DYNAMICS OF THE CONVEYOR SPEED STABILIZATION SYSTEM AT VARIABLE LOADS

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Abstract. *The dynamic processes of the system of stabilization of the speed of the conveyor belt with a built-in hydraulic drive on the basis of an improved mathematical model, which takes into account the physical phenomena occurring in the hydraulic system during the action of alternating load. The influence of the main parameters of the hydraulic system and the magnitude of the load on the course of dynamic processes is analyzed and recommendations for their selection are formulated. The proposed scheme of the built-in hydraulic drive of a conveyor belt with system of stabilization of speed of movement automatically provides its uninterrupted work. The use of an additional hydraulic pump allowed to stabilize the speed of the conveyor belt to 7.8%, provided that the load on the working link is 2.3 times.*

Keywords: dynamics, belt conveyor, built-in hydraulic drive, speed stabilization system, variable load

DYNAMIKA SYSTEMU STABILIZACJI PRĘDKOŚCI PRZENOŚNIKA PRZY ZMIENNYCH OBCIĄŻENIACH

Streszczenie. *Rozważono procesy dynamiczne układu stabilizacji prędkości taśmy przenośnika z wbudowanym napędem hydraulicznym w oparciu o udoskonalony model matematyczny uwzględniający zjawiska fizyczne zachodzące w układzie hydraulicznym podczas działania obciążenia zmiennego. Przeanalizowano wpływ głównych parametrów układu hydraulicznego i wielkości obciążenia na przebieg procesów dynamicznych oraz sformułowano zalecenia dotyczące ich doboru. Zaproponowany schemat wbudowanego napędu hydraulicznego przenośnika taśmowego z systemem stabilizacji prędkości ruchu automatycznie zapewnia jego nieprzerwaną pracę. Zastosowanie dodatkowej pompy hydraulicznej pozwoliło ustabilizować prędkość taśmy przenośnika na poziomie 7,8% przy założeniu, że obciążenie ogniwa roboczego jest 2,3 razy większe.*

Słowa kluczowe: dynamika, przenośnik taśmowy, wbudowany napęd hydrauliczny, system stabilizacji prędkości, zmienne obciążenie

Introduction

The belt conveyors are the important components of current production technologies in various industries and the national economy. To a large extent, the efficiency of their use is determined by the technical characteristics of the drive device, the technical and economic indicators of which must provide the operating conditions of transport machines and their modes of operation [1, 4, 16].

Receipt and distribution of cargo moving on the conveyor belt, as a rule, can occur according to various laws.

Therefore, belt conveyors are mainly used in conditions of variable loads, the maximum excess of which in comparison with the nominal can become significant. When operating belt conveyor drives, conditions arise under which, due to the large load increase, the drive needs to be stopped immediately to prevent breakage of the drive. This reduces the productivity of the conveyor [3, 6, 14].

To increase the productivity of conveyors operated in the above conditions, it is proposed to use built-in hydraulic actuators with a control system for switching on an additional hydraulic motor. This will ensure uninterrupted operation, as well as greatly simplify the kinematic scheme according to the work [9, 13, 18]. Such drives work in the mode of constant supply of working liquid therefore the hydraulic system does not allow to provide stability of speed of movement of a belt of the conveyor [8,10]. When the additional hydraulic motor is switched on, the fluid flow is reduced by the value of the characteristic volume of this hydraulic motor and the speed of cargo transportation is reduced. The instability of the speed of movement of the conveyor belt leads to a decrease in productivity and reduces the efficiency of the control system with hydraulic automation [15, 19, 20].

1. Statement of the research problem

The purpose is to study the dynamic processes in the hydraulic drive conveyor speed stabilization system (SSS) of the belt and recommend the choice of parameters of the drive system.

To achieve this goal, the following tasks were solved in the work:

- for the developed hydraulic system of the belt conveyor, what operates in the modes of variable cargo flows, in which the means of hydro- and electroautomatics ensure the stability of the speed of the belt movement [11];
- the analysis of the influence of the main parameters of the hydraulic system and the magnitude of the load on the course of dynamic processes in the built-in hydraulic drive with the SSS was carried out.

