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Study of the dynamic stability of the belt conveyor adaptive drive

***Abstract**. Dynamic stability of the adaptive hydraulic drive of the belt conveyor is studied with the help of an improved mathematical model constructed taking into account the physical processes that occur during the mechanical system of the conveyor under load changes. The parameters of the adaptive drive, which provide stable operation of the mechanical system in conditions of its overload, are determined.*

***Streszczenie.** Dynamiczna stabilność adaptacyjnego napędu hydraulicznego przenośnika taśmowego jest badana za pomocą ulepszonego modelu matematycznego skonstruowanego z uwzględnieniem fizycznych procesów zachodzących podczas mechanicznego systemu przenośnika pod zmianą obciążenia. Określono parametry napędu adaptacyjnego, zapewniające stabilną pracę układu mechanicznego w warunkach jego przeciążenia.. (**Badanie stabilności dynamicznej napędu adaptacyjnego przenośnika taśmowego**).*

Keywords: adaptive hydraulic drive, control system, conveyor, overload, dynamic processes, mathematical model.
Słowa kluczowe: adaptacyjny napęd hydrauliczny, układ sterowania, przenośnik, procesy dynamiczne, model matematyczny.

Introduction

Goods flows, arriving on the belts of the conveyors of various technological applications are characterized by considerable irregularity both regarding the amplitude of loading and goods supply intervals. Irregularity ratio of goods flows can vary within the range of Kn=1,97...2,02 [1]. For this reason the utilization factor of the conveyors, for instance, at mining enterprises, is on average, 50...70% by power and 60...70% by operation duration. The drives of agricultural mobile machines accept loads, the intensity of which differs greatly at different phases of the technological cycle [1]. Transverse conveyors and conveyor unloader of root-cutting machine in the process of the motor transport change, operating with the harvester, stop and beets are accumulated in the tank of the harvester. After the restarting of the conveyors drives the technological loading on the conveyors increases 2,5...3 times as compared with the nominal loading. Similar operation modes are typical for the receiving conveyor of clamp former in the process of root crops unloading into the hopper from the motor transport. In this case, the failure of drive's elements in electro mechanic drive of the conveyor, and in hydraulic drive [2] – emergency outage as a result of safety valve operation and the shutdown of the hydraulic motor is possible. For further resumption of the conveyor operation the load at its working unit is manually decreased and the restarting of the drive is performed.

Such operation modes of the conveyor cause unjustified electric energy consumption, wear of the belt, idlers, increase of the idle run of the belt, downtime of the equipment. That is why, numerous studies [3–5] are aimed at the provision of the coordination of the operation parameters of belt conveyors drive with variable modes of goods flows. One of the directions of the research is the study of drives operation with the regulators of belt motion speed at periodic arrival of the load, another direction – is automatic adaptation of its parameters at considerable change of loads to avoid non technological outages of the conveyor.

In order to provide continuous operation of the conveyor drive, subjected to short-term or long-term overloads, and increase the output of the machine, it is expedient to equip the drive with the additional motor, installed in parallel to the main motor, this will allow to apply active reservation of the torque at drive drum. At the same time the drive must be equipped with the device or the system of control, sensitive to load change at working organ that will enable to adapt instantly its parameters to variable operation modes. The solutions of such problems are considered in [6–10]. For the implementation of such drives in the conveyors, operating with variable loads the study of such mechanical systems stability is obligatory. Thus, the aim of the research is the substantiation of the parameters of adaptive hydraulic drive, equipped with main and additional hydraulic motors and control system, providing stable continuous operation modes of the conveyors drive under the condition of its overload. The following problems are solved to reach the aim of the research:

1. mathematical model of the adaptive drive dynamic processes was improved, taking into account elastic-inertia characteristics of the transporting part of the conveyor;
2. the analysis of the impact of control system parameters and hydraulic built-in drive on the stability of the investigated hydraulic mechanical system was performed, the parameters of the adaptive drive providing the absence of continuous mechanical oscillations excitation were substantiated.

