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DEVELOPMENT AND OPTIMIZATION OF THE CONTROL DEVICE FOR THE HYDRAULIC DRIVE OF THE BELT CONVEYOR

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Abstract. *Designing new hydraulic equipment requires consideration of the peculiarities of the operating modes of the technological machines. This particularly applies to the hydraulic drive of a belt conveyor, which operates under variable load conditions. Therefore, investigations aimed at reducing static and dynamic characteristic indicators, design calculations, as well as three-dimensional modelling, are relevant tasks for engineers and scientists. Various optimization methods for dynamic processes have been considered. Preference has been given to the linear programming method as the main one for optimizing dynamic processes in the hydraulic drive under overload conditions. Fundamental schemes of the belt conveyor hydraulic drive and control device have been developed and their principles of operation described. Nonlinear differential equations in Cauchy form have been formulated and solved using the MATLAB Simulink software package. A comprehensive criterion for optimizing the static and dynamic characteristics of the belt conveyor hydraulic drive has been developed. Graphs of dynamic processes before and after optimizing static and dynamic characteristics have been provided depending on the parameters of the control device construction. The minimum value of the comprehensive criterion optimization corresponding to rational parameters of the control device construction has been calculated. Based on these parameters, a three-dimensional model of the control device has been developed. The obtained research and calculations will be useful for engineers and scientists during the development and design of new hydraulic equipment.*

Keywords: optimization, hydraulic drive, control device, mathematical model

ROZWÓJ I OPTYMALIZACJA URZĄDZENIA STERUJĄCEGO NAPĘDEM HYDRAULICZNYM PRZENOŚNIKA TAŚMOWEGO

Streszczenie. *Projektowanie nowego sprzętu hydraulicznego wymaga uwzględnienia specyfiki trybów pracy maszyn technologicznych. Dotyczy to w szczególności napędu hydraulicznego przenośnika taśmowego, który pracuje w warunkach zmiennego obciążenia. Dlatego też badania mające na celu obniżenie wskaźników charakterystyk statycznych i dynamicznych, obliczenia projektowe, a także modelowanie trójwymiarowe stanowią istotne zadania dla inżynierów i naukowców. Rozważono różne metody optymalizacji procesów dynamicznych. Preferencję nadano metodzie programowania liniowego jako głównej metodzie optymalizacji procesów dynamicznych w napędzie hydraulicznym w warunkach przeciążenia. Opracowano podstawowe schematy napędu hydraulicznego przenośnika taśmowego i urządzenia sterującego oraz opisano ich zasady działania. Sformułowano i rozwiązano nieliniowe równania różniczkowe w postaci Cauchy'ego przy użyciu pakietu oprogramowania MATLAB Simulink. Opracowano kompleksowe kryterium optymalizacji charakterystyk statycznych i dynamicznych napędu hydraulicznego przenośnika taśmowego. Przedstawiono wykresy procesów dynamicznych przed i po optymalizacji charakterystyk statycznych i dynamicznych w zależności od parametrów konstrukcji urządzenia sterującego. Obliczono minimalną wartość kompleksowej optymalizacji kryterium odpowiadającej racjonalnym parametrom konstrukcji urządzenia sterującego. Na podstawie tych parametrów opracowano trójwymiarowy model urządzenia sterującego. Uzyskane badania i obliczenia będą przydatne dla inżynierów i naukowców podczas opracowywania i projektowania nowych urządzeń hydraulicznych.*

Słowa kluczowe: optymalizacja, napęd hydrauliczny, urządzenie sterujące, model matematyczny

Introduction

The primary type of drive for the executive units of mobile machines and complexes is hydraulic, which provides quite extensive functional capabilities with relatively small dimensions and metal consumption. The development of hydraulic drives based on new or improved hydraulic equipment with optimized dynamic processes to improving both the technical indicators and the efficiency and productivity of these machines [2, 21, 32].

The presented research was conducted for the developed fundamental scheme of the integrated hydraulic drive of the belt conveyor. Significant contributions to the development of hydraulic conveyors, bucket elevators, and monorails have been made by the authors of the studies [27, 31], which take into account their application features in the food industry. Works [5, 20] examine the fundamental scheme of the belt conveyor hydraulic drive and analyze the main reasons for failures during the transportation of bulk materials caused by harsh operating conditions and failures [40].

