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JUSTIFICATION OF RATIONAL DESIGN FOR CONVEYOR ROLLER BEARINGS TO ENSURE THEIR ENERGY EFFICIENCY

The article considers various types of belt conveyors and the results of their operation, and demonstrates advantages of a reasonable choice of design parameters for bearings and rollers of belt conveyors in reduction of energy consumption and operating costs. The influence of design parameters of roller bearings on their rolling resistance is analyzed. The research results include the theoretical models for accurate prediction of the main rolling resistance of belt conveyor rollers. The tests to obtain the data directly usable in design and operation of roller bearings are described. The presented reasonable choice of design solutions can reduce not only energy consumption but also total cost of a conveyor. Thanks to the combination of the presented theoretical models and data obtained as a result of the studies, directions for designing rollers are determined to ensure energy efficiency of belt conveyors.

Keywords: *belt conveyor, rolling resistance, roller, belt, bearing assembly, seal, labyrinth, lip, grease, viscosity*
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Introduction

An optimized design of a belt conveyor line includes a rational design of the conveyor rollers, such that minimize energy consumption and operation cost for the whole life of the conveyor. It often happens that belt conveyor designs aim at minimization of capital costs rather than at reduction of the overall cost of conveyor operation. Justified designs of sealing for conveyor roller bearings also result in their economic and energy efficient operation [1–3]. In this respect, it is relevant to determine main design factors of energy-efficient bearings of conveyor rollers as they influence design and configuration of conveyor belt drives.

Resistance to rolling of belts is addressed in many studies [4–12]. Also, many researchers focused on the other methods of minimizing energy consumption by belt conveyors [13–18].

The prime areas of possible minimization in energy consumption by belt conveyors are determined, namely, the decrease in rolling resistance due to pressing-in of the roller barrel to the belt, which is a priority area of research as it greatly influences the overall rolling resistance of belts.

The other analyzed variants include: rollers with an increased diameter; special conveyor belts; drums with low rolling resistance [19]. It is studied how the increased spacing of the loaded belt rollers influence the energy and economic variables of long conveyors [20, 21]. It is shown that some studies may be successfully used in design of energy-efficient belt conveyors.

The present approach includes the analysis of individual components of rolling resistance of belts, presentation of analytical models for the prediction of each component in combination with the laboratory test data, and demonstration of the approach applicability in design of energy-saving conveyor bearings. Furthermore, some standards and methods of testing belt rollers, having influence on the value of rolling resistance, from various reference books are reviewed.

At the present time, some scientists publish only results inapplicable in design as these values are usually beyond the real-life operating conditions of conveyors.

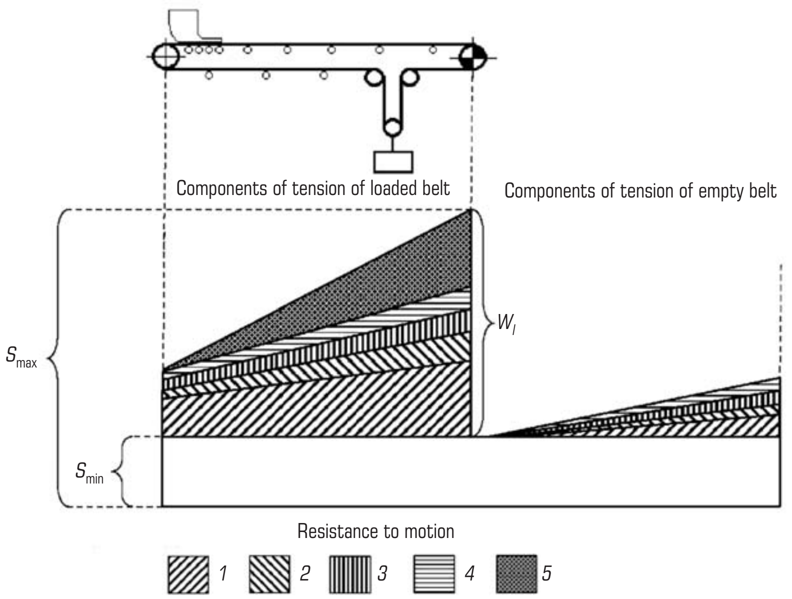

Fig. 1. Forces of resistance to motion along the length of horizontal conveyor belt in steady run: *1* – resistance due to roller pressing-in in belt; *2* – resistance of belt deformation; *3* – rolling resistance of roller; *4* – roller displacement resistance; *5* – resistance of cargo deformation on belt; S_{min} – minimum tension in the conveyor belt, N; S_{max} –maximum tension in the conveyor belt, N; W_l – Resistance to motion of loaded belt, N

The aim of the research: is to select and justify rational design of sealing for roller bearings along the whole length of a conveyor.