Analysis of the methods of variants moving systems stability study

Machines and mechanisms, particularly those, equipped with the systems of automatic control, at certain parameters can operate in non-stable operation modes. That is why, stability provision is one of the main problems to be solved in the process of the development of the devices with the systems of automatic control [11, 27,28]. Especially, it concerns non-stationary operation modes when saving of equilibrium state or the preset motion law by the mechanical system is very important.

Mathematical systems, constructed for the study of the dynamic state of the mechanical devices with the systems of automatic control, are described by differential equations of higher orders. Study of mathematical models, described by the equation of higher than the fourth order, is often accompanied by certain difficulties in the process of their solution. Main difficulties arise in the process of parameters selection for regulation of oscillating operation mode. That is why, first of all, there appears the necessity to determine the character of the transient process and solve the problem of adaptive drive stability.

For the study of the stability of non-disturbed and disturbed motions of the system, state of which is described by the system of differential equations method of Lyapunov is often used. For the offshore winch with PID-control on the base of hydraulic system the mathematical model of dynamic processes is built [12], its dynamic stability in non-stationary operation modes is investigated.

Long lasting behavior of 1D hydraulic open channel [13], often used in hydraulic engineering to characterize the dynamics of non-stationary flow in river channel for the case, when the oscillations of the flow are not taken into account, is analyzed. Rational parameters, providing system stability with the feedback are determined and it is proved by numerical modeling.

For the system of the mixed -spectrum supercritical water reactor core the equation of flow dynamics disturbances is built and 1D operation stability is studied on the base of the analysis of mathematical model frequency characteristics [14].

In [15] non-linear mathematical model of dynamic processes in hydraulic mechanical servomechanism, performing the control of aircraft systems, is developed. The impact of the rigidity of mounting structure on the transient processes is established, the analysis of the stability is performed applying Lyapunov-Malkin theorems. Stability graphs for control system by Routh-Hurwitz criterion are built. The ways of stability provision of control system operation are shown, using positive impact of structural feedback.

For the mechanical systems, described by the equations higher than the fourth order, the most widely-used stability assessment criterion is Nyquist-Mikhailov criterion, according to this criterion transient function is written and amplitude-phase characteristics of control system are studied [16-18].

The above-mentioned methods of stability study are rather complicated and labour consuming for studying higher than quantic systems [30, 31,32].

Modern MATLAB programming products, namely Simulink, allow to study the stability of drive operation during mathematical model adjustment. Calculation of transient processes is performed during minutes and by the results of control system parameters selection, for instance, of the hydraulic drive it is easy to determine parameters, that provide attenuating oscillation processes.

In [19] non-linear mathematical model of operation process of hydro-turbine governing system with sloping ceiling tailrace tunnel under load disturbance is developed. Graphs of transient processes, phase space paths and stability diagrams of the system on the angle of slope of the ceiling tunnel and its sectional form are obtained.

Dynamic stability of rotor rotation of large hydraulic turbine is analyzed on the base of amplitude-time characteristics, obtained for non-linear mathematical model [20]. The impact of labyrinth seal parameters on oscillating processes in the turbine is determined.

The results of theoretical studies regarding the stability of transient processes in the hydraulic system of rod cylinder, expressed in [21, 29, 32], are proved by the experimental studies.

Experimental studies of the hydraulic stability of the developed testing stands [22, 23, 27,29] allow to avoid tedious calculations and composing of mathematical models. However, such methods are rather costly and require certified equipment.

Thus, study of dynamic processes stability in the machines and mechanisms with control system, described by the systems of differential equations of higher orders is expedient of perform, applying program packages, for instance, MATLAB Simulink.

Materials and methods of stability study of the conveyor adaptive drive

Construction of the calculation scheme of the conveyor adaptive drive was performed in two stages. First, the calculation scheme of the control system of the conveyor hydraulic drive (Fig. 1) was developed for the investigation of dynamic characteristics of its operation [24, 31,32]. Then its improvement was carried out at the expense of jointing the calculation model of the transporting part of the belt conveyor, that takes into account elastic-inertia characteristics of small length belt conveyor (Fig. 2).