Modeling dynamic processes [3, 16, 37] in machine hydraulic drives aims to achieve stable operation [8, 26], high-quality static and dynamic characteristics [1, 18, 39] and rational design parameters of the hydraulic equipment [14, 30] for its design. The obtained research results allow improving equipment performance indicators and creating new methods and methodologies [4, 25, 36], justifying modern needs in engineering, transportation, agriculture, and other fields.

In studies of hydraulic processes, researchers pay attention to the influence of temperature and viscosity of the working fluid in the channels of the hydraulic drive on its effective operation [9, 19], energy loss of the hydraulic drive, and delay in its operation associated with pipe connections and fittings [13, 33]; the effect of wave processes in hydraulic oil on the functional characteristics of the hydraulic drive [11]. These research results are taken into account when making assumptions in developing the mathematical model and directly in nonlinear differential equations.

To optimize the characteristics of the control device design, the linear programming method (LP method) [6, 7] was used, with the optimization criterion ensuring the search for the minimum value of negative characteristics. In addition, other optimization methods [12] have been analyzed, and the particle swarm optimization [17, 24] has been identified as a modern and promising method when the exact value of the optimization function gradient is unknown. However, premature convergence of the swarm may settle prematurely around a non-optimal solution, hindering further exploration of the search space.

After calculating the rational parameters of the hydraulic equipment design, the next step is to construct its three-dimensional model [38]. This process is carried out in the modern CAD system environment SolidWorks, using advanced modeling approaches [35].

The obtained three-dimensional model is useful for manufacturing an experimental prototype and conducting a complex of virtual experiments [28, 31], allowing to confirm the theoretically obtained results and assumptions. Additionally, to save costs on research, the three-dimensional model can be used for simulation modeling of the flow of the working fluid in the control device channels. Simulation modeling [10, 33] allows finding the characteristics of the hydraulic equipment and taking them into account during mathematical modeling of the hydraulic drive as a whole. This contributes to increasing the adequacy of the mathematical model to physical processes occurring in the studied device.

1. Statement of the research problem

Therefore, the purpose of the study is to develop a control device by optimization based on static and dynamic characteristics in the belt conveyor hydraulic drive to find the necessary parameters. The conducted research will allow to develop a three-dimensional model of the control device, which will be useful to engineers and scientists for conducting simulation studies or making an experimental prototype.

2. Research and modelling method

This section covers the transition from fundamental scheme of the hydraulic drive with the control device for the belt conveyor to describing its mathematical models in MATLAB Simulink software package.

The fundamental diagram of the belt conveyor hydraulic drive (see Fig. 1) consists of a drum 1, in which two hydraulic motors are embedded: the main 2 and the reserve 3, and two transmission mechanisms: the main 4 and the reserve 5. The pumping 6 and drain 7 hydraulic lines are connected to the drum body 1, which are connected to the main pump 8 and the tank 9, respectively. A relief valve 10 is connected to the pumping hydraulic line 6, which is also connected to the tank 9. In addition, a hydraulic pump check valve 11 is connected to the pumping hydraulic line 6, which is also connected to the tank 9 through the reserve pump 12 and the normally open valve 13. The main pump 8 and the reserve pump 12 receive energy simultaneously from one electric motor 14. The normally open valve 13 is connected to the power supply unit 15 at 12 V and the relay contact 16 from the relay 17. The power supply unit 15 is connected to the relay 17 and contacts X and Y of the control device 18 (see Fig. 2). The control device 18 is also connected to: the pumping hydraulic line 6 with connector P, the drain hydraulic line 7 with connector T, the pressure plunger 19 with connector A, the main hydraulic motor 2 with connector B, the reserve hydraulic motor 3, and the check valve 20 with connector C. Outputs from the main 2 and reserve 3 hydraulic motors and the check valve 20 are also connected to the drain hydraulic line 7. The pressure plunger 19 is designed to interact with the friction clutch 21, which provides torque transmission from the reserve hydraulic motor 3 to the drum body 1 through the reserve transmission mechanism 5.

