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APPLICATION OF HYDRAULIC AUTOMATION EQUIPMENT FOR THE EFFICIENCY ENHANCEMENT OF THE OPERATION ELEMENTS OF THE MOBILE MACHINERY

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Abstract: The application of adaptive drives in working units of various technological and mobile machines is considered. The results of theoretical and experimental research have proved the effectiveness of their use for improving the dynamic characteristics of machines, reducing unproductive power losses during their operation.

Keywords: hydroautomatics, tools, workstations, mobile machines

ZASTOSOWANIE WYPOSAŻENIA AUTOMATYZACJI HYDRAULICZNEJ W CELU POPRAWY EFEKTYWNOŚCI ELEMENTÓW OPERACYJNYCH MASZYN MOBILNYCH

Streszczenie. W pracy rozważane jest zastosowanie napędów adaptacyjnych w jednostkach roboczych różnych maszyn technologicznych i mobilnych. Skuteczność zastosowania tych algorytmów do poprawy charakterystyk dynamicznych maszyn, przy jednoczesnym zmniejszeniu nieproduktywnych strat mocy podczas pracy potwierdziły wyniki badań teoretycznych i eksperymentalnych.

Słowlukuczowe: hydraulika, narzędzia, stacje robocze, maszyny mobilne

Introduction

Greater parts of the mechanisms of various technological designation operate in variable loads mode. In particular, in civil engineering and industry mobile working mechanisms with the manipulators on the base of trucks and wheeled tractors, equipped with various movable working tools are widely used. The operation of the manipulators of the mechanisms with movable working tools to provide optimal performing of the operation requires the proportion regulation of the working fluid consumption, supplied from the pump to the hydraulic motor which is not available in the mobile mechanisms, manufactured in our country and abroad.

Variable loads in the drive systems of the conveyors occur as a result of the irregularity of the load flows, arrived on the conveyor belts of the mobile mechanisms, for instance, clamp formers, beet harvesting machines, etc. This stipulates the onset of the mechanical oscillations, creating the conditions for the fatigue failure of the belt, parts and units of the mechanism. The solution of these problems requires the application of new drive constructions with the automatic adaptation of their parameters to variable operation modes. It enables to provide the decrease of the dynamic efforts in the elements of the drives construction and executive elements, decrease of the unproductive power losses in the process of their operation.

1. Analysis of the hydraulic automatic equipment application for the technological machines with variable loading modes

The development of the hydraulic drives of the mobile and technological machines is accompanied by the wide application of the proportional hydraulic equipment and regulated pumps. Stable trend is the transition to the hydraulics drives, sensitive to loading [4, 5]. Application in such hydraulic drives of the equipment and pumps with the electrohydraulic control enables to improve the accuracy of the machines with hydraulic drive operation, decrease the unproductive power losses [21, 24]. Usage of the hydraulic equipment with electric hydraulic control enables to apply in the hydraulic systems of the last generation freely programmable controllers and provides semi-automatic and automatic operation of modern machines [2, 8, 11]. The controllers, available in the adaptive hydraulic drives allow to realize the adaptive control of the machine, that enables to improve considerably static, dynamic and energy characteristics of the machines, adapt their operation

modes to the constantly changing conditions in the process of the operation [12, 23].

Variable operation modes are also common for the mobile complexes, used in the process of the surface mining of the minerals resources or sugar beet clamping at the sugar mills.

Efficiency factor of the conveyors usage at mining enterprises is, on average, 50...70% by power and 60...70% by the operation duration [22]. Such inefficient usage of the conveyors is connected with the irregular load flows by the amplitude of the loading and interruptions in load supply, etc. These operation modes of the conveyor cause unjustified energy losses, belt and conveyor rollers wear, increase of the no-load run of the belt, down time of the equipment. That is why, greater part of the research [1, 13, 25, 26] is directed at the provision of the coordination of the belt conveyor drive operation modes with the parameters of the load flows by means of belt motion speed control and drum torque.

To provide the continuous operation of the belt conveyors drives, mobile complexes are equipped with and increase their production efficiency it is expedient to apply in the drive the additional hydraulic motor, installed in parallel with the main motor and enables to realize the active reservation of the torque at the drive drum. To switch on the drive it is necessary to install the hydroautomatic devices, sensitive to the load variations on the working organ. The solution of such problems is considered in [3, 6, 7, 16]. However, the suggested methods and control devices require both the technical improvement and theoretical substantiation of their parameters, which will provide the efficient operation of the device in non-stationary operation modes. This problem requires more detailed study and substantial analysis of the dynamic processes, occurring in the conveyors drives with devices or control systems.

2. Hydraulic automatic devices for the operation elements of the mobile machines with variable loads

In the adaptive hydraulic drive of the manipulator for the trucks and working machines of different designation the regulated pump 1 with the regulator 6, proportional hydraulic distribution valve 2, freely programmable controller 3 and sensor 9, 10 (Fig. 1) are used.