Calculation schemes in Fig. 1 and 2 comprise the constant delivery pump 1with the safety valve 2, control system with distribution valve 3, sensor 4, friction coupler 5, plunger 6 and the drive with main 7 (HM1) and additional 8 (HM2) hydraulic motor, transmission mechanism 9, drive drum 10 and also the belt 11 and tail drum 12. Drain and delivery of working fluid is carried out from hydraulic reservoir 13.

The succession of the conveyor hydraulic drive control system elements operation is shown in the sequence diagram of its work (Fig. 3) [25]. Operation cycle of the conveyor hydraulic drive control system during load change can be conventionally divided into such stages:

1) pressure p_n change in hydraulic supply line of the hydraulic drive control system (Graph 1);

2) displacement of ball sensor 4 over the run h_c (Graph 2);

3) displacement of the distribution valve 3 over the value h (Graph 3);

4) displacement of the press plunger 6 of the half coupling for the connection of the shaft of additional 8 hydraulic motor (HM2) with the shaft of drive drum 10 (Graph 4).

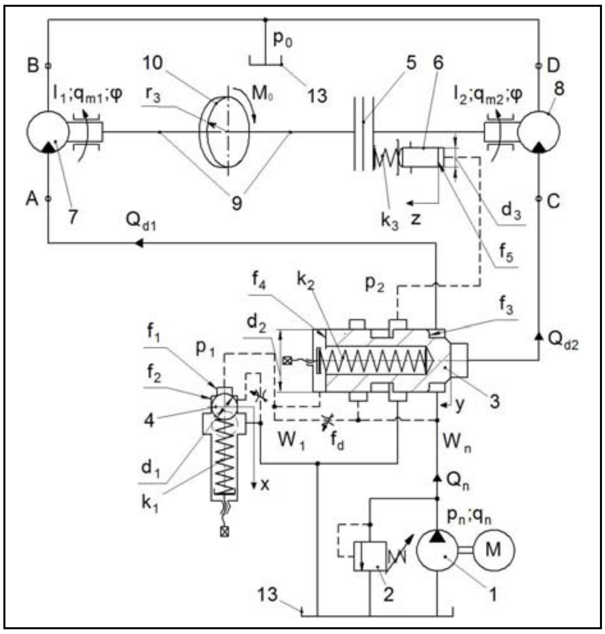


Fig.1. Calculation scheme of the conveyor hydraulic drive without taking into account elastic-inertia characteristics of its transporting part

The sequence diagram shows the phases of ball sensor 4 direct route travel during the overload and serial operation of distribution valve 3 with press plunger 6 of the half-coupling for turning on the additional 8 hydraulic motor (HM2). During load reduction to the preset value and, correspondingly, the pressure p_n, in the hydraulic supply line reverse motion of ball sensor 4 takes place. At the same time successive switching of the distribution valve 3 into the initial state and return of the press plunger 6 of the half-coupling that disconnects the shaft of the additional 8 hydraulic motor (HM2) from the drive shaft 10 takes place.

The suggested sequence diagram of the conveyor hydraulic drive control system operation in case of load change is idealized and does not take into account its transient processes.