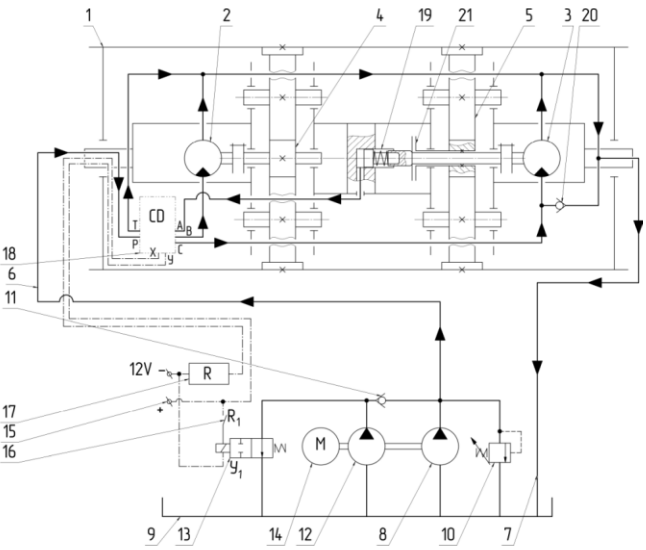

Fig. 1. The fundamental scheme of the hydraulic drive for the belt conveyor with a control device

The control device 18 is designed as a two-stage hydraulic unit: the first 22 and the second 23 stages. The first stage 22 includes a ball-type valve 24 pressed by the adjustable spring of the valve 25. The end face of the valve 23 has a stepped shape and two sealing areas f_1 and f_2 ($f_2 > f_1$), which determine the pressures for its "opening" and "closing". Area f_1 is connected to the spring-loaded end face of the spool 26 and the throttle of the spool 27, which are connected to port P. Area f_2 is connected through the throttle of the valve 28 to port T. The second stage 23 includes a spool valve 26 of the valve-spool type, equipped with a magnetic ring 29 and pressed by the adjustable spring of the spool 30 to the bore in the housing, connected to port C. The magnetic ring 29 operates in conjunction with the sensor 31, which is connected to contacts X and Y. Port P is directly connected to port B, the throttle of the spool 27, and through the spool 26 to port C. Port A is connected to port T if window h_b is open or to port P if window h_d is open (where $h_d > h_b$).

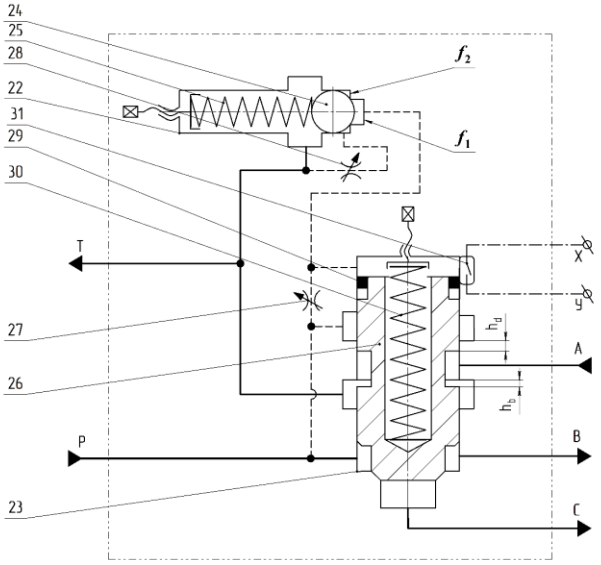

Fig. 2. The fundamental scheme of the two-stage control device

The control device 18 provides the following functions: tracking of the conveyor belt load, activation of backup power during overload, and disconnection of backup power during normal load conditions. Backup power is supplied to the hydraulic drive by connecting the backup hydraulic motor 3 and the backup hydraulic pump 12. According to previous calculations [22], backup power should not exceed 25% of the main power to ensure stable operation of the control device.

The load tracking function of the conveyor belt is achieved by using the first stage 22 and the throttle of the spool 27 in the control device 18. An increase in load above the nominal value will increase the pressure in the pumping hydraulic line 6, causing the valve 24 to open and creating a pressure drop across the throttle of the spool 27. As a result, the spool 26 of the second stage 23 will switch.

