Innovative: International Multi-disciplinary Journal of Applied Technology
(ISSN 2995-486X) VOLUME 04 ISSUE 07, 2026

ANALYSIS OF CALCULATION RESULTS OF VIBRATION OF BELT CONVEYOR ROLLER MECHANISMS

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Abstract:

The article presents the determination of the roller mechanisms of the belt conveyor according to the vibrations of the outer shell, taking into account the values of the technological load and friction force. The laws of the displacement of the outer shell of the part acting as a sliding support used in the roller mechanism, the change in the vibration speed are obtained. Their recommended values are based on the analysis of the parameters and changes in the modes of movement of the roller, as well as the graphs of the dependence of the technological resistance of the transported product on the parameters of the roller.

Keywords: belt conveyor, roller, guide, vibrations, mass, load, stiffness, dispersion, spacing, mixture, speed, regularity, mechanical engineering, manufacturing, processing, gear cutting tool.

Introduction

Belt conveyors are widely used in various sectors of the economy of the Republic of Uzbekistan and foreign countries, in particular in the mining industry, their design, continuous delivery of products over long distances, and in general, they are fundamentally different from other means of transport in terms of economic efficiency [1]. One of the components of a belt conveyor is roller mechanisms, which are distinguished by their certain place in the equipment. The share of roller mechanisms in quantitative terms is also significant, which allows for the effective solution of a number of important technical and economic problems in single-belt conveyors [2].

Therefore, the creation of a single design method for belt conveyors with maximum formalization of all design procedures is currently a very urgent scientific and technical task. Its solution will significantly simplify the work of design organizations, allow finding optimal solutions in various design situations, allow comparing the effectiveness of their use on a particular conveyor, and also expand the scope of areas for creation [3].

Methodology

The main contribution to the low-frequency vibration of conveyor rollers, which include rolling bearings in their design, is made by the constituent vibration components at frequencies that are multiples of the roller frequency and do not depend on the condition of the bearings, but on the correct centering of the roller supports. In the process of a detailed analysis of the low-frequency vibration spectrum of the pulley, it was revealed that it contains a significant number of weakly expressed components characterized by the quality of assembly of working elements and bearing installation, as well as the influence of various operational factors on the bearing units.

Bearing vibration also depends on the overall dimensions of the bearing, in particular, due to periodic changes in the stiffness of the bearing, the conveyor roller only loads a small area along the outer ring, where the frequency of rotation periodically changes.

The specified bearing rotation frequencies cause vibrations not only at low, but also at medium frequencies, including, first of all, vibrations of bearings or roller support bearings at harmonics of these frequencies with a high coefficient of friction. Medium-frequency periodic components of bearing vibrations are often dynamic in nature, but they are caused not by extended and smooth irregularities of the rolling surfaces, but by small irregularities with sharp edges. Good-quality lubrication and low radial loads on the bearing smooth out the edges of these irregularities, which leads to a decrease in medium-frequency vibrations of the bearing units. However, in loaded bearings, medium-frequency bearing vibrations can increase:

– when the properties of the lubricant deteriorate;

– when there are assembly and installation defects in the bearing, which lead to an increase in static or rotating loads;

– when the vibration of at least one of the bearing components or their harmonics coincides with at least one of the many resonances of the conveyor belt or bearing assembly elements.

Results

During the operation of the belt conveyor, the transported minerals directly affect the three-row roller mechanism. As a result, various repair services are observed in the roller mechanism. In particular, the premature repair of the bearing support in the roller mechanism is observed [4].

In the process of assembling a roller mechanism with a length of L and a diameter of D, it is necessary to take into account the interchangeability of its structural parts relative to each other and the compressive force relative to each other [5]. In this case, to determine the stiffness of the part as a result of the process that occurs when the sliding bearing with a sliding element rubs against the shaft during operation, we determine it using the following expression:

$C = (1 + h_g)\cdot(N + \lambda_q E_b F_b) R^2$,

where, C – is the wear hardness of the graphite-caprolon part of the roller sliding bearing with a sliding element, h_g – is the thickness of the sliding element bearing, N – is the effect function of the friction forces between the sliding element bearing and the shaft, λ_q – is the effect function of the deformation of the sliding element, E_b, F_b – is the modulus of elasticity and the cross-sectional area [6].

To consider the vibration process of a belt conveyor roller mechanism, we consider Hamilton's method and Lagrange's second-order equations.

