dynamics of the change in operation of the safety valve (Fig. 17) has shown that the safety valve in a mechatronic system with a 160 cm^3 hydraulic motor was triggered at the initial moment of time and completely open at t=0...0.02 s (curve 1). At this moment, the rate of the working fluid flow through the valve reached a maximum value of Q_3=100 l/min. Further (t=0.02...0.78 s), flow through the safety valve began to fluctuate, decreased and approached zero (curve 1) at t=0.78 s. The nature of the change in the rate of flow through the valve in the mechatronic system with a 630 cm^3 hydraulic motor was characterized by two peak-like surges (curve 2), up to 100 l/min, and the valve was completely closed at t=0.11 s.

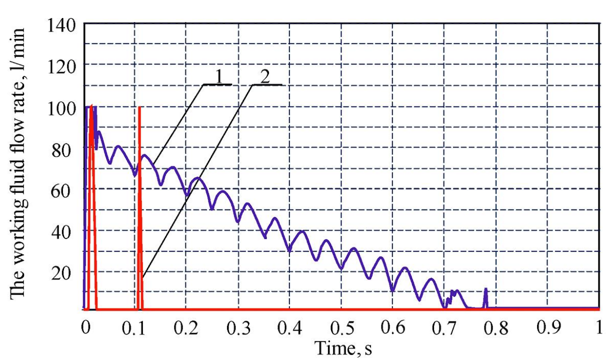

Fig. 17. Dynamics of the change in the rate of flow through the safety valve operating in the mechatronic system with the hydraulic motor working volume 160 cm^3 (1) and 630 cm^3 (2)

Dynamics of change in the working fluid flow rate, Q_2, of the hydraulic motors taking into account operating conditions (Fig. 18) was similar to the change in the flow rate of the working fluid under ideal operating conditions (Fig. 11). Except that the steady operation (t>0.78 for the 160 cm^3 hydraulic motor (curve 1) and t>0.11 for the 630 cm^3 hydraulic motor (curve 2)) was characterized by insignificant fluctuations of flow rate with an amplitude up to 5 l/min.

When analyzing the obtained dependencies characterizing dynamics of the changes in functional parameters of the mechanically driven mechatronic system, it is necessary to note two unfavorable phenomena that impede its normal functioning. The first phenomenon is the pressure "overshoot" in the initial period of acceleration (t=0...0.02 s) caused by inertial loads and the second phenomenon is the fluctuations caused by the set operating conditions, that is, the fluctuations of the pump feed and the moment of resistance (in the steady operation).

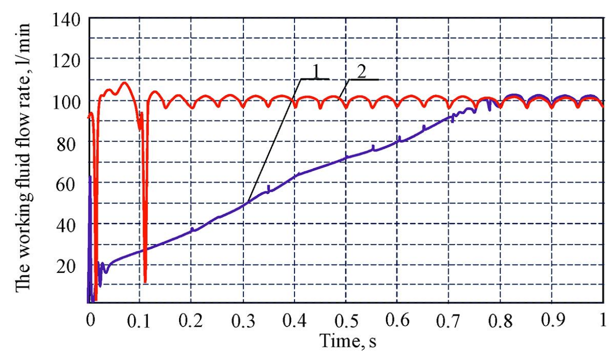

Fig. 18. Dynamics of change in the flow rate of hydraulic motor operating in the mechatronic system: 160 cm^3 working volume (1); 630 cm^3 working volume (2)

The conducted studies have resulted in construction of the universal model of the hydraulically driven mechatronic system which has made it possible to study dynamics of the changes in functional parameters of this system and its elements taking into account the working fluid features.

6. Discussion of results obtained in the simulation of operation of the hydraulically driven mechatronic system

As a result of the conducted studies, the universal mathematical model was proposed. It describes the working processes occurring in the hydraulically driven mechatronic system. The proposed model makes it possible to study dynamics of changes in functional parameters of the mechatronic system which includes the pump, the hydraulic motor, the safety valve and the working fluid during operation. Any hydraulic device of volumetric action can be used as the pump and as the hydraulic motor. Also, the model enables the use of safety valves, both of direct and indirect action.

Dynamics of the changes in functional parameters of the mechatronic system was studied on the example of planetary hydraulic motors of the PRG-22 series with working volumes of 160 cm^3 and 630 cm^3.