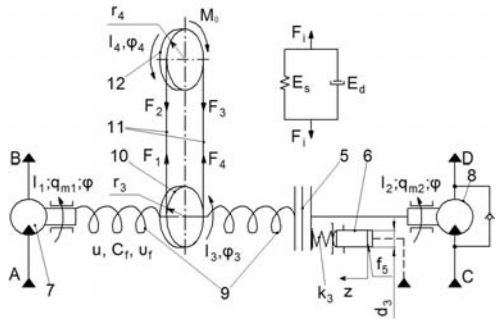


Fig.2. Calculation scheme of the transporting prat of belt conveyor drive

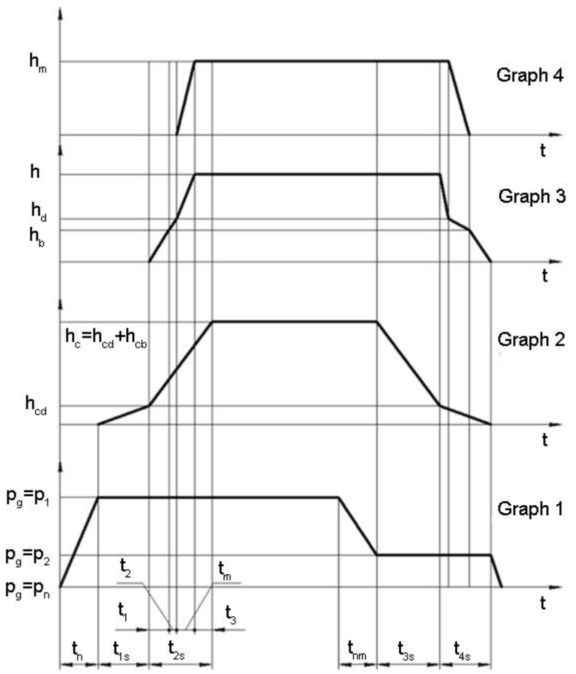


Fig.3. Sequence diagram of the conveyor adaptive hydraulic drive control system

Mathematical model for the calculation of transient processes in the control system of the conveyor adaptive drive is constructed this model takes into consideration the elastic-inertial characteristics of its transporting part.

In the process of model development the generally accepted assumptions [24] were used, their correctness was proved by the comparison of theoretical results and results of experimental studies of the similar systems [6, 33].

Mathematical model of the adaptive drive with the conveyor control system comprises:

– equilibrium equation of the forces, acting on the sensor 4, distribution valve 3 and plunger 6:

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

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

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

– equation of the flows continuity condition for hydraulic supply line and hydraulic lines of the sensor 4 and plunger 6:

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

(5) $$\beta\cdot W_1\frac{dp_1}{dt}=\mu\cdot f_d\sqrt{\frac{2\left|p_n-p_1\right|}{\rho}}\operatorname{sgn}(p_n-p_1)-\mu\cdot\pi\cdot d_1\cdot x\sqrt{\frac{2p_1}{\rho}};$$

(6) $$\beta\cdot W_2\frac{dp_2}{dt}=\mu\cdot\pi\cdot d_2\cdot y\sqrt{\frac{2\left|p_n-p_2\right|}{\rho}}\operatorname{sgn}(p_n-p_2)-f_5\frac{dz}{dt};$$

– equations of moments equilibrium on the shafts of driving and transporting parts of the belt conveyor, after corresponding transformation they have the following forms:

(7) $$\left(I_1+I_2\right)\frac{d^2\varphi}{dt^2}=q_{m1}\cdot p_n+q_{m2}\cdot p_n-C_f\left(\varphi-u\varphi_3\right)-\nu_f\left(\frac{d\varphi}{dt}-\frac{d\varphi_3}{dt}\right);;$$

(8) $$\begin{aligned}I_3\frac{d^2\varphi_3}{dt^2}&=u\cdot C_f\left(\varphi-u\varphi_3\right)+\nu_f\left(\frac{d\varphi}{dt}-u\frac{d\varphi_3}{dt}\right)-\\&-2\cdot C_s\left(\varphi_3\cdot r_3-\varphi_4\cdot r_4\right)r_3-2\cdot\nu_s\left(\frac{d\varphi_3}{dt}r_3-\frac{d\varphi_4}{dt}r_4\right)\end{aligned};$$

(9) $$I_4\frac{d^2\varphi_4}{dt^2}=2\cdot C_s\left(\varphi_3 r_3-\varphi_4 r_4\right)r_4+2\cdot\nu_s\left(\frac{d\varphi_3}{dt}r_3-\frac{d\varphi_4}{dt}r_4\right)-M_0.$$