2. Research and modeling method

Theoretical studies of the dynamic processes of the SSS of the conveyor for the action of variable loads were carried out by methods of physical and mathematical modeling. The solution of the improved mathematical model was performed using the software package MATLAB Simulink, which was used in previous studies of the hydraulic system [2, 12, 13]. 0.1% modeling accuracy and the 2nd order Rosenbrock research method have proven to be a qualitative way to calculate such systems.

To construct an improved mathematical model of the dynamics of the SSS of the conveyor, which operates under variable loads, a cyclogram of the operation of the hydraulic drive with the specified control system has been developed. For each phase of the cyclogram, differential equations were added, which took into account the physical processes characteristic of the corresponding time interval.

Fig. 1 shows a calculation scheme of the hydraulic system of the conveyor drive, operating in the mode of alternating cargo flows, for theoretical studies of dynamic processes.

The moving parts of the conveyor are represented by two discrete masses (12 and 14). The masses of rotating parts of drive devices with the moment of inertia I_3 are reduced to the first mass, to the second – moving links of the transporting part of the conveyor with the moment of inertia I_4. The moment of resistance of the useful forces M_0 acting on the conveyor is applied to the tail drum. The tensile forces of the tape F_1, F_2, F_3, F_4 are applied to the discrete masses 12 and 14. The conveyor belt is presented by Voigt's rheological model with elastic-dissipative connections.

SSS contains main 1 and additional 2 hydraulic pumps with characteristic volumes q_{n1} and q_{n2}, respectively. The maximum pressure created by them is determined by adjusting the safety valve 3. The executive units of the SSS hydraulic drive of the conveyor are the main 9 and additional 10 hydraulic motors with the corresponding characteristic volumes q_{m1} and q_{m2}. The flow of working fluid from the pumping station is supplied to the hydraulic motors with the corresponding supply. The executive link is a plunger 8 with a corresponding mass m_3 of friction clutch 7, which turns on the transmission mechanism of the additional hydraulic motor 10. The plunger 8 is loaded with a spring with a stiffness k_3. The working cavities of the hydraulic motors and the pressure plunger are connected to the corresponding cavities of the main component of the SSS by short hydraulic lines. The control device (CD) of the valve type performs the function of a distributor [5, 7, 17].
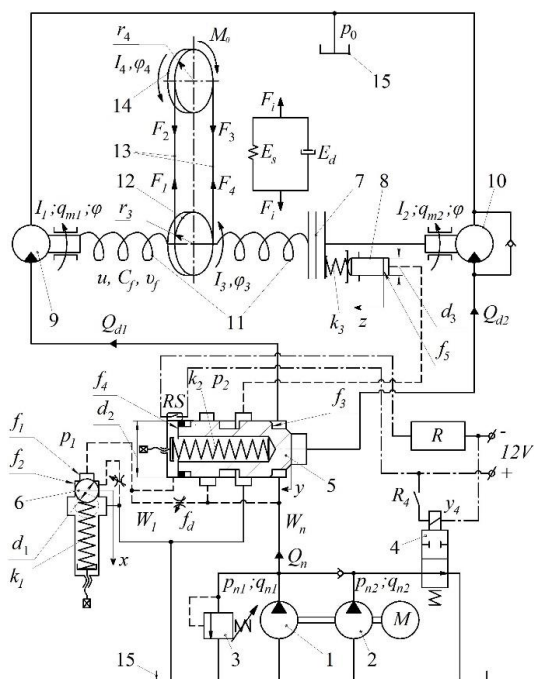

Fig. 1. Calculation scheme of the built-in hydraulic drive with system of stabilization of speed of movement of a belt of the conveyor

The main components of the CD are the valve of the first stage 6 (sensor), the ball shut-off element of which mass m_1 is loaded by a spring with stiffness k_1. The valve of the second stage 5 in the form of a shut-off and distribution element (SDE) with a mass of m_2 is loaded with a control spring of stiffness k_2. A control throttle with a cross-sectional area f_{d2} is installed between the intermediate and drain cavities of the sensor. In the hydraulic line between the cavities of the shut-off and distribution element is a control throttle with a cross section of area f_d. The shank of the shut-off and distribution element is equipped with a magnetic ring. On the side surface of the case there is a sensor type reed switch RS. Friction clutch 7 is activated when the set value of the pressure in the hydraulic system, which corresponds to the increased load. After that, the second transmission mechanism is activated by the additional hydraulic motor 10.