The next function, supplying power to the mechanical system during overload, is provided by switching the spool 26 of the second stage 23. In this case, the sensor 31 operates, and by turning on the electrical circuits of the sensor 31 and relay 17, the relay contact 16 and the electromagnet of the normally open valve 13 are connected, allowing the backup pump 12 to be connected to the pumping hydraulic line 6. When the spool 26 opens, the flow of working fluid enters to activate the reserve hydraulic motor 3 and connect its output shaft to the backup transmission mechanism 5 through the interaction of the plunger 19 with the friction clutch 21.

After reducing the load below 70% of the maximum value, the function of disconnecting the backup power until returning to the nominal load mode is implemented. During this, actions in the hydraulic drive are performed in reverse order to those described in previous functions. However, disconnection of the friction clutch 21 occurs with a delay, which may lead to the formation of a vacuum in the hydraulic line. To prevent this negative phenomenon, the hydraulic drive scheme includes the installation of a check valve 20 [22]. This allows the reserve hydraulic motor 3 to pump the working fluid from the drain line 7 back into it without significant positive or negative impact.

An important element of the fundamental scheme of the hydraulic drive of the belt conveyor is the presence of a relief valve 10, which operates when the available reserve power is insufficient to overcome the current load on the belt conveyor. The relief valve 10 protects the hydraulic drive of the belt conveyor from breakdowns by diverting the working fluid under high pressure from the main 8 and reserve 12 pumps directly into the tank 9.

According to the calculated scheme of the hydraulic drive of the belt conveyor, a mathematical model has been developed based on the adopted assumptions, which are characteristic for the hydraulic drives of mobile machines [15, 23]. The equations of the mathematical model are built on the principle of D'Alembert regarding the forces and moments acting on the moving elements, as well as the balance of the working fluid flow in the hydraulic drive. In addition, the actions of viscous friction forces, changes in the directions of movement of the working fluid, energy dissipation during the movement of moving elements, and hydrodynamic forces opening the spools are taken into account. The research results allow for the optimization of dynamic processes in the hydraulic drive of the belt conveyor to determine the rational parameters of the control device.

To develop an algorithm for solving the nonlinear differential equations of the mathematical model, they are transformed into Cauchy form as follows:

– equations of equilibrium of forces acting on the valve 24, spool 26, and plunger 19:

$$x=\iint \frac{p_1 f_2 - k_1(x_0+x) - b_1\dfrac{dx}{dt} - F_{g1}}{m_1} \tag{1}$$

$$y=\iint \frac{p_n f_3 - k_2(y_0+y) - b_2\dfrac{dy}{dt} - p_1 f_4 - F_{g2}}{m_2} \tag{2}$$

$$z=\iint \frac{p_2 f_5 - k_3(z_0+z) - b_3\dfrac{dz}{dt} - F_a}{m_3} \tag{3}$$

– equations of flow continuity condition for pumping hydraulic line 6, hydraulic lines of the sensor 24 and the plunger 19:

$$p_n=\int \frac{(q_{n1}+q_{n2})n_n - \mu f_d\sqrt{\dfrac{2|p_n-p_1|}{\rho}}\,sign(p_n-p_1)}{\beta W_n} - \frac{(q_{m1}+q_{m2})\dfrac{df}{dt} + \mu d_2(y-h_b-h_d)\pi\sqrt{\dfrac{2|p_n-p_2|}{\rho}}\,sign(p_n-p_2)}{\beta W_n} \tag{4}$$

$$p_1=\int \frac{\mu f_d\sqrt{\dfrac{2|p_n-p_1|}{\rho}}\,sign(p_n-p_1) - \mu d_1\pi x\sqrt{\dfrac{2|p_1|}{\rho}}\,sign(p_1)}{\beta W_1} \tag{5}$$

$$p_2=\int \frac{\mu d_2(y-h_b-h_d)\pi\sqrt{\dfrac{2|p_n-p_2|}{\rho}}\,sign(p_n-p_2) - f_5\dot{z}}{\beta W_2} \tag{6}$$