The objectives of the research: are to determine components of roller resistance depending on bearing sealing design, temperature, ambient environment and lubricant.

Initial conditions

Design of a belt conveyor for certain operating conditions usually includes calculations of drive power input, belt tensions in various sites of the belt (inclined, horizontal, curved), and the belt factor of safety. **Figure 1** depicts different components of rolling resistance of horizontal belt conveyor in steady run.

The components of resistance to motion in Fig. 1 sensibly differ from the German standard DIN 22101 [22] and European standard ISO 5048 [19], but are similar to the standards of Conveyor Equipment Manufacturers Association (CEMA) [20]. The overall rolling resistance in Fig. 1 comprises: resistance due to roller pressing-in in the belt; resistance of belt deformation; rolling resistance of roller; roller displacement resistance; resistance of cargo deformation on the belt. According to DIN and ISO standards, the overall resistance of the whole belt is calculated by summing up all values of resistances to motion in Fig. 1.

Secondary resistance to belt motion can be included as a coefficient of the belt length, which is comprehensively described in DIN and ISO standards. As shown in Fig. 1, the components of resistance due to belt deformation and cargo deformation increase with the increasing length of the belt, due to increased tension of the belt along its length. It is assumed that the other components of rolling resistance are independent of the belt tension if the belt route is free from curved vertical and horizontal bends. One of the components of resistance to belt motion—rolling resistance of rollers F_{rol}—is analyzed in detail. The analytical models are presented in combination with the laboratory test data, and the potential ways of decreasing resistance to belt motion are revealed.

Rolling resistance of conveyor roller

The main function of rollers of belt conveyor is to support the belt along its whole length. The calculation of the overall rolling resistance of rollers is particularly important in calculation of belt tensions in certain belt branches and, as a consequence, in determination of energy intake of drives, especially on long main conveyors, with more than three thousand rollers per each kilometer of length.

Rolling resistance of conveyor rollers appears due to friction of balls in bearings, viscous resistance of lubricant and friction of seals. Rolling resistance of rollers is usually 5 to 15% of the overall resistance to belt motion in case of long belts [12, 13]. In the design of main belt conveyors several kilometers long, exact determination of such resistance is critical for selecting parameters of drives and belt length.

Rolling resistance of rollers depends, first of all, on the type and design of seal, type of bearings, temperature of lubricant and rotation speed of rollers.

The lip- and labyrinth-type seals with grease usually prevent enter of dust and water in bearings. The labyrinth seals can be totally replaced by grease to enhance their efficiency, which leads to initiation of viscous resistance as a result of shear of grease layers between rotating and immobile surfaces of the seals.

The external lip seals prevent entry of external pollutants in the labyrinth seals, and the internal lip seal keeps grease inside bearings. The external and internal lip seals increase rolling resistance of rollers because of the structural features of the sealing mechanism. Rollers usually include bearings, and friction appears in the latter as function of their type and size, rotation speed, properties and amount of lubricant and, to a lesser degree, actual load. Overall rolling resistance of bearings consists of rolling and sliding friction between balls and separator, and internal friction of lubricant [23].

Experimental results on rolling resistance of conveyor roller

There exist a number of standards to measure rolling resistance of rollers on special testing machines [6, 13, 21, 24], and the methods of measurement coincide in many ways. The South African National Standards (SANS) use a rotating disk placed immediately above a roller. The disk puts the roller into motion at a speed of 750 min^{-1} by creating a vertical load of 250 N at a temperature of +20 C. The rotation of the roller shaft axis is limited by a lever. The rolling resistance is calculated on the basis of the moment conditioned by the lever length and force at the lever end.

Figure 2 shows the test values of the overall rolling resistance of a conveyor roller with a diameter of 152 mm at the belt velocity of 6 m/s and temperatures of bearing and ambient environment from +20° to 0°C. Both results demonstrate a decrease in the rolling resistance with time as the temperature of lubricants in the bearing and seal increases.

Theoretical rolling resistance of rollers

Theoretical calculation of rolling resistance of rollers determines each component of the resistance individually. The main components of rolling resistance of rollers are: viscous resistance dependent on lubricant shear inside a labyrinth seal, F_{lab}; friction of balls in bearings during their rotation, dependent on actual loading of roller, F_{brg}; rolling resistance of lip seals, F_{ls}.