To compensate for the losses of the working fluid entering the main hydraulic motor 9 when the additional hydraulic motor 10 is connected to the hydraulic system, it is necessary to make up for the losses of the working fluid from the pumping station in the hydraulic system. This is ensured by switching on an additional hydraulic pump 2. Automation of this process is carried out due to operation of the sensor of the reed switch RS type at the beginning of movement of the shut-off and distributing device. According to the specified cyclogram with a small delay in time, it connects an additional hydraulic motor 10, and then triggers the friction clutch 7 of the drive gear of its transmission. The reed

switch RS supplies an electrical signal to the switch R_4 to turn on the power supply of the hydraulic lock 4, which turns on the supply of an additional hydraulic motor 10 to the hydraulic system. Due to the replenishment of the flow of working fluid in the hydraulic system, the speed of rotation of the shafts remains unchanged, as in the nominal load on the conveyor. The speed of the conveyor belt remains unchanged

The improved mathematical model of SSS is built on the basis of the calculation scheme (Fig. 1). D'Alembert's principle is applied to the forces acting on the moving elements of the mechanical system and the balance of the working fluid flow [13]. In the differential equations of change of the moment of forces of useful resistance, action of forces of viscous friction on rotating elements of hydraulic motors, change of directions of movement of working liquid in the course of work of CD, energy dissipation during movement of moving elements of system are considered.

The given mathematical model is an improved model of the adaptive hydraulic drive of the belt conveyor, the hydraulic system of which works in the mode of constant liquid consumption [13, 14].

Equation of equilibrium of moments on the shafts of the main 9 and additional 10 hydraulic motors during CD operation ($y > 0$; $F_a > F_{tr}$):

$$M + M_0 = q_{m1} \cdot p_n + q_{m2} \cdot p_n - \beta_m \frac{d\varphi}{dt} \tag{1}$$

The opening of the locking element of the sensor 6 occurs under the condition of increasing the pressure in the hydraulic drive to the value

$$p_1 \ge (k_1 \cdot x)/f_1 \tag{2}$$

The movement of the ball locking element after opening the sensor, provided that the pressure in the drain cavity $p_0 \approx 0$, is written by the equation:

$$f_2 \cdot p_1 = m_1 \frac{d^2 x}{dt^2} + k_1 (x_0 + x) + b_1 \frac{dx}{dt} + F_{g1} \tag{3}$$

where F_{g1} – hydrodynamic force in the sensor to be determined

$$F_{g1} = F'_{g1} - F''_{g2} = \rho Q \upsilon_w \cdot \cos \beta_0 - \rho Q \upsilon_c = \rho Q (\upsilon_w \cdot \cos \beta_0 - \upsilon_c) \tag{4}$$

If $0 \le x \le h_c$, $h_c = h_{cd} + h_{cb}$ flow rate through the sensor

$$Q = \mu \cdot \pi \cdot d_1 \cdot x \sqrt{2|p_1|/\rho} \cdot \text{sgn}(p_1) \tag{5}$$