– equations of balance of moments on hydraulic drive shafts:

$$\phi=\iint \frac{q_{m1}p_n + q_{m2}p_n - C_f(\phi-u\phi_3) - \nu_f(\dfrac{d\phi}{dt}-\dfrac{d\phi_3}{dt})}{I_1+I_2} \tag{7}$$

$$\phi_3=\iint \frac{uC_f(\phi-u\phi_3) - \nu_f(\dfrac{d\phi}{dt}-u\dfrac{d\phi_3}{dt})}{I_3} - \frac{2C_s(\phi_3 r_3-\phi_4 r_4)r_3 + 2\nu_s(\dfrac{d\phi_3}{dt}r_3-\dfrac{d\phi_4}{dt}r_4)}{I_3} \tag{8}$$

$$\phi=\iint \frac{2C_s(\phi_3 r_3-\phi_4 r_4)r_4 + 2\nu_s(\dfrac{d\phi_3}{dt}r_3-\dfrac{d\phi_4}{dt}r_4) - M - M_0}{I_4} \tag{9}$$

Solving nonlinear differential equations by Cauchy form in Eq. (1) – (9) was performed using the computer software package MATLAB Simulink. The MATLAB Simulink software package allows creating block diagrams of equation systems from a library of standard blocks, conducting dynamic process simulations, and obtaining data for further analysis. The main block diagram of solving the system of nonlinear differential equations is shown in Fig. 3. The main block diagram includes nine subsystems corresponding to the equations of the mathematical model written by Cauchy form in Eq. (1) – (9). Scope blocks serve as a means of visualizing graphical dependencies of state variables over time obtained as a result of solving the differential equations of the mathematical model.

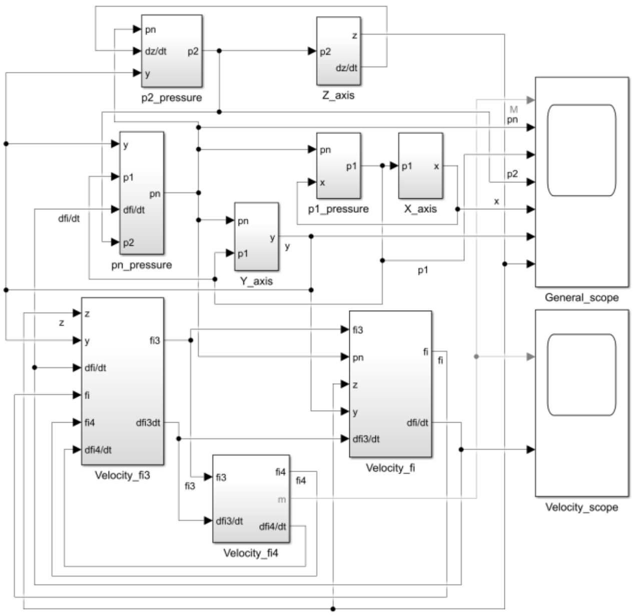

Fig. 3. The main block diagram for solving the system of nonlinear differential equations of the mathematical model in the MATLAB Simulink environment

For accurate and fast computation of dynamic processes in the mathematical model, the ode23s function based on the second-order Rosenbrock modified one-step method was used. It ensures high computation speed for stiff systems with an accuracy of up to 0.1% with our settings.

We perform optimization of static and dynamic characteristics of the belt conveyor hydraulic drive to calculate rational combinations of control device design parameters. Optimization was carried out based on the LP method. The comprehensive optimization criterion W_{opt} is calculated by the formula:

$$W_{opt}=0.3\frac{\sigma_{oni}}{\sigma_{on\max}}+0.2\frac{\sigma_{offi}}{\sigma_{off\max}}+0.5\frac{\delta_i}{\delta_{\max}} \tag{10}$$

where i is the experiment number; σ_{oni}, $\sigma_{on\max}$, σ_{offi}, $\sigma_{off\max}$ – are the overshoot values of pressures p_n and their maximum overshoot in the series of experiments; δ_i, δ_{max} – are the stabilization error values of the rotation velocity $d\varphi/dt$ and the maximum stabilization error of the rotation velocity $d\varphi/dt$ in the series of experiments.

The optimization coefficients in the Eq. 10 are selected as 0.3, 0.2, and 0.5 considering the purpose of the work: to find rational control device parameters that primarily provide the best performance in stabilizing the rotation speed of the hydraulic motor shafts, as well as minimal overshoot values after turning the control device on and off. Moreover, the control device is activated at higher pressures, hence the optimization coefficient for this parameter is 0.3, and for the overshoot value after turning off the control device – 0.2. As a result, the comprehensive optimization criterion W_{opt} will ensure a result less than 1. Rational values of control device parameters correspond to a lower comprehensive optimization criterion W_{opt}.