In this case, the vibrations between the sliding bearing with a sliding element and the bearing shell are distributed according to a certain law [7].

fmpi·(z) = *ρp·Fpi* (z)*·u·fmni·*(z) = *ρn·Fni* (z)

If the number of saws is j, then their mass distribution function is $(j-1)$. In this case, the stiffness of the outer ring of the sliding bearing with a sliding element is determined by $c = 2NR^2$, then the potential and kinetic energies of the roller mechanism during oscillations are as follows:

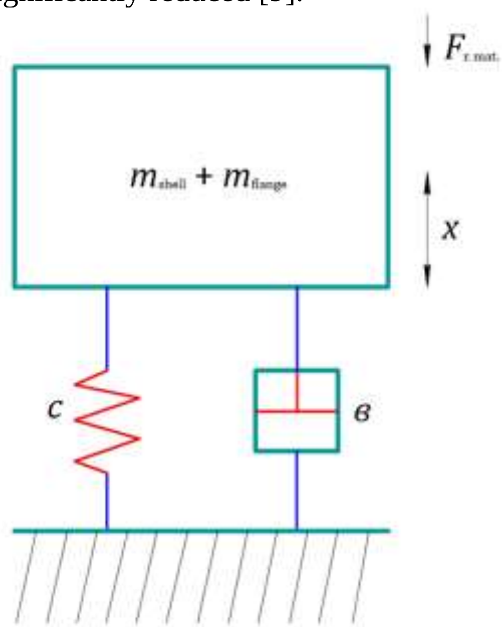
$$U = \frac{1}{2}\int_0^L 2NR^2 \left(\frac{\partial^2 u}{\partial z^2}\right)^2 dz; \qquad T = \frac{1}{2}\int_0^L \left(\frac{\rho_n j l_p F_p}{L} + \frac{\rho_n (j-1) l_n F_n}{L}\right)\left(\frac{\partial u}{\partial t}\right)^2 dz \qquad (1)$$

where, U, T – are the potential and kinetic energies of the roller mechanism during oscillations; u – is the displacement of the outer ring of the sliding bearing with a sliding element from the geometric Z axis in the OX direction [8].

Accordingly, using the method presented in [4,5], we obtain the following:

$$\frac{\partial^2}{\partial z^2}\left(2NR^2\frac{\partial^2}{\partial z^2}\right) + \left(\frac{\rho_n j L_p F_p}{L} + \frac{\rho_n (j-1) L_n F_n}{L}\right)\frac{\partial^2 u}{\partial t^2} = 0 \qquad (2)$$

If we consider the solution of the given (2), i.e., $u(z_1,0) = f(z)\ \frac{\partial u}{\partial z}\ (z_1,0) = f(z)$, then the boundary conditions depend on the form of their connection ($z = 0$; $z = e$). When installing the roller mechanism on the supports of sliding bearings with a sliding element (Fig. 1), its vibrations are significantly reduced [9].



where, F_{rm} – is the impact force of the raw material; x – is the direction of vibration; m_{shell} – is the mass of the bearing shell; m_{flange} – is the mass of the outer metall flange; c, $в$ – is the elasticity and dissipation coefficients of the rubber bushing [10].