The displacement of the shut-off element 5 for $0 \le y \le h_b$ is written by the equation:

$$p_n f_3 = p_1 f_4 + k_2 y_0, \text{ if } y = 0 \tag{6}$$

$$p_n f_3 = m_2 \frac{d^2 y}{dt^2} + k_2 (y_0 + y) + b_2 \frac{dy}{dt} + p_1 f_4 + F_{g2}, \text{ if } y > 0 \tag{7}$$

where F_{g2} – hydrodynamic force in the SDE to be determined

$$F_{g2} = \psi_g p_n f_3 \tag{8}$$

Equation of working fluid flow balance in a pressure hydraulic line:

$$Q_{n1} = Q_{m1} + Q_{fd} + \beta W_n \frac{dp_n}{dt}, \text{ if } y = 0 \tag{9}$$

$$Q_{n1} + Q_{n2} = Q_{m1} + Q_{m2} + Q_{fd} + \beta W_n \frac{dp_n}{dt} \quad \text{if } 0 < y < h$$

$$Q_{n1} + Q_{n2} = Q_{m1} + Q_{m2} + Q_{fd} + Q_k + \beta W_n \frac{dp_n}{dt}, \text{ if } y \ge h_d \tag{11}$$

there is an opening of the cavity of the plunger of SDE, where:

$$Q_{fd} = \mu \cdot f_d \sqrt{2|p_n - p_1|/\rho} \cdot \text{sgn}(p_n - p_1) \tag{12}$$

$$Q_k = \mu \cdot \pi \cdot d_2 \cdot (y - (h_b + h_d)) \sqrt{2|p_n - p_2|/\rho} \cdot \text{sgn}(p_n - p_2) \tag{13}$$

The flow rate of the working fluid Q_{fd} after the choke with a cross section f_d and the shut-off and distribution element in the cavity of the plunger 8 Q_k are determined by the expressions:

$$Q_{fd} = Q + \beta W_1 \frac{dp_1}{dt} \tag{14}$$

$$f_5 \cdot \frac{dz}{dt} + \mu \cdot \pi \cdot d_2 \cdot (h_b - y) \sqrt{2|p_2|/\rho} \cdot \text{sgn}(p_2) + \beta W_2 \frac{dp_2}{dt} = 0 \tag{15}$$

if $y = 0 ... h_b$ – there is a drain of the working fluid from the cavity of the plunger 8.

$$f_5 \cdot \frac{dz}{dt} + \beta W_2 \frac{dp_2}{dt} = 0 \text{ , if } y = h_b ... h_d \qquad (16)$$

$$Q_k = f_5 \cdot \frac{dz}{dt} + \beta W_2 \frac{dp_2}{dt} \text{ , if } y = h_d ... h \qquad (17)$$

it is the working fluid is injected into the cavity of the plunger 8.

The equation of motion of the plunger 8 of the friction clutch:

$$p_2 f_5 = m_3 \frac{d^2 z}{dt^2} + k_3 (z_0 + z) + b_3 \frac{dz}{dt} + F_a \qquad (18)$$

The force F_a begins to act at the moment of contact of the plunger with the coupling.

Closing of the ball locking element of the sensor occurs under the condition of reducing the pressure to the value [2]

$$p'_1 \le p_1 \cdot f_1 / f_2 + (k_1 \cdot x) / f_2 \qquad (19)$$

The following notations are used in equations (1-19): M – torque on the shafts of hydraulic motors; p_n – pressure in the hydraulic system at rated load on the working link; p_1 and p'_1 – pressure "opening" and "closing" of the ball locking element corresponding to the calculated maximum load on the working link; p_2 – pressure in the cavity of the plunger of the pressure mechanism; x, y, z – coordinates of movement of the corresponding masses; x_0, y_0, z_0 – initial deformation of compression springs; β_m – coefficient of viscous friction in the hydraulic motor; β – coefficient of pliability taking into account the compressibility of the working fluid; μ – coefficient of supply; ρ – the density of the working fluid; $S = f_1 / f_2$ – the ratio of the area of contact of the shut-off element with the valve seat to the area of the cylindrical part of the valve; f_3, f_4, f_5 – respectively, the surface area of the ends of the shut-off and distribution element and the plunger 8; h_c, h, h_m – respectively the course of the ball locking element 6, shut-off and distribution element 5 and the friction clutch discs; W_n, W_1, W_2 – the volume of the pressure line, the cavity of the sensor 6 and the cavity of the plunger 8, respectively; d_1, d_2, d_3 – diameter of ball locking element 6, shut-off and distribution element 5 and friction clutch disks, respectively; b_1, b_2, b_3 – viscous damping factor; β_0 – the angle of inclination of the jet of working fluid; v_w, v_c – the speed of the fluid in the slit and the saddle, respectively; ψ_g – force factor; Q_{fd}, Q_k – fluid flow rate for the throttle with a cross-sectional area fd, in the cavity of the plunger 8, respectively; F_{g1}, F_{g2} – hydrodynamic forces acting on the ball valve 6 and shut-off and distribution element 5; F_a – coupling reaction of couplings; F_{tr} – friction force between the clutch discs.