3. Research results

Let's consider the optimization of dynamic processes based on static and dynamic characteristics in the hydraulic drive according to the results of mathematical modeling. The output variables for the study are the rotational speed indicators $d\varphi/dt$ of the hydraulic motor shafts and the pressure p_n in the delivery hydraulic line. The load on the belt conveyor will increase from the nominal mode of 5 kN·m to the overload mode of 11 kN·m.

The influence of parameters on static and dynamic characteristics has been analyzed. Table 1 presents the correspondence of the investigated ranges and their impact on static and dynamic characteristics. Conditional designations were used in table 1: "s" – a parameter whose change directly affects the system stability (this is due to the parametric structure of the valve and the characteristic ratio of 0.7 for valve opening and closing); "+++" – parameters that significantly (70 – 100% of the maximum value) change the characteristics of the hydraulic drive in the specified range; "++" – parameters that have a minor (30 – 70%) impact on the characteristics of the hydraulic drive in the specified range; "+" – parameters that have almost no effect (5 – 30%) on the characteristics of the hydraulic drive in the specified range; "0" – parameters that do not affect (up to 5%) the characteristics of the hydraulic drive in the specified range.

Table 1. Impact of parameters on static and dynamic characteristics e

parameters	range of change	units	influences
Valve of the first cascade			
d_1	$7.94\cdot10^{-3}$	[m]	s
k_1	$30\cdot10^3…46\cdot10^3$	[N/m]	+++
b_1	400…1200	[N·s/m]	+++
m_1	$20\cdot10^{-3}...50\cdot10^{-3}$	[kg]	0
Spool of the second cascade			
d_2	$10\cdot10^{-3}…25\cdot10^{-3}$	[m]	+
k_2	$10\cdot10^3…27\cdot10^3$	[N/m]	++
b_2	20…70	[N·s/m]	++
m_2	$50\cdot10^{-3}…150\cdot10^{-3}$	[kg]	0
Throttle and hydraulic line volume			
f_d	$1.8\cdot10^{-6}…5.0\cdot10^{-6}$	[m²]	+++
W_n	$0.5\cdot10^{-3}…1.5\cdot10^{-3}$	[m³]	++

For mathematical modeling, we select the ranges of values from table 1 as follows: $k_1 = (30\cdot10^3…46\cdot10^3)$ N/m; $b_1 = (400…1200)$ N·s/m; $f_d = (1.8\cdot10^{-6}…5.0\cdot10^{-6})$ m². Modeled dynamic processes in the conveyor belt hydraulic drive under adverse static and dynamic characteristics within this range of pressure p_n in the pumping hydraulic line and rotational velocity $d\varphi/dt$ of the hydraulic motor shaft are shown in Fig. 4.

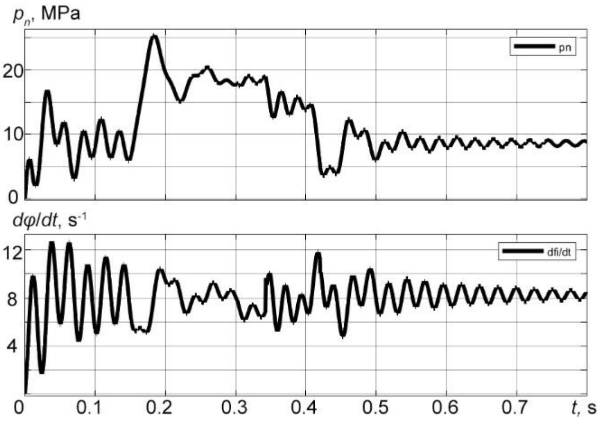

Fig. 4. Adverse dynamic processes within the pressure p_n in the pumping hydraulic line and the rotational velocity $d\varphi/dt$ of the hydraulic motor shaft

During the research, 27 experiments were conducted at minimum, average, and maximum values of k_1, b_1, f_d from the specified ranges in table 1. The research results are provided in table 2.