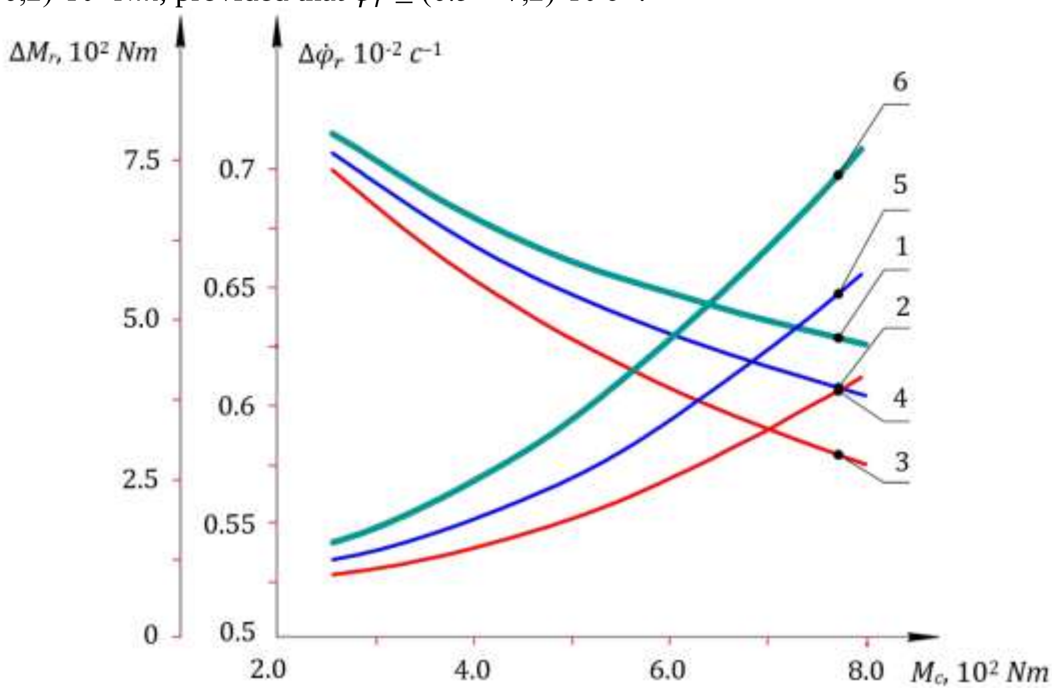
Figure 1. Calculation diagram representing the vibration of the proposed belt conveyor guide roller mechanism shell and flange

The solution representing the free vibrations of the bearing shell and outer metal flange, according to, has the following form:

$$x = \sqrt{x_0 \frac{\dot{x}_0^2}{p^2}\cos\left[pt + arctg\left(\frac{\dot{x}_0^2}{px_0}\right)\right]} \qquad (3)$$

As a result of the numerical solution of (3), it is possible to determine the free vibrations of the composite bearing shell and outer metal flange, that is, the law of motion in the absence of load on the belt conveyor [11]. However, from a practical point of view, it is important to determine the forced vibrations of the bearing shell and outer metal flange, to evaluate the influence of the damping-dissipative properties of the rubber bushing on vibrations, and to determine the influence of the working capacity on the change in vibrations [12].

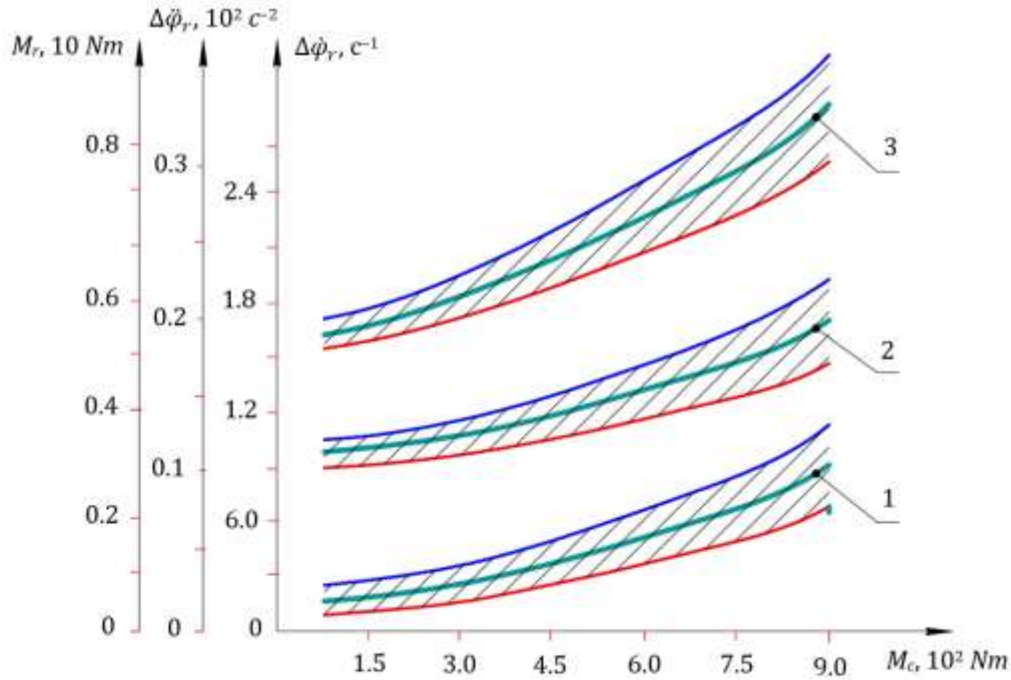
Figure 2 shows graphs of the dependence of the angular velocity and torque of roller mechanisms on the change in the technological load. Accordingly, with an increase in M_r, the graphs show a nonlinear law, $\dot{\varphi}_r$ decreases, and $M_{q.t}$ increases, respectively. As a result, with an increase in the average value of the technological load from $2{,}8\cdot10^2$ Nm to $7{,}92\cdot10^2$ Nm, the average value of the angular velocity of the roller mechanism decreases from $0{,}739\cdot10^2$ c^{-1} to $0{,}631\cdot10^2$ c^{-1} [13]. As a result, the moment of inertia of the roller mechanism is 1,05 $kg\cdot m^2$. In this case, the average value of M_r increases from $1{,}07\cdot10^2$ Nm to $4{,}07\cdot10^2$ Nm. This is explained by the fact that at small reduced values of the roller mechanism, the influence of the technological load on the change in $\dot{\varphi}_r$ is significant [14]. With a moment of inertia of J_n = 1,5 $kg\cdot m^2$, the angular velocity of the roller mechanism decreases to $0{,}592\cdot10^2$ c^{-1}, and the torque increases to $7{,}108\cdot10^2$ Nm. It should be noted that a significant decrease in angular velocity also leads to a decrease in the periodicity of the roller mechanism. Therefore, the recommended parameter values are J_n = (1.0 ÷ 1,25) $kg\cdot m^2$; $M_r \leq$ (4.5 ÷ 6,2)$\cdot10^2$ Nm, provided that $\dot{\varphi}_r \geq$ (6.5 ÷ 7,2)$\cdot$10 c^{-1}.