Adaptive hydraulic drive provides the operation of the execution mechanism of the machine with trivial power losses. Proportional control of the hydraulic cylinder provides the possibility to minimize the amount of the manipulator motions to the object, that increases the performance of the machine and quality of the executed work.

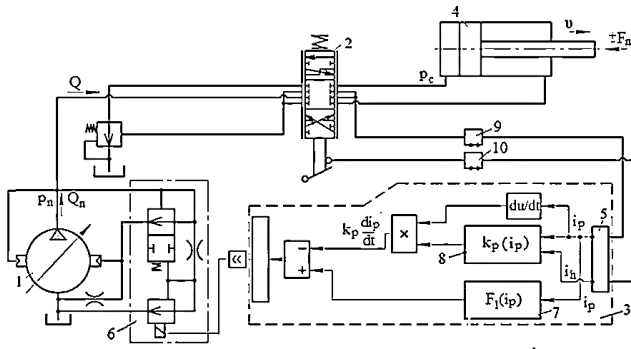


Fig. 1. Adaptive hydraulic drive of the manipulator

Freely programmable controller 3 performs the function of the adaptive regulator, changing the adjustment of the hydraulic drive when the external conditions of its operation change.

Adaptive regulator contains the switching unit 5, feedback unit 7, and correction unit 8. Feedback unit provides the transfer of the signals from the hydraulic cylinder to the regulator of the regulated pump. Unit of the feedback signal correction provides the suppression of the oscillations in the hydraulic drive in the transient processes. Feedback signal correction is realized by means of the addition to it the component $\Delta p_c = -k_p \frac{di_p}{dt}$. k_p is the coefficient of the correction component of the adaptive regulator signal, di_p/dt – is the derivative from the feedback signal. The value of the k_p coefficient greatly influences the quality of the transient processes in the adaptive hydraulic drive. The modeling of the hydraulic drive operation with the adaptive regulation is carried according to the mathematical model [10, 12].

In the adaptive hydraulic drive of the small-size mobile drilling installation (Fig. 2) regulated pump 1, hydraulic motor 2 of the main motion of the tool, hydraulic cylinder 3 of the tool supply, regulated throttle valves 4 and 5 with the electromagnetic control, controller 6, valve 7, hydraulic distribution valves 8, 9, and sensors 10, 11, 12, 13 are used.

Hydraulic drive operates in two modes. In the first mode both branches of the hydraulic drive operate independently. Pump 1 provides the motion of the hydraulic motor 2 and hydraulic cylinder 3 in the independent speed modes.

In the second mode the adaptive regulator 6 provides the coordinated operation of the hydraulic motor 2 and the hydraulic cylinder 3. When the load changes at the hydraulic motor M_1 , the adaptive regulator changes the value of the supply v_2 of the tool in accordance with the algorithm, which is determined by the program of the controller.

When the value of the pressure p_1 , which is determined by the value of the moment M on the hydraulic motor increases, the speed v_2 of the hydraulic cylinder 3 piston motion proportionally decreases. When the pressure value p_{c1} decreases, the speed value v_2 increases.

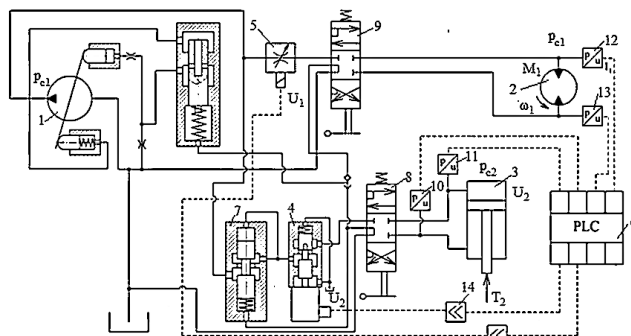


Fig. 2. Adaptive hydraulic drive for the small-size drilling installation

This process is provided by the controller 6 which realizes the cross-coupling in the hydraulic drive. Simulation of the adaptive hydraulic drive operation using the model presented in [9] was

carried out. In case of the step-by-step change of the moment M_1 value from 200 Nm to 800 Nm (Fig. 3) the transient process in the branch, which feeds the hydraulic cylinder 3 passes more dynamically with the readjustments by the value of the pressure p_{c2} .

It is possible to decrease the negative impact of the cross-coupling on the dynamic characteristics of the adaptive hydraulic drive as a result of the introduction of the correction of the cross-coupling signal. Such correction is provided by the controller, which changes the cross coupling signal by the necessary law. It is suggested to form the cross-coupling signal as $U_2 = k_c(p_{c1} - k_p \frac{dp_{c1}}{dt})$. The following denotations are used: k_c is amplification factor of the amplifier 14, k_p is the coefficient of the correction component of the cross-couplingsignal. In the process of the simulation of the mechanotronic hydraulic drive operation the signal of the cross-coupling U_2 was supplied by the controller to the regulated throttle valve 6 with the delay ΔT_1 .