In experiment no 4, the combination of selected parameters $k_1 = 30\cdot10^3$ N/m; $b_1 = 800$ N·s/m; $f_d = 1.8\cdot10^{-6}$ m² ensures optimal static and dynamic characteristics: $\sigma_{on4} = 7.46\%$, $\sigma_{off4} = 35.09\%$, $\delta_4 = 7.84\%$, corresponding to the comprehensive optimization criterion $W_{opt} = 0.4438$. In this case, the duration of the switching-on dynamic process of the control device is $t_{on4} = 0.11$ s, and the duration of the switching-off dynamic process of the control device is $t_{off4} = 0.14$ s.

Table 2. Impact of control device parameters on static and dynamic characteristics of the hydraulic drive

No	k_1, [N/m]	b_1, [N·s/m]	f_d, [m²]	σ_{oni}, [%]	σ_{offi}, [%]	δ_i, [%]	W_{opt}
1	**$30\cdot10^3$**	400	$1.8\cdot10^{-6}$	8.03	37.21	7.71	0.4562
2			$3.0\cdot10^{-6}$	7.14	34.88	13.01	0.5722
3			$4.2\cdot10^{-6}$	13.01	33.72	19.57	0.782
4		**800**	**$1.8\cdot10^{-6}$**	**7.46**	**35.09**	**7.84**	**0.4438**
5			$3.0\cdot10^{-6}$	6.8	34.89	13.76	0.5886
6			$4.2\cdot10^{-6}$	6.05	33.72	19.57	0.7248
7		1200	$1.8\cdot10^{-6}$	7.48	35.67	7.84	0.447
8			$3.0\cdot10^{-6}$	6.8	35.09	13.01	0.5705
9			$4.2\cdot10^{-6}$	6.26	34.5	19.57	0.7306
10	$38\cdot10^3$	400	$1.8\cdot10^{-6}$	27.21	37.21	6.45	0.5814
11			$3.0\cdot10^{-6}$	26.53	36.04	11.49	0.6985
12			$4.2\cdot10^{-6}$	25.85	36.05	16.19	0.8131
13		800	$1.8\cdot10^{-6}$	34.01	38.01	6.87	0.6521
14			$3.0\cdot10^{-6}$	31.29	37.91	11.49	0.7473
15			$4.2\cdot10^{-6}$	31.29	36.04	16.2	0.8579
16		1200	$1.8\cdot10^{-6}$	34.01	38.5	6.87	0.6547
17			$3.0\cdot10^{-6}$	34.36	35.26	11.49	0.7588
18			$4.2\cdot10^{-6}$	36.55	36.05	16.9	0.9191
19	$46\cdot10^3$	400	$1.8\cdot10^{-6}$	35.37	38.01	5.77	0.6352
20			$3.0\cdot10^{-6}$	32.65	37.65	10.29	0.7265
21			$4.2\cdot10^{-6}$	34.01	37.2	13.01	0.8048
22		800	$1.8\cdot10^{-6}$	32.88	38.37	5.1	0.5995
23			$3.0\cdot10^{-6}$	33.56	36.78	9.27	0.7034
24			$4.2\cdot10^{-6}$	35.14	37.21	13.7	0.8318
25		1200	$1.8\cdot10^{-6}$	31.29	37.21	5.1	0.5804
26			$3.0\cdot10^{-6}$	33.6	37.21	9.27	0.7059
27			$4.2\cdot10^{-6}$	34.25	36.63	13.29	0.811

The results of mathematical modeling of dynamic processes within the pressure p_n in the pumping hydraulic line and rotational velocity $d\varphi/dt$ of the hydraulic motor shaft during experiment no 4 are shown graphically in Fig. 5.

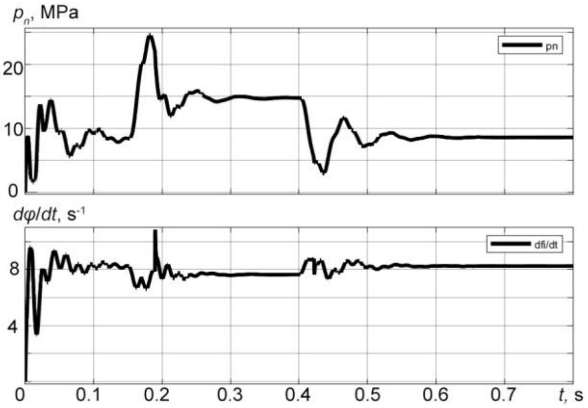

Fig. 5. Dynamic processes after selecting rational control device parameters

The obtained rational control device parameters, which ensure optimal dynamic processes in the hydraulic drive, allowed for its construction (see Fig. 6).

