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Dynamics of Adaptive Drive of Mobile Machine Belt Conveyor

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1. Introduction

Conveyors utilization rate at mining enterprises on the average is 50…70% of power rate and 60…70% of operation time rate. Such inefficient usage of the conveyors is connected with non-uniformity of freight traffic [1, 2] by load amplitude and interruptions in the influx of goods, etc.

Conveyors of agricultural mobile machinery also take loads, intensity of which differs greatly at various phases of technological cycle. Unloading conveyors and cross conveyors of root-cutting machine PKM-6 in the process of changing the auto transport, operating with the combine harvester stop and beets are accumulated in the root hopper. During the next start of the conveyor drive the technological loading on conveyor increases 2,5…3 times as compared with the nominal loading.

Similar operation modes are typical for receiving conveyor of K-65M2Б3-K type clamp former during root plants unloading into the hopper from motor transport. In such case, driving elements of electric mechanical drive of the conveyors may break down, and in hydraulically – powered drive [3] emergence disconnection due to relief valve operation could lead to shutdown of hydraulic motor. For further manual restart of the conveyor the load on its working element is reduced and restart of the drive is performed.

Greater part of the research [4-7] is aimed at coordination of operation modes of belt conveyors drives with freight traffic parameters by means of belt speed control.

For conveyors, operating in conditions of variable loads, the necessity arises to regulate torque on drive drum. To provide the continuous operation of conveyor drive, subjected to short-term or long-term overloads and, as a result, increase the machine performance, it is expedient to equip the drive with the devices for the control of switch on and switch off the additional hydraulic motor. This motor is installed in parallel to main motor, this allows to apply active back-up of the motor torque on drive drum. Papers [8-13] consider the solution of the given problem. For efficient application in belt conveyors, these drives require both engineering improvement and comprehensive studies of the impact of mechanical system parameters on its operation processes. That's why, thorough analysis of dynamic processes, taking place in adaptive hydraulic drive of mobile machine conveyor is relevant.

2. Mathematical modeling of dynamic processes, taking place in conveyor adaptive drive

Adaptive hydraulic drive, intended for operation of belt conveyor of the mobile machine (for instance, clamp former) in conditions of variable loads, is developed. This drive enables to enlarge the functional possibilities of the conveyor, improve its reliability, efficiency, performance and decrease energy consumption.

The operation of control system for adaptive hydraulic drive of belt conveyor is reduced to the following: when the load on transporting element reaches certain value, the control system switches on the additional hydraulic motor [14, 15]. Main and additional hydraulic motors, transmission mechanisms and control system are located in drive drum housing, enabling to obtain compact construction of the drive with high specific power [16].

Computational scheme for the investigation of mutual impact of adaptive hydraulic drive parameters and transposing element of belt conveyor is developed (Fig. 1).