1,2,3 – $\dot{\varphi}_r = f(M_r)$; 4,5,6 – $M_{rq} = f(M_r)$; 1,4 – J_r = 1.05 $kg\cdot m^2$;
2,5 – J_r = 1.3 $kg\cdot m^2$; 3,6 – J_r = 1.5 $kg\cdot m^2$.

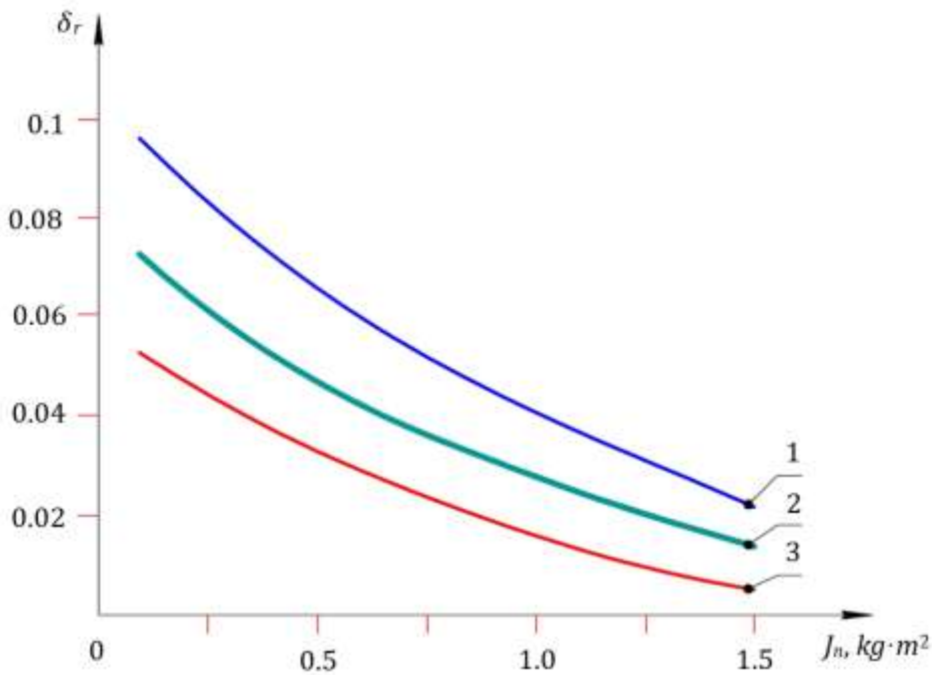
Figure 2. Graphs of the dependence of the change in angular velocity and torque of roller mechanisms on the change in technological load