This design fully corresponds to its functional purpose in controlling the hydraulic drive of the belt conveyor under variable operating conditions with overload. The designed control device has a mass of 6.29 kg. The design of the control device takes into account the possibility of adjusting the initial spring compression; replacing valve elements of various designs; installing a reed switch and a permanent magnet with the ability to adjust their positioning.

The obtained three-dimensional model of the control device also allows for studying the fluid flows through its channels and conducting further optimization to reduce local pressure losses.

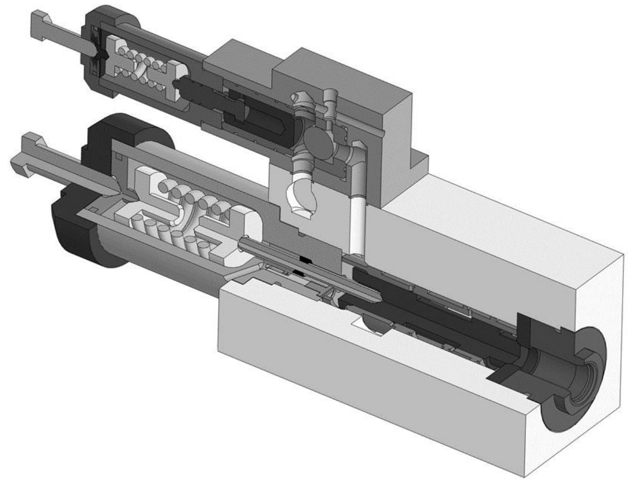

Fig. 6. The three-dimensional model of the control device was developed based on the research results

4. Conclusions

The goal of developing a control device by selecting its parameters during the optimization of the dynamic processes of the belt conveyor hydraulic drive has been achieved. The results and methods of scientific research on hydraulic drives of conveyors with various technological purposes have been analyzed, and the main directions for solving the set tasks have been identified. The fundamental scheme of the belt conveyor hydraulic drive with a control device has been developed. Based on the mathematical model of the belt conveyor hydraulic drive, an algorithm for calculating nine nonlinear differential equations described in Cauchy form has been developed. The block diagram of the mathematical model has been devised in the MATLAB Simulink environment. Solutions to the equations of the mathematical model have been found using the modified Rosenbrock's method with an accuracy of 0.1% in overload mode ranging from 5 kN·m to 11 kN·m. The comprehensive criterion for optimizing the static and dynamic characteristics of the belt conveyor hydraulic drive in overload mode using the LP method for selecting rational control device parameters has been formulated.

The significant influence of k_1 the stiffness of the adjustable valve spring, b_1 the coefficient of viscous friction of the valve against the housing, and f_d the area of the throttle valve's working window, on the dynamic processes of the belt conveyor hydraulic drive has been established through the investigation of the mathematical model. Twenty-seven experiments have been conducted, from which the values of σ_{oni} and σ_{offi}, the pressure p_n overshoot in the pressure line during the activation and deactivation of the control device, as well as δ_i, the velocity stabilization error of the motor shafts rotation $d\varphi/dt$, have been calculated. Optimal static and dynamic characteristic values have been found: $\sigma_{on4} = 7.46\%$, $\sigma_{off4} = 35.09\%$, $\delta_4 = 7.84\%$. These correspond to the optimization criterion $W_{opt} = 0.4438$, for experiment No 4 and the combination of selected rational parameters $k_1 = 30\cdot10^3$ N/m; $b_1 = 800$ N·s/m; $f_d = 1.8\cdot10^{-6}$ m^2. A three-dimensional model of the control device has been designed using SolidWorks software for the belt conveyor hydraulic drive with rational parameters. The obtained three-dimensional model will allow conducting simulation studies of the flow of the working fluid through the control device channels, reducing local losses in the hydraulic drive. The research will be beneficial for engineers and scientists involved in the development and improvement of belt conveyor hydraulic drives, as well as other technological devices operating under variable load conditions.

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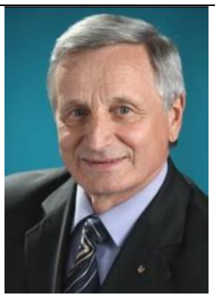

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