In equations (1 – 9) such designations are used: M_0 – moment of forces of useful resistance; p_n – pressure in hydraulic supply line of the hydraulic drive; p_1, p_2 – are pressures of sensor 4 opening and in the cavity of plunger 6, correspondingly; m_1, m_2, m_3 – are ball masses of sensor 4, distribution valve 4 and plunger 6, correspondingly; x, y, z – are the coordinates of corresponding masses displacement; k_1, k_2, k_3, x_0, y_0, z_0 – are rigidities and initial deformation of the springs, correspondingly; q_n ,n_n – is characteristic volume of the pump 1 and rotation frequency of its shaft, correspondingly; q_{m1}, q_{m2} – is characteristic volume of the hydraulic motor 7 and 8, correspondingly; β – is the compliance coefficient with the account of work fluid compressibility; μ – is the discharge coefficient; ρ – is the density of working fluid; f_1, f_2, f_3, f_4, f_5 – are the contact areas of the fluid with the sensor 4 in the open and close positions and areas in distribution valve 3 ends surfaces and plunger 6, correspondingly; W_n, W_1, W_2 – is the volume of pressure pipe of hydraulic drive , sensor 4 and plunger 6, correspondently; d_1, d_2, d_3 – is the diameter of sensor 4 ball, distribution valve 3 and plunger 6, correspondingly; b_1, b_2, b_3 – are the viscous damping coefficients for hydraulic drive; F_{g1}, F_{g2} – are hydro dynamic forces, acting on the ball of the sensor 4 and distribution valve 3 [24]; Fa – is the reaction of half-couplings clutching, I_1, I_2, I_3, I_4 – are the inertia moments of the main 7, additional 8 hydraulic motors, drive 10 and tail 12 drums, correspondingly; φ_1, φ_2, φ_3, φ_4 – are turning angles of the main 7, additional 8 hydraulic motors, drive 10 and tail 12 drums, correspondingly; r_3, r_4 – are the diameters of drive and tail 12 drums; C_f – is the torsional stiffness of the communicator 9; ν_f – is the damping of the communicator 9; C_s – is the rigidity of the conveyor belt 11; ν_s– is the dynamic viscosity of the belt 11; u – is the transfer ratio of the communicator 9; A_s – is the area of the belt cross-section; L_s – is the length of the belt; E_s – is the static modulus of elasticity of the conveyor belt; E_d – is the dynamic modulus of elasticity of the belt.

Calculation were carried out, using such initial values of control system parameter: M up to 12000Nm; M_0=1200Nm; $E_s=510\cdot10^6$MPa; E_d =2400MPa; $A_s=2\cdot10^{-2}m^2$; L_s=11m; $C_f=0{,}25\cdot10^6$Pa; ν_f =300Ns/m; $q_n=15{,}73\cdot10^{-6}m^3/rad$;

q_{m1}=200,32·10^{-6}m^3/rad; q_{m2}=50,96·10^{-6} m^3/rad; n_n=105 rad; β=0,6·10^{-9} m^2/N; μ=0,6; ρ=850kg/m^3; S=f_1/f_2=0,7; f_d=2,2·10^{-6}m^2; W_n=0,5·10^{-3}m^3; m_1=33·10^{-3}kg; d_1=7,94·10^{-3} m; x_0=10·10^{-3}m; k_1=50·10^{3}N/m; b_1=500kg/s; W_1=0,5·10^{-5} m^3; m_2=120·10^{-3} kg; d_2=16·10^{-3} m; y_0=0; k_2=9·10^3 N/m; b_2=50kg/s; W_2=0,1·10^{-3}m^3; F_{g1}=0; F_{g2}=0; m_3=120·10^{-3}kg; d_3=19·10^{-3}m; z_0=0; k_3=15·10^{3}N/m; b_3=300kg/s.

Results and discussion

Mathematical model, consisting of the system of non-linear differential equations of the fifteenth order, was solved by means of compiler programming package MATLAB Simulink, applying Rosenbrock pattern search of the second order. This method provides high rate of calculations as compared with Adams and Runge-Kutta methods, at the accuracy of calculations 0,001 that is sufficient for obtaining reliable results of the calculations. The calculation of one transient process lasted approximately 3 minutes.

Study of the dynamic characteristics of the mechanical system of the conveyor was performed on the base of transient processes by the pressure p_n in the hydraulic supply line of the hydraulic drive (Fig. 4) [26,34].