3. Results and discussions

Consider the case when the disturbance for the hydraulic system is a change in load torque of 2.3 times compared to the nominal. In Fig. 2 shows the dependence of the load torque M and the angular velocity $d\varphi/dt$ of rotation of the drive drum of the conveyor from time t without the use of an additional hydraulic pump. The overload mode occurs when the moment of force of resistance M increases from 4500 to 10500 N·m. The mechanical system in this case perceives a decrease in the speed of rotation of the drive drum by 27.7%. When the additional hydraulic motor 10 is switched on, the angular velocity $d\varphi/dt$ is exceeded by 13% relative to the nominal one. After the overload disappears and the additional hydraulic motor 10 is switched off, the value of exceeding the angular velocity $d\varphi/dt$ is 18.1%, which is negatively reflected in the dynamics of the conveyor.

In Fig. 4 shows the influence of the characteristic volume of the additional hydraulic pump 2 and the characteristic volume of the additional hydraulic motor 10 of the conveyor hydraulic drive on the accuracy of the speed stabilization δ, the value of σ_{on} the excess of the angular speed $d\varphi/dt$ relative to the nominal one when the additional hydraulic motor 10 is turned on and the value of σ_{off} – during its shutdown at a constant overload value. On the abscissa axis, point 1 corresponds to the specified characteristics of the conveyor hydraulic drive at a ratio of 25%

of the characteristic volume of the additional hydraulic motor 10 to the characteristic volume of the main hydraulic motor 9 and in the absence of an additional hydraulic pump 2.

When using the additional hydraulic pump 2 in the hydraulic system, the dynamic and static characteristics of the conveyor drive are significantly improved (Fig. 3). Consider the case of using in the hydraulic system of the additional hydraulic pump 2 with a characteristic volume of 25% of the main hydraulic pump 1, and the same ratio of the characteristic volumes of the additional 9 and the main 10 hydraulic motors

It is established that the accuracy of speed stabilization δ improves to 7.8%, and the dynamics of the drive drum improves when turning off the additional hydraulic motor 10. The value of exceeding the angular velocity $d\varphi/dt$ relative to the nominal is 8.4%, which is 2.2 times less than when working with one hydraulic pump and reduced belt speed of the conveyor.

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Point 2 – corresponds to the characteristics of the hydraulic system, the design diagram of which is shown in Fig. 1, with a ratio of 25% of the characteristic volume of the additional hydraulic motor 10 to the characteristic volume of the main hydraulic motor 9, and the same ratio of the characteristic volumes of the additional 2 and main 1 hydraulic pump in the overload mode. Point 3 – corresponds to the above ratio, which is 37.5%, and point 4 – 50%.

The analysis of the obtained characteristics showed that with the increase of the ratios of the characteristic volumes of hydraulic motors and hydraulic pumps from 25% to 50%, the accuracy of stabilization of the speed δ of the conveyor increases from 7.8% to 4.4%. However, there is a deterioration in the quality of dynamic characteristics: the value of σ_{on} exceeding the angular velocity $d\varphi/dt$ relative to the nominal when turning on the additional hydraulic motor 10 varies from 20.7% to 51.9%, and the value of σ_{off} when turning off the additional hydraulic motor 10 – from 8.4% to 26.5%.
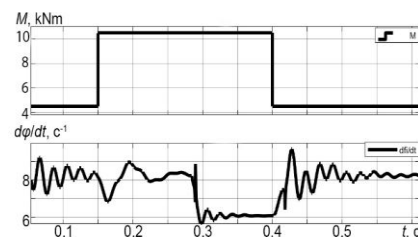