Analysis of the graphs in Figure 3 shows that with an increase in the technological load, the range of oscillations of $\Delta\dot{\varphi}_r$, $\Delta\ddot{\varphi}_r$ and M_r increases significantly. Thus, $M_r = 1.5\cdot10^2$ Nm, $\Delta\dot{\varphi}_r$ – 3.89 c^{-1}, and M_r = $8{,}92\cdot10^2$ Nm, reaches $\Delta\dot{\varphi}_r$ = 7.2 c^{-1}. The acceleration range increases from $0.12\cdot10^2$ c^{-1} to $0.185\cdot10^2$ c^{-1}. Therefore, to reduce random oscillations, it is advisable to reduce $\Delta\dot{\varphi}_r$, $\Delta\ddot{\varphi}_r$ and M_r to (5,5 ÷ 6,5)$\cdot10^2$ Nm, especially random oscillations, which depend on the uniformity of external technological loads.



1-$\Delta M_r = f(M_c)$; 2-$\Delta\dot{\varphi}_r = f(M_c)$; 3-$\Delta\dot{\varphi}_r = f(M_c)$ da $\delta \mathrm{M}_r = (10 \div 12)\%\ M_r$

Fig. 3. Graphs of the dependence of the amplitude of fluctuations of the angular velocity, angular acceleration and torque of a belt conveyor, taking into account random vibrations of its roller mechanisms



1- $M_r = 4.3 \cdot 10^2$ *Nm*; 1- $M_r = 6.74 \cdot 10^2$ *Nm*; 1- $M_r = 8.21 \cdot 10^2$ *Nm*.

Fig. 4. Graphs of the dependence of the change in the angular velocity unevenness coefficient of the belt conveyor roller mechanism on the change in the reduced moment of inertia at different values of the technological load

Figure 4 shows the graphs of the dependence of the change in the angular velocity unevenness coefficient of the roller mechanism of the belt conveyor on the change in the reduced moment of inertia of the roller mechanism. Accordingly, the study of the dynamics of the belt conveyor allows you to study the unevenness of the angular velocity of the working bodies and select parameters that provide the required values of the uneven movement of the working bodies [15]. It is known [8] that the greater the moment of inertia of the working body, the lower the angular velocity unevenness coefficient. For the case under consideration, it is recommended to choose $J_r = (1{,}0 \div 1{,}25)\ kg{\cdot}m^2$ to ensure $\delta \leq (0{,}08 \div 0{,}09)$.

Discussion

During the operation of belt conveyor roller mechanisms, vibrations occur due to external shocks and high loads, as well as unbalanced loads.

The existing belt conveyor has three rows of guide rollers installed at a certain angle (20°-40°) to each other, spaced at 500-600 mm. The smooth rotational movement of the guide rollers is carried out by rolling bearings. As noted above, these rolling bearings fail very quickly due to the loads. In order to eliminate this drawback, rollers equipped with a sliding bearing with a damping belt element of a new design were recommended [16].

The analysis of the research showed that it is possible to increase the service life of the roller by using special high-strength plastic and rubber element materials instead of the rolling bearing of the belt conveyor roller. Because when using a rubber element sliding bearing in the roller, the vibration amplitude in it decreases, and the moment of resistance and friction force are reduced as a result of the use of plastic lubricants, the UWC can be increased compared to the traditional one [17].

The proposed belt conveyor operates at high loads, the transported ore is not evenly distributed along the width and length of the belt. Therefore, the load on each guide roller affects the values of varying values in a certain range. To sufficiently absorb these loads, rubber bushings are used in the structure of the rollers. Vibration absorption reduces the amplitude of vibration of the conveyor belt. This, in turn, also partially saves energy for rotating the belt [18].

Conclusion

The effect of the stiffness of the roller bearing support of the belt conveyor roller mechanisms on the amplitude of vibrations was studied. A mathematical model of the vertical vibrations of the roller mechanism was obtained, taking into account the technological load received from the loading of the roller bearing support. As a result, the laws of changes in the vertical displacements and speeds of the roller mechanism were obtained based on the numerical solution of the problem. Formulas were obtained for calculating the amplitude and frequency of the roller mechanism vibrations, as well as the dynamic coefficient of the system. In addition, graphs of the dependence of the change in the amplitude of the roller mechanism vibrations on the change in the rotation speed and the stiffness coefficient of the roller bearing support were constructed.

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