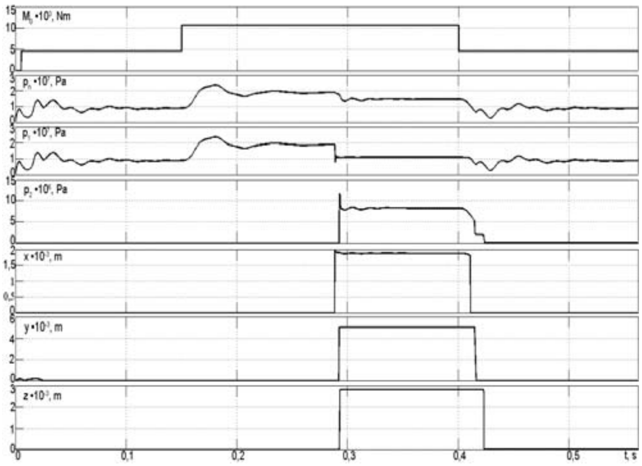


Fig. 4. Theoretical graphs of transient processes of $M_0(t)$, $p_n(t)$, $p_1(t)$, $p_2(t)$, $x(t)$, $y(t)$, $z(t)$ change

Sequence of $p_n(t)$, $x(t)$, $y(t)$, $z(t)$ change almost corresponds to the developed sequence diagram of belt conveyor hydraulic drive control system operation (see Fig. 3). The absence of the identity is explained by the appearance of the transient processes and their different duration.

Disturbing factor in the system is the change of load moment, which at the calculation scheme is reduced to tail dump, from the value that corresponds to normal operator mode to the value of overload mode and vice versa. Overload mode was achieved during the increase of the moment of forces of the useful resistance from 5000 to 12000Nm.

In the process of parametric adjustment of the mathematical model in order to find the best (optimal) parameters, at which the degree of the model and mechanical system of the conveyor [6] was minimal, the modes of non-stable operation were determined. That is why, there appeared the need in more detailed study of the dynamic stability of the conveyor adaptive drive operation.

Theoretical research showed that if the range of transporting part of parameters change is: E_s=(210…1010)·10^{6}MPa; E_d=988…4750MPa; A_s=(0,824…3,96)·10^{-2}m^2; L_s=11...42,4m; I_4=5...9kg·m^2; C_f=(0,25…0,75)·10^6 Pa; v_f =150...600Ns/m; and the control system of conveyor hydraulic drive is: k_3=(10…42)·10^3 N/m; b_3=(200…850)kg/s; W_3=(0,1…0,5)·10^{-3}m^3; m^3=(120…1200)·10^{-3}kg, stable operation modes of conveyor adaptive drive are provided.

At the same time, if the range of the parameters change is: k_1=(15…65)·10^{3}N/m; b_1=(200…850)kg/s; k_2=(10..27)·10^{3}N/m; b_2=(20…85)kg/s; f_d=(0,5…5)·10^{-6}m^2; Wn=(0,2…1,5)·10^{-3} m^3, the areas of stable and non-stable operation are determined (Fig. 5). Besides, at certain values of the parameters the operation of the corresponding elements did not take place. The absence of the result of elements operation, for instance, the switching of the distribution valve after sensor's operation on overloading was considered as the disability of the mathematical model and absence of stable operation of the conveyor's adaptive drive.

The obtained graphs (Fig. 5) enabled to determine the parameters of the adaptive drive of the conveyor in the ranges than provide stable operation: k_1=(28…46)·10^{3}N/m; b_1=(400…860)kg/s; k_2=(15…22)·10^{3}N/m; b_2=(25…80)kg/s; f_d=(1,8…5)·10^{-6}m^2; W_n=(0,4…1,2)·10^{-3}m^3.