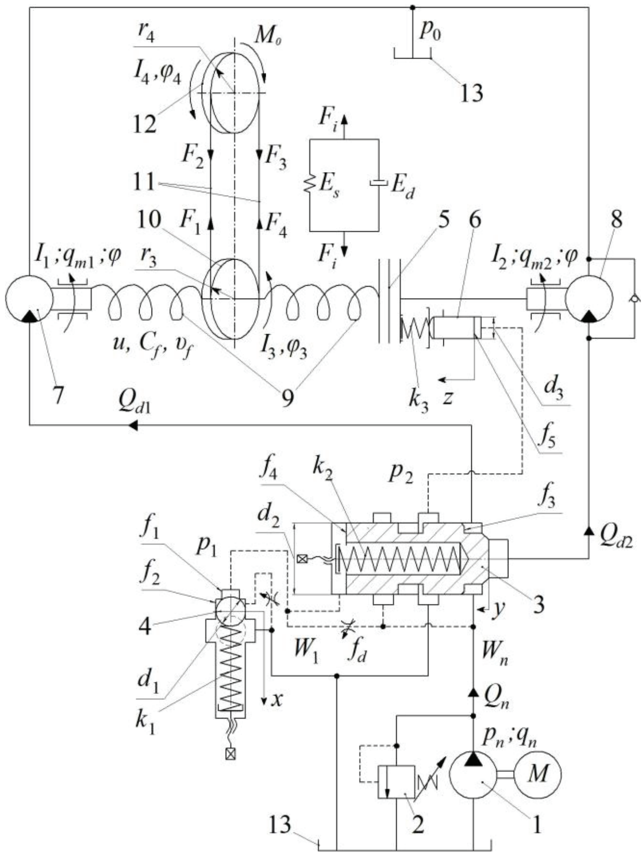

Fig. 1 Computational scheme of adaptive hydraulic drive of belt conveyor

Computational scheme contains: constant-flow pump 1 with overload relief valve 2, distribution valve 3, sensor 4, friction clutch 5 with plunger 6, main 7 and additional 8 hydraulic motors, transmission mechanism 9, drive drum 10, belt 11 and tail drum 12. Drain and injection of working fluid is performed from the hydraulic tank 13.

Mathematical model of adaptive hydraulic drive of belt conveyor is developed for carrying out the research. Taking into consideration small length of the transporting element, the conveyor belt is presented by Voigt rheological model with spring-dissipative connections. Equations of mathematical model are constructed, applying the assumptions, common in hydraulic drive.

Motion equation of drive and transporting part of belt conveyor after corresponding transformations will have the form:

$$\left.\begin{aligned}
&(I_1+I_2)\frac{d^2\varphi}{dt^2}=q_{m1}p_n+q_{m2}p_n-C_f\left(\varphi-u\varphi_3\right)-\\
&-\nu_f\left(\frac{d\varphi}{dt}-\frac{d\varphi_3}{dt}\right);\\
&I_3\frac{d^2\varphi_3}{dt^2}=uC_f\left(\varphi-u\varphi_3\right)+\nu_f\left(\frac{d\varphi}{dt}-u\frac{d\varphi_3}{dt}\right)-\\
&-2C_s\left(\varphi_3 r_3-\varphi_4 r_4\right)r_3-2\nu_s\left(\frac{d\varphi_3}{dt}r_3-\frac{d\varphi_4}{dt}r_4\right);\\
&I_4\frac{d^2\varphi_4}{dt^2}=2C_s\left(\varphi_3 r_3-\varphi_4 r_4\right)r_4+\\
&+2\nu_s\left(\frac{d\varphi_3}{dt}r_3-\frac{d\varphi_4}{dt}r_4\right)-M_0,
\end{aligned}\right\} \tag{1}$$

where $C_s=\frac{A\,E_s}{L}$ – rigidity of the belt; $\nu_s=\frac{A\,E_d}{L}$ – its dynamic viscosity.

In the first equation components I_2 and $q_{m2}p_n$ are taken into account, when additional hydraulic motor 8 is switched on after the actuation of friction clutch 5.

Mathematical model of control system is constructed on d'Alembert principle, regarding the forces, acting on its moving elements and balance of working fluids consumption.

Opening of sensor ball 4 occurs on condition of pressure increase in hydraulic drive to the value:

$$p_1 \ge \frac{k_1 x}{f_1}. \tag{2}$$

Closing of sensor ball 4 occurs on condition of pressure drop to the value:

$$p_1' \le p_1\frac{f_1}{f_2}+\frac{k_1 x}{f_2}. \tag{3}$$

Equations of balance of forces, acting on distribution valve 3, sensor 4 and plunger 6 are:

$$m_1\frac{d^2x}{dt^2}=f_2 p_1-k_1(x_0+x)-b_1\frac{dx}{dt}-F_{g1}; \tag{4}$$

$$m_2\frac{d^2y}{dt^2}=p_n f_3-k_2(y_0+y)-b_2\frac{dy}{dt}-p_1 f_4-F_{g2}; \tag{5}$$

$$m_3\frac{d^2z}{dt2}=p_2 f_5-k_3(z_0+z)-b_3\frac{dz}{dt}-F_a; \tag{6}$$

Equations of flow continuity condition for pressure mains of hydraulic drive and hydraulic lines of sensor 4 and plunger 6 are:

$$\begin{aligned}
&\beta W_n\frac{dp_n}{dt}=q_n\cdot n_n-\left(q_{m1}+q_{m2}\right)\cdot\frac{d\varphi}{dt}-\\
&-\mu\cdot f_d\sqrt{\frac{2|p_n-p_1|}{\rho}}\cdot sgn(p_n-p_1)-\\
&-\mu\cdot\pi\cdot d_2\cdot y\sqrt{\frac{2|p_n-p_2|}{\rho}}\cdot sgn(p_n-p_2);
\end{aligned} \tag{7}$$