For the belt conveyor of the mobile machine operating in the mode of variable load flows, the built-in hydraulic drive, sensitive to load changes is developed [15], its construction diagram is shown in Fig 3a.

Basic elements of the built-in hydraulic drive construction is the housing of the drum 1, where the drive consisting of two separate hydraulic motors – principal motor 2 and additional motor 3, is built-in, transmission mechanism, consisting of the pinion 4, intermediate 5 and crown 6 toothed wheels. The housing of the drum 1 is mounted on the compound axle, consisting of two half-axes 7 and 8, connected to the housing of the transmission mechanism. The control device 9 in the form of the indirect action valve is installed in one of the radial channels of the half-axle 7 for the supply of the working fluid to the hydraulic motors 2 and 3. The control device (Fig. 3b) contains the valve of the first stage with the conical-cylindrical gate 10, which is pressed by the spring 11 to the seat 12, mounted in the housing 13 and regulating screw 14 for the adjustment for the preset pressure.

The valve of the second stage – tubular slide valve 15 is pressed by the spring 16 to the seat 17. The input of the additional hydraulic motor 3 in case of the switched off control device 9 is connected with the drain by the means of the relief valve 18.

The built-in hydraulic drive operates in the following way. If the pressure at the input in the hydraulic system does not exceed the regulating pressure of the control device 9, then the torque at the drum housing 1 is transferred from the shaft of the main hydraulic motor 2 on across the drive gear 4, intermediate 5 and crown 6 toothed wheels.

In case of the sudden load increase the pressure at the input of the hydraulic system reaches the value p_2 , that exceeds the nominal p_1 , control device 9 is activated. Conical-cylindrical gate 10 is shifted relatively the seat 12 and the working fluid arrives to the drain across the hole 19. At the same time the pressure drop occurs at the throttle valve 20. As a result, the tubular slide valve 15 shifts in the housing 21 together with the rubber seal ring 22 relatively the seat 17 to the left. The working fluid across the hole 23, hydraulic tube 24 arrives to the working chamber of the additional hydraulic motor 3, switching it on in parallel to the principal hydraulic motor 2. Throttle valve 25 is damping the tubular slide valve 15, increasing the dynamic stability of the control device. Besides it forms the valve pot (between the throttle valve 25 and the end face of the tubular slide valve 15), optimal value of its flexibility provides necessary transient characteristics of the valve. The throttle valve 25 does not influence the amplification factor by the losses of the first stage valve and does not worsen the static characteristics of the control valve. As a result of the interaction of the working fluid with the rotor elements of the hydraulic motors 2 and 3 the rotation of their output shafts takes place. The torque from these hydraulic motors across the driving gear 4 intermediate toothed wheels 5, crown wheel 6 is transferred to the drum housing 1. Having lost the energy, the working fluid across the output holes of the hydraulic

motors 2 and 3, pipelines 26 and 27, accordingly, radial 28 and axial 29 channels, located in the half shaft 8, arrives to the drain.

For the stable operation of the hydraulic drive, without vibrations, the characteristic of the control device must correspond to the graphic dependence shown in Fig. 4 (22). Such dependence of the valve opening on the pressure at its input is provided by the construction of the developed control device.

Theoretical studies of the dynamic processes in non-stationary operation modes of the hydraulic drive operation were carried out on the base of the developed mathematical model [14].

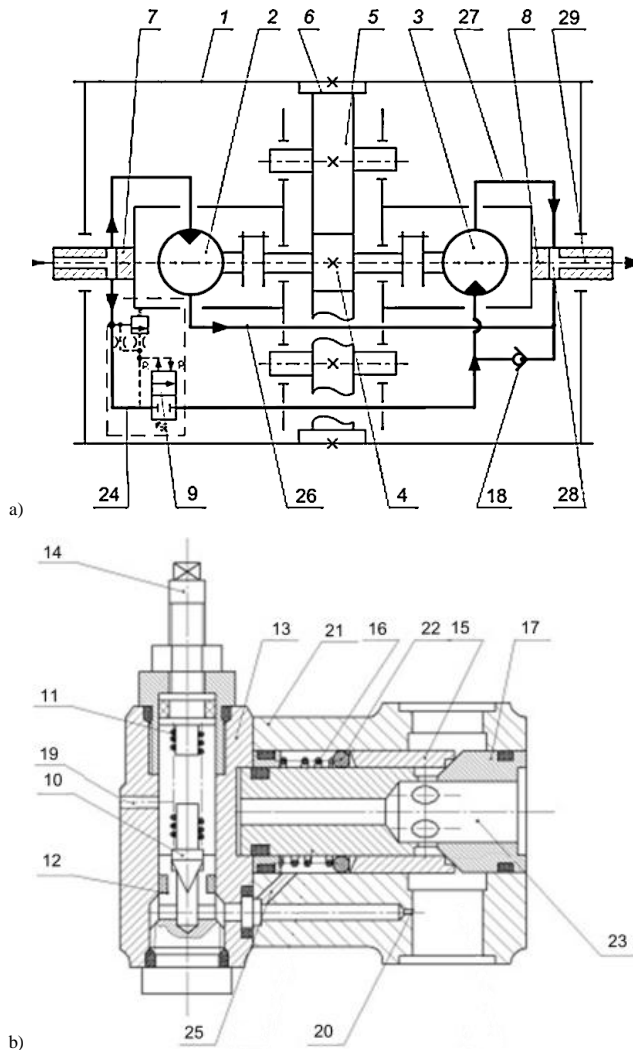


Fig. 3. Construction diagram of the built-in conveyor hydraulic drive, sensitive to the load change