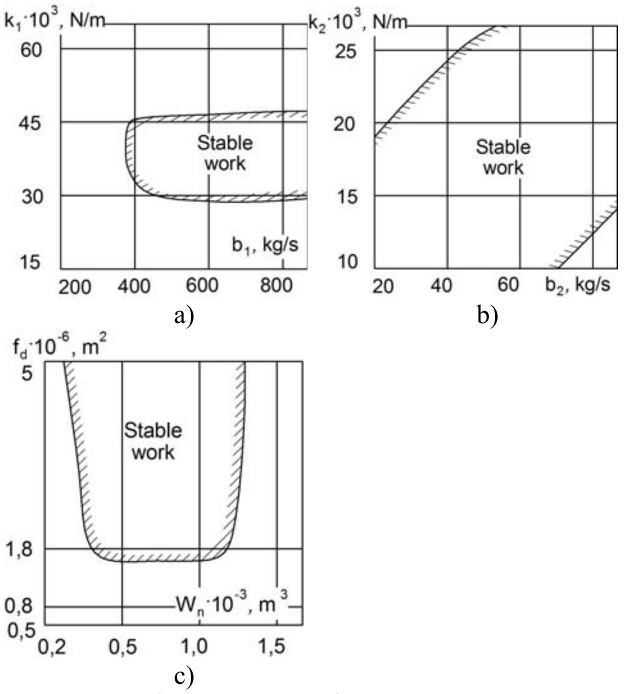


Fig. 5. Impact of the parameters of the conveyor adaptive drive on the stability of its operation: a – rigidities of spring k_1 and damping factor b_1 for the sensor; b – rigidities of spring k_2 and damping factor b_2 for the distribution valve; c – area f_d of the working window of the throttle valve and the volume of hydraulic supply line W_n

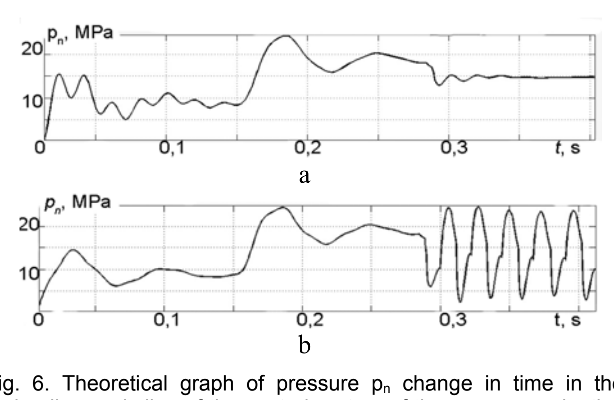


Fig. 6. Theoretical graph of pressure p_n change in time in the hydraulic supply line of the control system of the conveyor adaptive drive: a – $qm_2 \leq 0{,}5qm_1$, b – $qm_2 = 0{,}75qm_1$

Fig. 6 shows the impact of the characteristic volume qm2 of the hydraulic motor (HM_2) on the stability of the transient processes in the adaptive drive of the conveyor.

When the characteristic volume $q_{m2} \leq 0,5q_{m1}$ (Fig. 6, a), then after the switching on the hydraulic motor (HM_2) by the control system, the conveyor drive operates in stable mode. But when the valves of the characteristic volume of the additional hydraulic motor (HM_2) exceed the above-mentioned range (for instance, in Fig. 6, b for q_{m2} = $0,75q_{m1}$) after its switching on continuous oscillating processes are developing in the control system.

Fig. 7 illustrates the impact of the relation of the areas S = f_1/f_2 on the pressure of "opening" and "closing" of the control system sensor. In case of small relations (S = 0,1; 0,3) the transient processes take place in stable mode and the pressure p_1 "closing" has smaller values. The relation S = 0,5; 0,7 correspond to the pressure p_1 "closing", that provides normal operation mode of the conveyor's hydraulic drive, i.e., switching off of the additional hydraulic motor (HM2) occurs in accordance with the ideal sequence diagram of control system operation of the conveyor's adaptive drive (see Fig.3). If the value is S = 0,9, non-stable operation mode of the control system sensor is observed.