$$\begin{aligned}
&\beta W_1\frac{dp_1}{dt}=\mu\cdot f_d\sqrt{\frac{2|p_n-p_1|}{\rho}}\cdot sgn(p_n-p_1)-\\
&-\mu\cdot\pi\cdot d_1\cdot x\sqrt{\frac{2|p_1|}{\rho}};
\end{aligned} \tag{8}$$

$$\begin{aligned}
&\beta W_2\frac{dp_2}{dt}=\mu\cdot\pi\cdot d_2\cdot y\sqrt{\frac{2|p_n-p_2|}{\rho}}\times\\
&\times sgn(p_n-p_2)-f_5\cdot\frac{dz}{dt}.
\end{aligned} \tag{9}$$

Supply of working fluid to additional hydraulic motor 8 with pressure space q_{m2} occurs after opening of distribution valve 3.

In equations (1-9) such designations are used: M_0 – moment of forces of useful resistance; p_n – pressure in pressure mains of hydraulic drive; p_2 – pressure in plunger cavity 6; m_1, m_2, m_3 – masses of sensor ball 4, distribution valve 3 and plunger 6, correspondently; x, y, z – coordinates of corresponding masses motion; k_1, k_2, k_3, x_0, y_0, z_0 – rigidities and initial deformations of springs, correspondingly; q_{m1}, q_{m2} – pressure space of hydraulic motors 7 and 8, correspondently; β – compliance coefficient with the account of working fluid compressibility; μ – cost ratio; ρ – density of working fluid; f_1, f_2, f_3, f_4, f_5 – areas of fluid contact with sensor 4 in open and close position, as well as areas of end surfaces of distribution valve 3 and plunger 6, correspondently; W_n, W_1, W_2 – volume of pressure pipe of hydraulic drive , sensor cavities 4 and plunger 6, correspondently; d_1, d_2, d_3 – diameters of sensor ball 4, distribution valve 3 and plunger 6, correspondently; b_1, b_2, b_3 – coefficients of viscous damping for hydraulic drive; F_{g1}, F_{g2} – hydrodynamic forces, acting on sensor ball 4 and distribution valve 3 [14]; F_a – reaction of half-couplings connection, I_1, I_2, I_3, I_4 – inertia moments of main 7, additional 8 hydraulic motors, drive 10 and tail 12 drums, correspondently; φ_1, φ_2, φ_3, φ_4 – turning angles of main 7, additional 8 hydraulic motors, drive 10 and tail 12 drums, correspondently; r_3, r_4 – diameters of drive 10 and tail 12 drums; C_s – torsion rigidity of transmission mechanism 9; C_f – belt rigidity 11 of the conveyor; ν_s – viscosity of the transmission mechanism 9; ν_f – dynamic viscosity of the belt 11; u – reduction ratio of transmission mechanism 9; A – belt cross-section area; L – belt length; E_s – belt modulus of elasticity; E_d – belt dynamic modulus of elasticity.

3. Results of theoretical study

Solution of the system of non-linear differential equations was performed by means of computer software package MATLAB Simulink, applying Rosenbrock pattern search method. Each equation of the system is presented in

Cauchy form for the formation of block-diagram of mathematical model.

Determination of dynamic characteristics of the conveyor mechanical system was performed on the base of pressure p_n transient processes in pressure mains of the hydraulic drive (Fig. 2).

Perturbing factor in the system is the change of load moment, that is reduced to tail drum, in case of normal operation mode to overload mode and vice versa. Overload mode was reached in the process of increasing the moment of forces of useful resistance from 5000 to 12000 Nm. It is characterized by three phases: the first phase – switching on of the additional hydraulic motor; the second phase – operation of drive with two hydraulic motors; the third phase – load decrease and switching off of additional hydraulic motor. For each of the above-mentioned phases of transient processes dynamic characteristics were analyzed. Studies were carried out in such ranges of drive and transporting parts of the conveyor parameters change: static elasticity modulus of conveyor belt E_s=(210…1010)·10^6 MPa, dynamic elasticity modulus E_d=988…4750 MPa, belt cross-section area A=(0,824…3,96)·10^{-2} m, conveyor length L=11...42,4 m, integrated inertia moment of tail drum I_4=5...9 kg·m^2, torsion rigidity of transmission mechanism C_f=(0,25…0,75)·10^6 MPa, damping of transmission mechanism v_f=150...600 Ns/m.