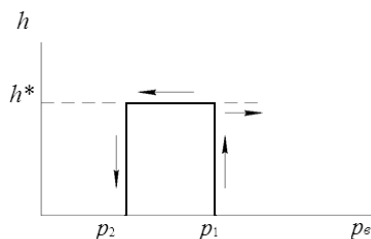


Fig. 4. Dependence of the value of the control drive opening on the pressure at the input of the hydraulic system

For the receiving conveyor of the clamp former K-65M253-K the construction [19] and technical documentation of the built-in hydraulic drive was developed, the construction diagram of the drive is shown in Fig. 5.

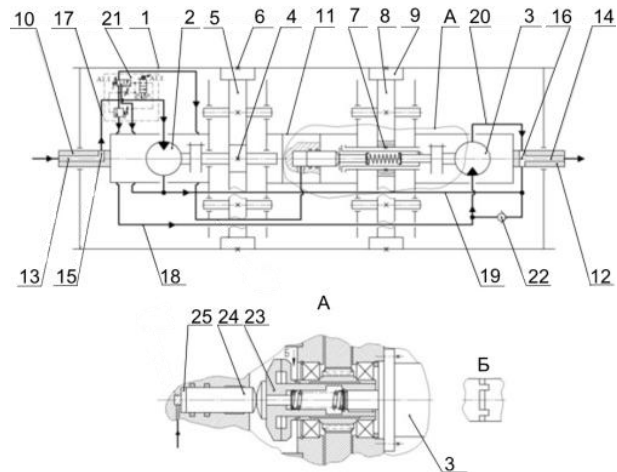


Fig. 5. Construction diagram of the built-in hydraulic drive of the receiving conveyor of the clamp former K-65M253-K

The efficiency of the drive operation is provided [17] applying in it the developed control system with the friction clutch for the switching of the additional hydraulic motor. To reduce the resistance forces during the start of the additional hydraulic motor the control device of such system operates in such mode: first, with the small time delay the shaft displacement of the additional hydraulic motor occurs, then the friction clutch of the driving gear pinion is activated.

The controlled hydraulic electric cylinder motor contains the housing of the drum 1, where the drive is built-in, the drive is realized in the form of two separate hydraulic motors – main 2 and addition 3, and two transmission mechanisms. Each of them consists of the driving gears 4 and 7, intermediate 5, 8 and crown 6 and 9 gear wheels. The housing of the drum 1 is mounted on the axle, consisting of three parts 10, 11, 12. For the supply and drain of the working fluid to the hydraulic motors 2 and 3 in the middle of the left and right parts of the axle 10 and 12 the axial passages 13 and 14 are made. They are connected with the working channels of the hydraulic motors 2 and 3 by means of radial channels 15 and 16, as well as pressure pipes 17 and 18 and drainage pipeline 19 and 20. In the pressure pipeline 17 the control device 21 is installed. The input of the additional hydraulic motor 3, if the control unit 21 is switched off, is connected with the drain across the non-return valve 22.

The control unit 21 is realized in the form of the two-stage valve, provided with the functions of the hydraulic distribution valve.

In the second transmission mechanism the friction coupling 23 is installed between the shaft of the additional hydraulic motor 3 and driving gear 7, its left half coupling across the cone has the contact with the end spherical surface of the pressing plunger 24. It is mounted in the greater diameter of the central stepped hole, realized on the right side of the middle part 11 of the axle. Flat surface of the pressing plunger forms the cavity 25, connected with the cavity of the smaller diameter.

At nominal load on the working element of the conveyor the main hydraulic motor 2 operates, this motor by means of the transmission mechanisms puts in motion the body of the drum 1. At sudden increase of the load, the value of which corresponds to the values of “opening” pressure, control unit 21 starts operation.

First the switching on of the additional hydraulic motor 3 occurs, then friction coupling 23 starts its operation, and the torque from the shaft of the additional hydraulic motor 3 is transferred across the transmission mechanism to the body of the drum 1.

After load reduction on the working element of the conveyor to the value that corresponds to “closing” pressure, control unit 21 is actuated, it disconnects from the supply the additional hydraulic motor 3.

For studying the mutual impact of the parameters of the built-in hydraulic drive of the receiving conveyour with the control system and spring-inertial characteristics of its transporting part mathematical model, consisting of the equations, which describe the motion of the transport and drive parts of the conveyour and the switching control system of the additional hydraulic motor [24] is developed.