Operating conditions were simulated by simulating unevenness of the pump feed and load. Unevenness of the pump feed was obtained by summing half-sinusoids with a time shift and the load fluctuations were obtained according to the exponential and sinusoidal laws with a time delay of 0.05 s.

An analysis of dynamics of the changes in functional parameters of the mechanically driven mechatronic system has shown that operation of its hydraulic drive can be divided into four stages determined by the safety valve operation time. The first stage of operation of the mechatronic system (t=0...0.02 s) is characterized by acceleration of the hydraulic drive and actuation of the safety valve. The second stage (t=0.02...0.78 s; a mechatronic system with a 160 cm^3 hydraulic motor and t=0,02...0.05 s with a 630 cm^3 hydraulic motor) is characterized by the safety valve closure. The third operating stage (t=0.78...0.9 s for a 160 cm^3 hydraulic motor volume and t=0.05...0.3 s for a 630 cm^3 hydraulic motor volume) is characterized by completion of the hydraulic drive acceleration. The fourth stage (t>0.9 s for the 160 cm^3 hydraulic motor and t>0.3 s for the 630 cm^3 hydraulic motor) is characterized by the steady operation.

The conducted studies have established that the operating conditions did not affect the change in the functional parameters of the mechatronic system from the moment of actuation of the safety valve up to its closure during the start of the hydraulic drive. However, fluctuations caused by uneven pump feed and load variation were observed in the steady operation. It should also be noted that the mechatronic system with a hydraulic motor of larger working volume had better dynamic characteristics than the system with a hydraulic motor of smaller working volume.

As a result of the performed studies, negative phenomena impeding normal functioning of the mechanically driven mechatronic system were revealed. Such phenomena include the pressure "overshoot" at the initial stage of acceleration (t=0...0.02 s) caused by inertial loads. Also, fluctuations of functional parameters of the mechatronic system (in a steady operation) caused by operating conditions, that is, fluctuations of the pump feed and the moment of resistance also had an adverse effect.

The proposed universal model of the mechanically driven mechatronic system represents the simplest scheme of the hydraulic power drive for movable operating elements of the self-propelled machinery. Therefore, further studies are supposed to be carried out to further develop the proposed model through a sequential addition of hydraulic elements to the design diagram (distributor, hydraulic cylinder, throttle, hydraulic lines, etc.). To this end, it is necessary to compile a library of the constructed models of hydraulic elements that take into account structural features of these elements. In order to improve the universal model under consideration and the subsequent variants of its further development, it is supposed that experimental studies will be carried out by confirming adequacy to the investigated process. To carry out experimental studies, it is necessary to develop a study conduction procedure and a bench corresponding to the design diagram.

7. Conclusions

1. A universal model of the mechanically driven mechatronic system consisting of the pump, the hydraulic motor and the safety valve has been developed. The proposed model enables simulation of operation of the mechanically driven mechatronic system taking into account the features of functioning and mutual influence of all its elements as well as the features of the working fluid. This model can be used to study dynamics of the change in functional parameters of the mechatronic system for driving movable operating elements of self-propelled machinery with the use of any hydraulic devices of volumetric action.

2. Analysis of dynamics of changes in the functional parameters of the mechatronic system without taking into account the influence of operating conditions has shown that the changes in torque on the shaft of the hydraulic motor completely repeat the changes in pressure in the injection line. These changes are valid at all stages of operation of the hydraulic drive of the mechatronic system determined by the safety valve operating time. When starting the hydraulic drive of the mechatronic system, pressure and torque values increase sharply and exceed their nominal values more than 4 times because of lag of the safety valve actuation. Dynamics of the change in the rotation frequency of the hydraulic motor shaft and the flow rate of the working fluid is virtually linear.

3. An analysis of dynamics of the changes in functional parameters of the mechatronic system taking into account operating conditions shows that when the hydraulic drive is accelerated, operating conditions do not affect the changes in torque and pressure. These changes in functional parameters take place at the time of operation of the safety valve. After the valve is closed, pressure and torque values significantly fluctuate with amplitude equal to 70...80 % and with frequency of the pump feed fluctuation. At the same time, the mean value of parameters varies according to the sinusoidal law brought about by fluctuation of the moment of resistance. Dynamics of the change in rotation frequency of the hydraulic motor shaft and the flow of the working fluid is analogous to the change of values under ideal operating conditions. The values of these parameters have minor deviations (up to 0.5 %) in the steady-state operation of the mechatronic system.

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