Fig. 2. Dependence of loading moment M and angular speed of rotation of a driving drum dφ/dt on time t without the auxiliary hydraulic pump
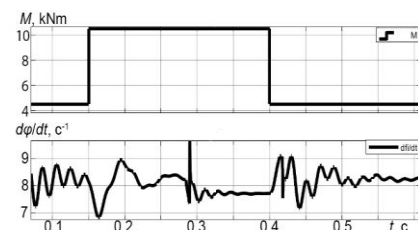

Fig. 3. Dependence of loading moment M and angular speed of rotation of a driving drum dφ/dt on time t with the auxiliary hydraulic pump
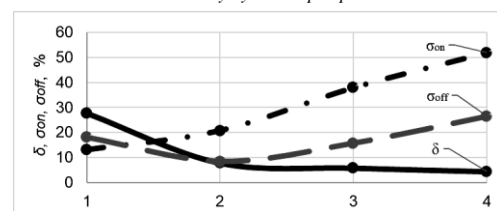

Fig. 4. Static and dynamic characteristics of the hydraulic drive of the conveyor with control systems at different ratios of characteristic volumes of hydraulic motors 9 and 10, and also hydraulic pumps 1 and 2

4. Conclusions

1. The proposed scheme of the built-in hydraulic drive of a conveyor belt with system of stabilization of speed of movement automatically provides its uninterrupted work. The use of an additional hydraulic pump allowed to stabilize the speed of the conveyor belt to 7.8%, provided that the load on the working link is 2.3 times.

2. Analysis of the transients of the dynamic characteristics of the mechanical system of the conveyor for an advanced mathematical model of the pressure pn in the pressure line of the hydraulic drive showed a slight effect of increasing the flow of working fluid when connected to the hydraulic system of an additional hydraulic pump 2.

3. The most appropriate for this increase in the load on the working link of the conveyor is the use of an additional hydraulic motor 10 and hydraulic pump 2 with a ratio of characteristic volumes of 25% to the main.

References

[1] Forental V., Forental M., Nazarov F.: Investigation of Dynamic Characteristics of the Hydraulic Drive with Proportional Control. Procedia Engineering 129, 2015, 695–701.

[2] Gubarev A. P., Hanpantsurova O. S., Belikov K. A. et al.: Logic correctness of control algorithms for mechatronic discrete systems with parallel processes. Proc. of SPIE 11176, 2019, 1117660.

[3] Gubarev A., Yakhno O., Ganpantsurova O.: Control Algorithms in Mechatronic Systems with Parallel Processes. Solid State Phenomena 164, 2010, 105–110 [http://doi.org/10.4028/www.scientific.net/SSP.164.105].

[4] He D., Liu X., Zhong B.: Sustainable belt conveyor operation by active speed control. Measurement 154, 2020, 107458.

[5] Khmara L. A., Shatov S. V., Polishchuk L. K. et al.: Algorithm to calculate work tools of machines for performance in extreme working conditions. Wójcik W., Pavlov S., Kalimoldayev M. (ed.): Mechatronic Systems I. Applications in Transport, Logistics, Diagnostics and Control. Taylor & Francis Group – CRC Press, London, New York 2021, 29–38.

[6] Kozlov L. G., Polishchuk L. K., Piontkevych O. V. et al.: Experimental research characteristics of counterbalance valve for hydraulic drive control system of mobile machine. Przeglad Elektrotechniczny 95(4), 2019, 104–109.

[7] Kozlov L., Polishchuk L., Piontkevych O. et al.: Optimization of design parameters of a counterbalance valve for a hydraulic drive invariant to reversal loads. Wójcik W., Pavlov S., Kalimoldayev M. (ed.): Mechatronic Systems I. Applications in Transport, Logistics, Diagnostics and Control. Taylor & Francis Group – CRC Press, London, New York 2021, 137–148.