3. Results of the study of the hydraulic automation equipment of the mobile machines with variable loading working elements

The studies of the adaptive hydraulic drive of the manipulator determined that the oscillation of the hydraulic drive considerably decreases from 8-10 oscillations to 2-4 oscillations in the transient process, depending on its operation modes. The value of the readjustment decreases on the value of up to 40% and does not exceed 30%, at the operation of the manipulator of 7000 N carrying capacity.

Adaptive hydraulic drives provides high degree of the stabilization of motion speed of the actuating mechanism if the load F_n changes. This enables to forecast better the motion of the actualing mechanism in the operation process, minimize the amount of the necessary motions when the manipulator is pointed on the object. The degree of the stabilization is evaluated by the dependence of the flow Q , that arrives to the hydraulic cylinder on the pressure value p_c at the input in the hydraulic cylinder (Fig. 6). If the pressure value p_c changes in the range of (0...16) MPa, the change of Q flow depends on the set value Q and changes in the range from 3% to 6% at the pressure drop value on the proportional hydraulic distribution valve 1.5 MPa.

The transition on the value of the pressure drop of 1.0 MPa increases the effort of the value Q change from 5% to 10%.

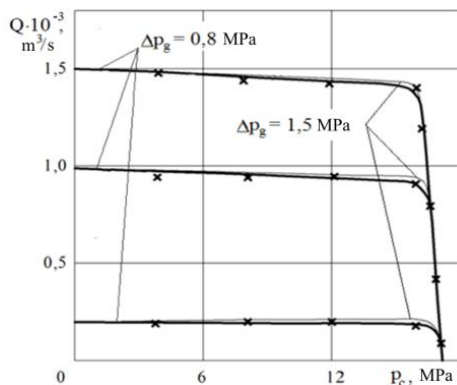


Fig. 6. Dependence of Q on the pressure p_c value at the input in the hydraulic cylinder

The important index of the manipulator operation quality is the pointing accuracy of the clamp on the object which determines the productivity of the machine operation stabilization of the manipulator motion speed allows to increase the accuracy of the clamp pointing on the object [12]. Study of the manipulator pointing accuracy, using the developed adaptive hydraulic drive is carried out. The research is performed on the stand and using the technical, described in [12].

Fig. 7. shows the distribution of the results of the manipulator position measurements while directing it on the object. The results of the measurements are in the range of (37...43) · 10⁻² m. The obtained distribution law is close to normal. Mean square error of the manipulator position deviation is $\sigma_1 = 1.02 \cdot 10^{-2}$ m and the distance between the middle of the field for the manipulator pointing and the arithmetic mean value of the manipulator position $\bar{x} = 40.3 \cdot 10^{-2}$ m is $\Delta x_1 = 0.42 \cdot 10^{-2}$ m.

The obtained results of measurements allow to assess the accuracy of the manipulator pointing on the object as such that allows to minimize the amount of motions in the operation cycle and increase the performance of the machine.

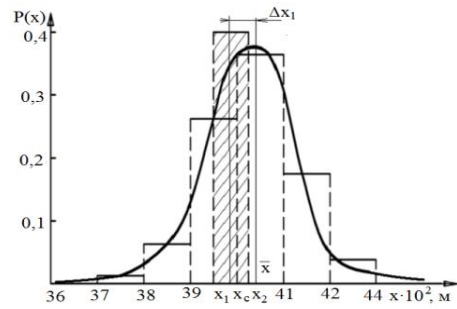


Fig. 7. Determination of the manipulator pointing accuracy on the object

At the stepped change of the moment M_1 value from 200 Nm to 800 Nm (Fig. 8) the transient process in the branch, which feeds the hydraulic cylinder, 3 occurs more dynamically with large readjustment by the pressure p_{c2} value.

The value of the delay changed in the process of the research in the range of $\Delta T_1 = (0.005...0.06)$ sec. The value of k_p coefficient of the correction component of the cross coupling signal varied in the range of $k_p = (40...320) \cdot 10^{-4}$ sec.

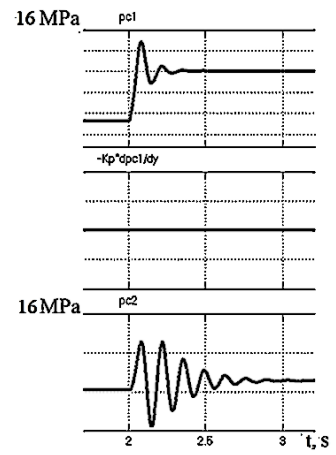


Fig. 8. Transient process in the adaptive hydraulic drive if the correction of the cross communication is missing

At the change of the delay value ΔT_1 of the transient coupling signal of more than 0.04 sec. and less than 0.02 sec. the readjustment in the adaptive hydraulic drive increases (Fig. 9)