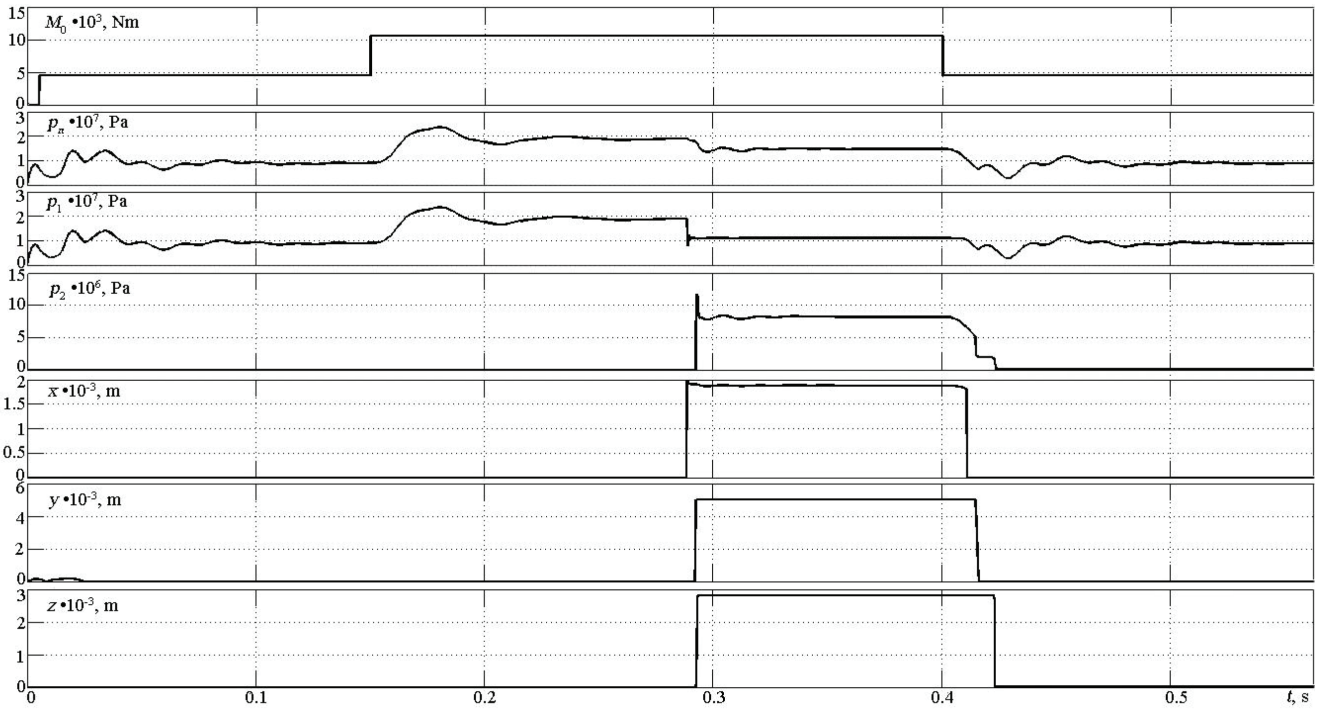

Fig. 2 Theoretical graphs of transient processes of $M_0(t)$, $p_n(t)$, $p_1(t)$, $p_2(t)$, $x(t)$, $y(t)$, $z(t)$ change

The impact of adaptive hydraulic drive and transporting part of the conveyor parameters on the time of transient process t_p and overregulation value Δ for pressure p_n in pressure mains was studied.

Fig. 3 and 4 show the dependence of transient process time and overregulation value for the first phase of the overload in the process of drive and transport part of the conveyor parameters change in the given ranges. Parameters of drive and transporting parts of the conveyor for the preset values of parameters range, at which transient process time is within the limits of 0,11s < t_{p1} < 0,15s and pressure overregulation value – 15% < Δ_1 < 45%, are calculated. To increase transient process rate t_{p1} it is necessary to decrease the values of belt length parameters L and torsion rigidity of transmission mechanism C_f and increase static E_s and dynamic E_d elasticity moduli. To reduce the value of overregulation Δ_1 it is expedient to decrease the parameters of static E_s and dynamic E_d belt elasticity and transmission mechanism rigidity C_f moduli.