[8] Kukharchuk V. V., Bogachuk V. V., Hraniak V. F. et al.: Method of magneto-elastic control of mechanic rigidity in assemblies of hydropower units. Proc. of SPIE 10445, 2017, 104456A.

[9] Kukharchuk V. V., Kazyv S. S., Bykovsky S. A. et al.: Discrete wavelet transformation in spectral analysis of vibration processes at hydropower units. Przeglad Elektrotechniczny 93(3), 2017, 65–68.

[10] Kushwaha P., Dasgupta K., Ghoshal S. K.: A comparative analysis of the pump controlled, valve controlled and prime mover controlled hydromotor drive to attain constant speed for varying load. ISA transactions 120, 2022, 305–317 [http://doi.org/10.1016/j.isatra.2021.03.020].

[11] Nykyforov V. V., Salamatin D. M., Digtiar S. V. et al.: Toxicity by Digestate of Methanogenic Processing of Biomass Wójcik W., Pawłowska M. (ed.): Biomass as Raw Material for the Production of Biofuels and Chemicals. Taylor & Francis Group – CRC Press, London 2021, 155–170.

[12] Obertyukh R., Slabkyi A., Polishchuk L. K. et al.: Method of project calculation of hydroimpulsive device for vibroturning with an incorporated cycle spring pressure pulse generator. Wójcik W., Pavlov S., Kalimoldayev M. (ed.): Mechatronic Systems I. Applications in Transport, Logistics, Diagnostics and Control. Taylor & Francis Group – CRC Press, London, New York 2021, 1–16.

[13] Polishchuk L. K., Kozlov L. G., Piontkevych O. V., Gromaszek K., Mussabekova A.: Study of the dynamic stability of the conveyor belt adaptive drive. Proc. of SPIE 10808, 10808, 2018, 1–10.

[14] Polishchuk L., Kharchenko Ye., Piontkevych O., Koval O.: The research of the dynamic processes of control system of hydraulic drive of belt conveyors with variable cargo flows. Eastern Eur. J. Enterp. Technol. 2(8(80)), 2016, 22–29.

[15] Polishchuk L., Mamyrbayev O., Gromaszek K.: Mechatronic Systems II. Applications in Material Handling Processes and Robotics. Taylor & Francis Group – CRC Press, Boca Raton, London, New York, Leiden 2021 [http://doi.org/10.1201/9781003225447].

[16] Semrád K., Draganová K., Koščák P., Čerňan J.: Statistical prediction models of impact damage of airport conveyor belts. Transportation research procedia 51, 2020, 11–19 [http://doi.org/10.1016/j.trpro.2020.11.003].

[17] Shatokhin V., Granko B., Sobol V. et al.: Vibration diagnostic of wear for cylinder-piston couples of pumps of a radial piston hydromachine. Wójcik W., Pavlov S., Kalimoldayev M. (ed.): Mechatronic Systems I. Applications in Transport, Logistics, Diagnostics and Control. Taylor & Francis Group – CRC Press, London, New York 2021, 39–52.

[18] Vedmitskyi Y. G., Kukharchuk V. V., Hraniak V. F.: New non-system physical quantities for vibration monitoring of transient processes at hydropower facilities, integral vibratory accelerations. Przeglad Elektrotechniczny 93(3), 2017, 69–72.

[19] Wójcik W., Pavlov S., Kalimoldayev M.: Mechatronic Systems I. Applications in Transport, Logistics, Diagnostics and Control. Taylor & Francis Group – CRC Press, London, New York 2021 [http://doi.org/10.1201/9781003224136].

[20] Yelizarov M. O., Pasenko A. V., Zhurav V. V. et al.: Fallen Leaves and Other Seasonal Biomass as Raw Material for Producing Biogas and Fertilizers. Wójcik W., Pawłowska M. (ed.): Biomass as Raw Material for the Production of Biofuels and Chemicals. Taylor & Francis Group – CRC Press, London 2021, 145–154.

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