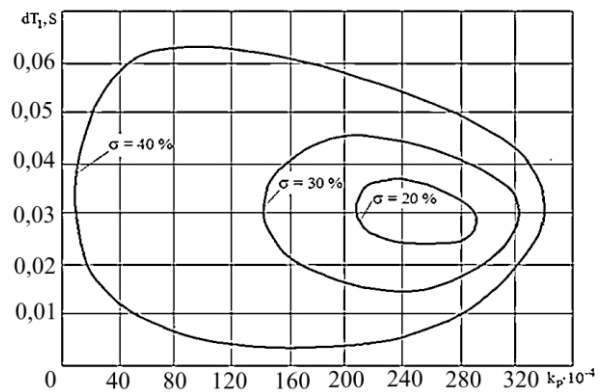


Fig. 9. Impact of the delay time ΔT_1 and the correction component coefficient k_p on the value of the readjustment in the adaptive hydraulic drive

At values of $\Delta T_1 = (0.025...0.04)$ sec. readjustment $\sigma = 20\%$ may be provided. For this purpose it is necessary to select the value k_p from the range $k_p = (200...280) \cdot 10^{-4}$ sec. Change of the coefficient k_p value of less than $200 \cdot 10^{-4}$ sec. and the greater value $k_p = 280 \cdot 10^{-4}$ sec. leads of the increase of the readjustment value σ . The selection of the rational values of k_p and T_1 provides the decrease of the regulation time t_p and readjustment value σ by the pressure p_{c2} at the input in the hydraulic cylinder 3.

Fig. 10 shows the transient process in the adaptive hydraulic drive, calculated by the mathematical model if the correction of the cross coupling signal is available.

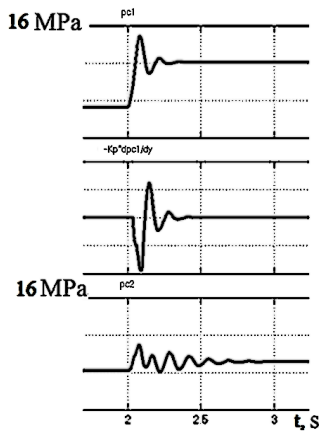


Fig. 10. Transient process in the adaptive hydraulic drive on case of the cross coupling correction

At the values of the correction component coefficient $k_p = 250 \cdot 10^{-4}$ sec. and correction component delay $\Delta T_1 = 0.02$ sec. the regulation time is $t_p = 0.9$ sec. and the readjustment value does not exceed 50%. For the simultaneous decrease of the regulation time t_p and readjustment value σ on the change of the coefficient k_p and delay value ΔT_1 the optimization nproblem must be solved.

The application of the adaptive regulator in the hydraulic drive of the small-size drilling installation enables to provide its operation in the semiautomatic mode.

The operation modes will vary depending on the type of the soil in which the drilling is performed.

The study of the dynamics of the hydraulic drive, sensitive to the load changes (Fig. 3) [18] was carried out on the base in the integration of the differential equations of the developed mathematical model, applying the computer program, created on the base of Matlab 13 package.

Analysis of the conditions of the hydraulically driven systems operation showed that the inertial loading, torsional shiffness, volume of the pressure pipe and the character of the load change influence the transient processes of the hydraulically driven installation [20].

Theoretical dependences regarding the character of the dynamic processes in the hydraulic drive, sensitive to load change have been obtained, one of these dependences is presented in Fig. 11: $p_1(t)$ – pressure in the main line of the main hydraulic motor; p_3 – pressure in the pipeline of the additional hydraulic motor after the activation of the control units; $Q_1(t)$, $Q_2(t)$ – working fluidconsumption across the main and additional hydraulic motor, correspondingly; $n(t)$ – drum rotationfrequency; $M(t)$ – moment on the drum from the acting load.

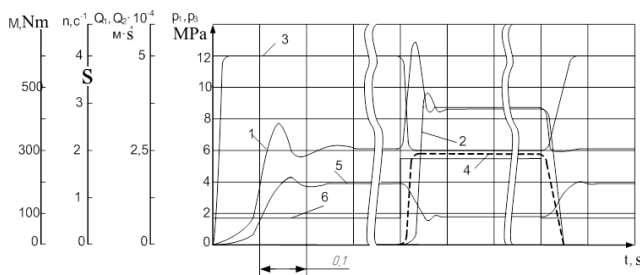


Fig. 11. Theoretical dependences $p_1(t)$, $p_3(t)$, $Q_1(t)$, $Q_2(t)$, $n(t)$, $M(t)$ (curves 1, 2, 3, 4, 5, 6 correspondingly) at $p_{00} = 12$ MPa, torsional stiffness $c_\varphi = 0.37 \cdot 10^{-3}$ Nm/rad, ratio of the area of the conic-cylindrical gate $f_3/f_4 = 0.6$ and step-by-step load change

For the verification of the results of the theoretical studies and to specify the values of certain parameters, needed for carrying out the dynamic calculations experimental studies were performed.