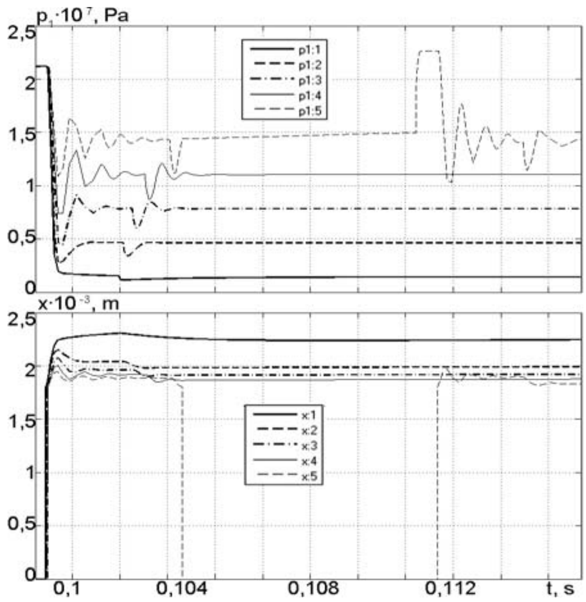
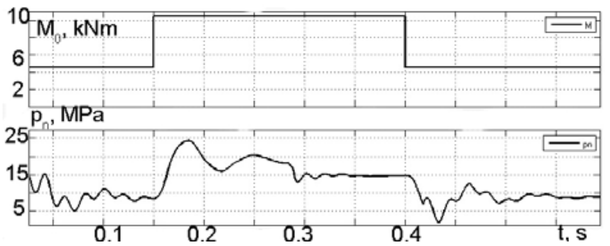
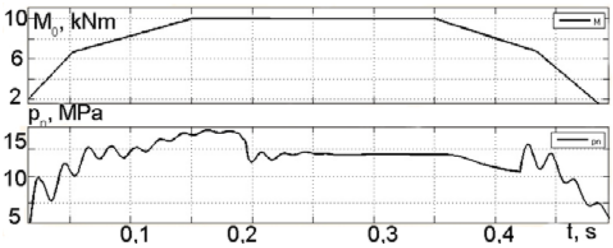


Fig. 7. Theoretical graph of p1(t) and x(t) change in time at S=0,1; 0,3; 0,5; 0,7; 0,9



a



b

Fig. 8. Theoretical graphs of load M(t) and pressure $p_n(t)$ change character in hydraulic supply line: a – stepwise load change; b – trapezoidal load change

The research, carried out, enabled to calculate the optimal relation of the conveyor's adaptive drive parameters with the control system, that provide stable operation mode. The values of these parameters are used in the process of these parameters are used in the process of studying the impact of load change character on the conveyor's drives in two operation modes: stepwise and trapezoidal (Fig. 8, a, b).

The analysis of the dynamic processes illustrated by these graphs, shows various impact of load change character at the phases of switching on and switching off of the additional hydraulic motor (HM2). In case of stepwise load change, switching on of additional hydraulic motor (HM2) is accompanied by large pressure overregulation (up to 40%) as compared with trapezoidal load change (up to 3%). During switching off of the additional hydraulic motor (HM2), vice versa – pressure overregulation for the stepwise load change (up to 5%) is smaller as compared with the trapezoidal one (up to 20%). However, the character of load change does not influence the operation stability of the conveyor's adaptive drive.

By the results of the research, recommendations, regarding the selection of adaptive drive parameters with the control system of the belt conveyor were formulated, technical documentation for the built-in hydraulic drive of the receiving conveyor of the clamp former К-65М2Б3-К for Private Joint-Stock Company "Kalynivskiy machine building plant" (Ukraine), the machines, manufactured at this plant are used at sugar mills of Ukraine and Europe.

Cocnclusions

1. Mathematical model of the conveyor's adaptive drive is improved as a result at taking into account elastic-inertia characteristics of the transporting part of the conveyor. By means of this model, using program product MATLAB Simulink the parameters of the adaptive drive, influencing the stability of the studied system with automatic regulation of motion parameters on condition of load change are determined. When the parameters of the rigidity and dumping factor for the sensor, distribution valve, area of working window of the throttle valve of the control system and the volume of hydraulic supply line changed, the area of stable operation of the conveyor's adaptive drive were revealed.
2. The analysis of theoretical graphs showed that in order to provide stable operation of the adaptive drive the characteristics volume of the additional hydraulic motor q_{m2} must not exceed 75% of the characteristic volume q_{m1} of the main hydraulic motor and the ratio of the sealing areas of locking element of the sensor must not exceed the value f_1/f_2=0,9. It is determined that the character of load change does not influence the stability of belt conveyor adaptive drive operation.

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