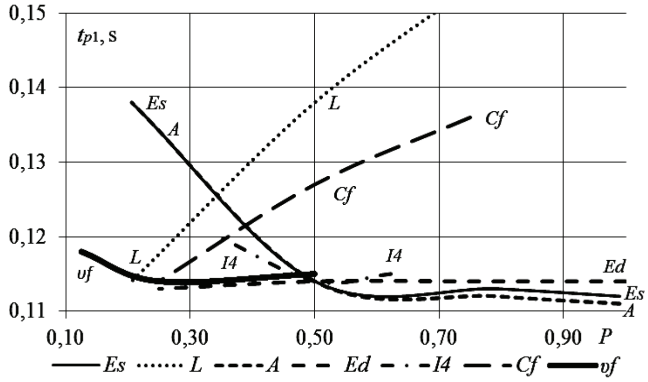

Fig. 3 Impact of the parameters of drive and transporting parts of the conveyor on transient process time for the first overload phase

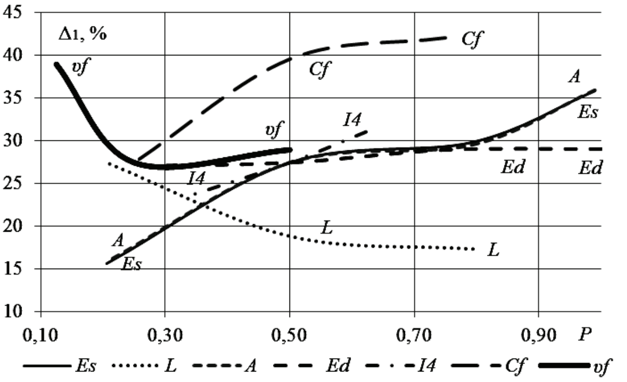

Fig. 4 Impact of the parameters of drive and transporting parts of the conveyor on the value of overregulation for the first overload phase

Fig. 5 and 6 show the dependences of dynamic characteristics, that corresponds to the second phase of the overload. This phase is characterized by minor changes of dynamic characteristic values in the range of $0{,}05s < t_{p2} < 0{,}12s$ and $4\% < \Delta_2 < 12\%$. To increase the rate of transient processes t_{p2} and decrease the value of overregulation Δ_2 it is necessary to decrease integrated inertia moments of the drums I_4 and torsion rigidity of transmission mechanism C_f and increase damping of transmission mechanism υ_f. Impact of other parameters is not substantial. Additionally, the decrease of overregulation value Δ_2 could be influenced, increasing the parameters of static E_s and dynamic E_d elasticity moduli.

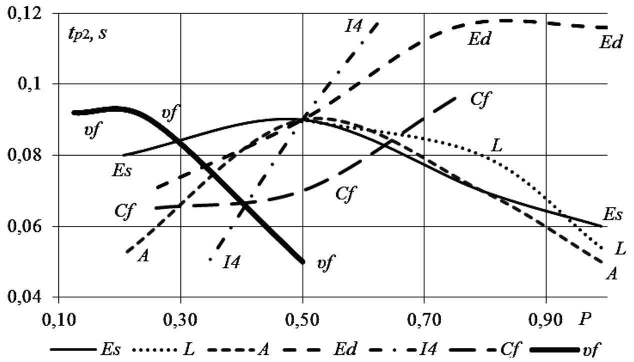

Fig. 5 Impact of drive and transporting parts of the conveyor parameters on the time of transient process for the second phase of overload

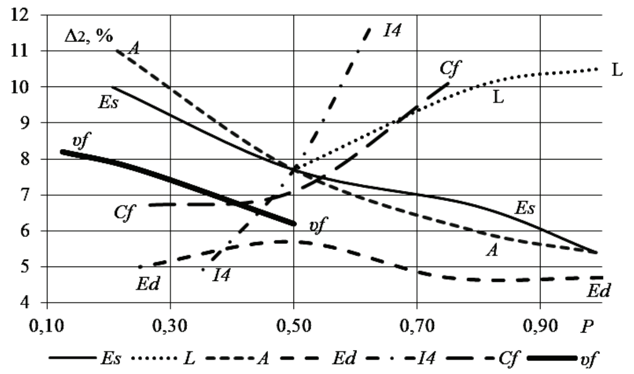

Fig. 6 Impact of drive and transporting parts of the conveyor parameters on the value of overregulation for the second phase of overload