Experimental and theoretical dependences of the impact on the time of the transient process of the start t and the activation of the control unit t_0 during its opening and closing, torsional stiffness of the mechanical part c_φ (Fig. 12), inertia moment I_4 (Fig. 13), volume of the pressure cavity V_1 (Fig. 14) and the character of loads change $M(t)$ are constructed. It is determined that the 3 times increase of the torsional stiffness of the mechanical system of the conveyor causes the decrease of the duration of the transient processes of the start by 26% and control unit opening by 24%, as well as the increase of the readjustment pressures Δp_1 by 80% and control unit opening Δp_{00} by 44%; increase of the inertia load causes the increase of these parameters by 23%, 41%, 30% and 29%, correspondingly; the increase of the volume of the pressure cavity V_1 increases the duration of the transient processes and reduces greatly the pressure readjustment. The comparison of the results of the theoretical and experimental research is carried out.

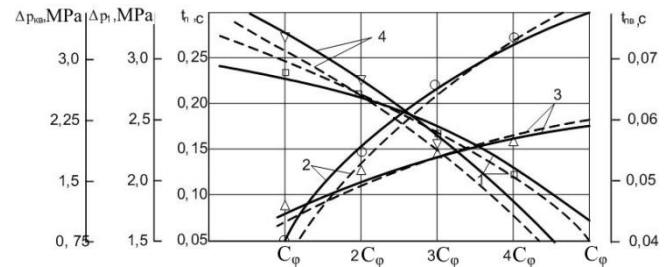


Fig. 12. Graph of the dependences $t_n(c_\varphi)$, $\Delta p_1(c_\varphi)$, $\Delta p_{00}(c_\varphi)$, $t_{00}(c_\varphi)$ (curves 1, 2, 3 and 4 correspondingly) at $p_{00} = 12$ MPa, torsional stiffness $c_\varphi = 0.37 \cdot 10^{-3}$ Nm/rad; $f_3/f_4 = 0.6$ and step-by-step change of the resistance moment M_0 from 87 Nm to 280 Nm (- - - theoretical, — experimental)

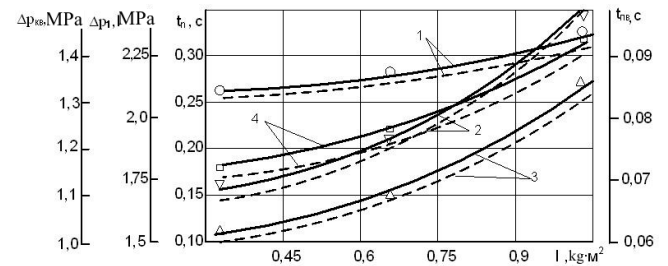


Fig. 13. Graph of the dependences $t_n(I)$, $\Delta p_1(I)$, $\Delta p_{00}(I)$, $t_{00}(I)$ (curves 1, 2, 3 and 4 correspondingly) at $p_{00} = 12$ MPa, torsional stiffness $c_\varphi = 0.37 \cdot 10^{-3}$ Nm/rad; $f_3/f_4 = 0.6$ and step-by-step change of the resistance moment M_0 from 87 Nm to 280 Nm (- - - theoretical, — experimental)

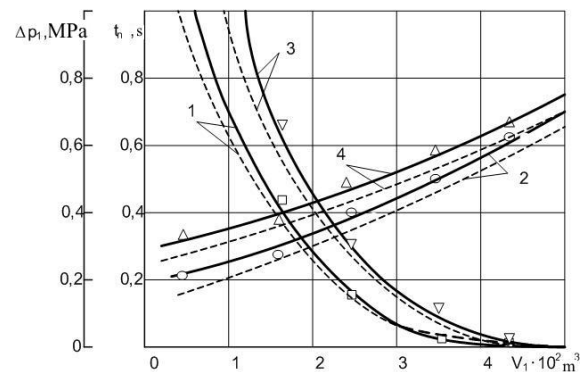


Fig. 14. Graph of the dependences $\Delta p_1(V_1)$, $t_n(V_1)$ (curves 1, 2 correspondingly) if the resistance moment is missing and $\Delta p_1(V_1)$, $t_n(V_1)$ (curves 3, 4 correspondingly) at $p_{00} = 12$ MPa, torsional stiffness $c_\varphi = 0.37 \cdot 10^{-3}$ Nm/rad; $f_3/f_4 = 0.6$ and step-by-step change of the resistance moment M_0 from 87 Nm to 280 Nm (--- theoretical, — experimental)

Theoretical investigations of the hydraulic drive of the clamp former receiving conveyor with the control system of the additional hydraulic motor switching were carried out. The solution of the system of the non-linear differential equations of the mathematical model was performed by means of the programming package Matlab Simulink. The study of the dynamic characteristics for the driving and transporting parts of the conveyor was performed on the base of the analysis of the

transient processes graphs $M(t)$, $p_n(t)$, $p_1(t)$, $p_2(t)$, $x(t)$, $y(t)$, $z(t)$ (Fig. 15). The moment on the drum changed from the nominal value of 5 kN·m to the overload value of 12 kN·m. Overload mode is characterized by three phases of the transient processes: the first – switching on of the additional hydraulic motor (0.11 sec $< t_{p1} < 0.15$ sec, 15% $< \Delta p_1 < 45\%$); the second – operation of the drive with two hydraulic motors (0.05 sec $< t_{p2} < 0.12$ sec, 4% $< \Delta p_2 < 12\%$); the third – load decrease and disconnection of the additional hydraulic motor (0.9 sec $< t_{p3} < 0.17$ sec, 20% $< \Delta p_3 < 55\%$).

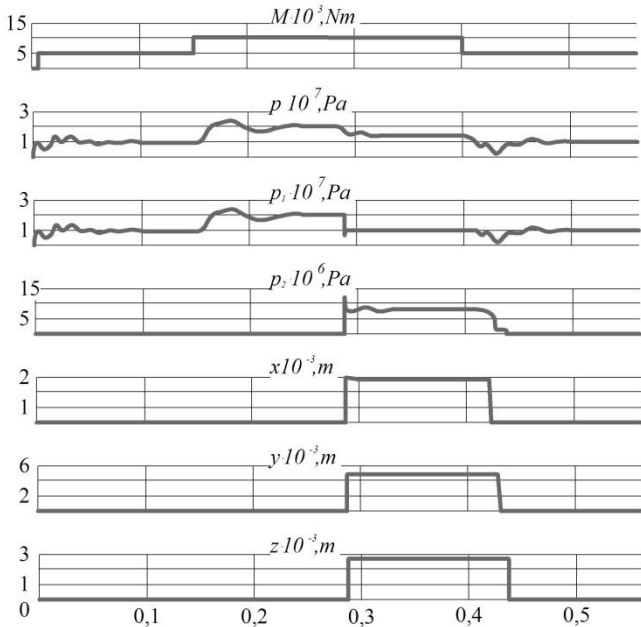


Fig. 15. Theoretical graphs of the transient processes of $M(t)$, $p_n(t)$, $p_1(t)$, $p_2(t)$, $x(t)$, $y(t)$, $z(t)$ change

For each of the above-mentioned phases of the transient process the impact of the pressure increase value Δp in the pressure pipe, torsional stiffness c_f and transmission mechanism damping, aggregate moment of the drum I_b inertia, static E_s and dynamic h_s belt elasticity modulus, conveyor length L_s was analyzed (Fig. 16, Fig. 17)

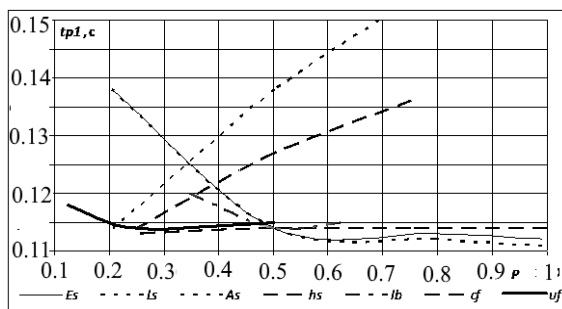


Fig. 16. Impact of the parameters of the driving and transporting parts of the conveyor on the transient process time for the first phase of the overloading

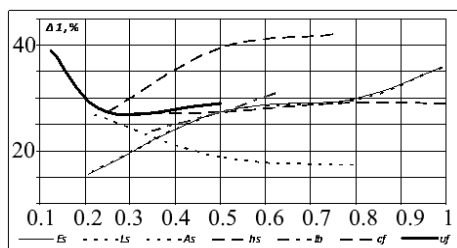


Fig. 17. Impact of the parameters of the driving and transporting parts of the conveyor on the pressure readjustment for the first phase of the overloading

The obtained dependences will allow to optimize the duration of the transient processes t_p and the pressure increase Δp (and the dynamic component of the loading in the belt) at the expense of the selection of the above-mentioned parameters.

4. Conclusions

The research proves the efficiency of using the hydraulic automatic devices for the improvement of the technical-economic indices of the operation elements of the mobile machines.

The operation of the adaptive regulator at the optimal valve of the greatly decreases the readjustment value and regulation time at the expense of the intensive quench of the drive oscillations in the transient processes. It is determined that the value of coefficient depends on the operation modes of the hydraulic drive, namely, on the value of the working fluid supply Q_n to the hydraulic motor and pressure value p_c in the hydraulic drive.

The operation of the adaptive hydraulic drive in the small-size drilling installation provides the increase of the performance and decreases the readjustment by 30 – 60 %, depending on the operation modes that reduces the dynamic loading on the elements of the hydraulic drive.

The suggested hydraulic automation facilities provide trouble-free operation of the conveyor in the conditions of varying useful resistance forces and decrease of the dynamic impact in the transporting element.

Application of the control system with the valve-type device, having the functions of the distributor and the friction clutch for the disconnection of the additional hydraulic motor of the built-in hydraulic drive enabled to improve its dynamic characteristics.

Compatibility of the considered adaptive drives of the technological and mobile machines [21] has been proved by the comparative economic calculations.

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