Change of dynamic characteristics of the third phase of transient processes in the hydraulic drive is shown in Fig. 7 and 8. Parameters of drive and transporting parts of the conveyor, at which after unloading the time of transient process changes in the range of $0{,}9s < t_{p3} < 0{,}17s$ and the value of overregulation – $20\% < \Delta_3 < 55\%$ are determined. To increase the rate of transient processes t_{p3} of adaptive hydraulic drive of conveyor at decreasing the loading to nominal it is necessary to increase the parameters: damping of transmission mechanism υ_f, static E_s of elasticity modulus and belt cross-section area A; decrease parameters: inertia moments of the drums I_4, torsion rigidity of transmission mechanism C_f. Main parameters of drive and transporting parts of the conveyor, that decrease the value of overregulation Δ_3 are smaller values of transmission mechanism rigidity C_f and inertia moments of drums I_4. Increase of transmission mechanism damping υ_f leads to decrease of overregulation value Δ_3.

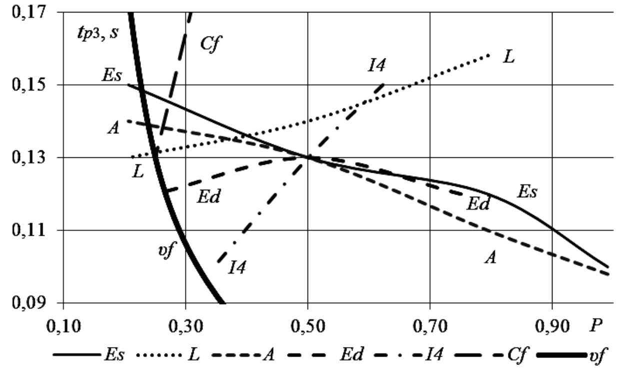

Fig. 7 Impact of drive and transporting parts of the conveyor parameters on the time of transient process for the third phase of overload

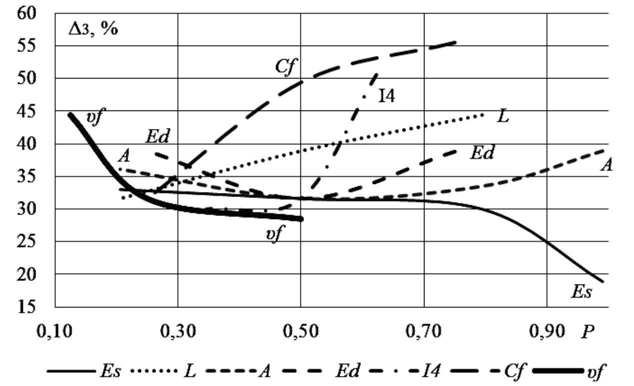

Fig. 8 Impact of drive and transporting parts of the conveyor parameters on the value of overregulation for the third phase of overload

4. Conclusions

1. Mathematical model of dynamic processes of the developed adaptive hydraulic drive with parallel installed hydraulic motors is constructed, the given model enables to perform the selection of parameters of hydraulic and transporting parts of the conveyor, that provide rational operation modes of the conveyor at variable loads on the transporting organ.

2. On the base of theoretical research, carried out,

recommendations regarding the design of adaptive hydraulic drive with transporting part of belt conveyor of mobile machine-clamp former К-65М2Б-К are worked out.

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DYNAMICS OF ADAPTIVE DRIVE OF MOBILE MACHINE BELT CONVEYOR

S u m m a r y

Dynamic processes occurring in hydraulic drive of belt conveyor with the control system of parallel installed hydraulic motors are studied by means of mathematical model, constructed with the account of physical phenomena, taking place in the process of hydraulic system operation at variable load on operation organ.

The parameters of reception conveyer of clamp former hydraulic drive control system have been theoretically substantiated, that provide continuous operation of hydraulically-actuated system in conditions of short-term or long-term overloads. The impacts of control system parameters on transient processes in hydraulic drive are analyzed, recommendations, regarding the increase of such drive operation efficiency are suggested.

Keywords: hydraulic drive, belt conveyor, mathematical model.