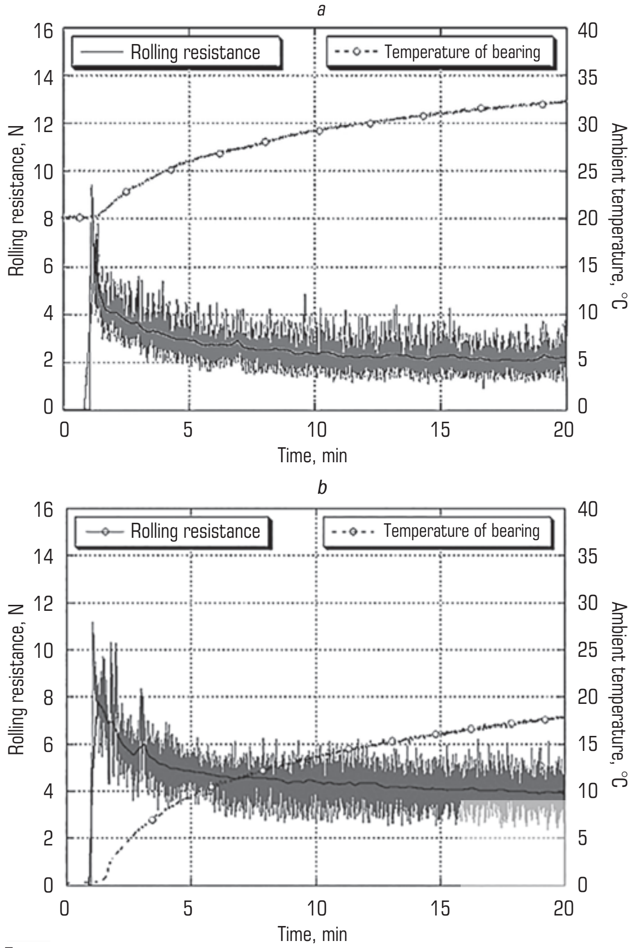

Fig. 2. Relationship of rolling resistance of roller with a dimeter of 152 mm, temperature of a bearing, belt velocity of 6 m/ and ambient temperature of +20 °C (*a*) and 0 °C (*b*)

Let us discuss these components to some detail.

Labyrinth seal of roller with regard to viscous resistance of lubricant

The labyrinth seal of a roller consists of two parts—immobile part set on the roller axis and a rotating part placed in the roller barrel. A typical design of the labyrinth seal of conveyor roller for the analysis of rolling resistance of the roller is shown in **Fig. 3**.

For the analysis of seal work, the scheme can be simplified by withdrawing effects at the edges of seals; thus, the seal structure can be assumed as a series of concentric cylinders. Each layer of the labyrinth has one immobile surface attached to the roller axis and one rotating surface attached to the roller barrel. Figure 3 also shows the forces at the seal and grease interface [25].

The subscripts of the symbols in Fig. 3 relate to the concentric surfaces of seal in line from the center to the barrel of the roller. The roller lubricant is assumed as a quasi-viscous liquid in normal operating conditions, and the force balance for a uniform flat flow is given by:

$$\tau_{yx} = \mu \frac{du}{dy}. \quad (1)$$

The shearing stress τ_{yx}, Pa, in the lubricant depends on its shear rate du/dy, where u is the velocity of the lubricant, m/s; y is the coordinate of the lubricant position. The constant in (1) is the absolute of dynamic viscosity of lubricant — μ. Given that the dynamic viscosity of grease is constant over

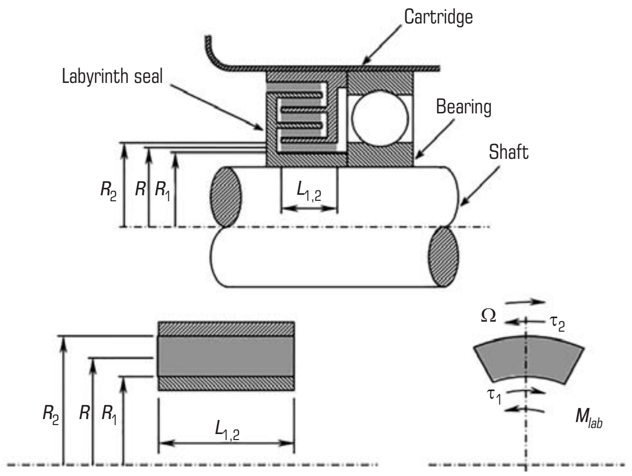

Fig. 3. Typical design of labyrinth seal of conveyor roller for rolling resistance analysis:

M_{lab} – moment of viscous resistance of labyrinth seal, N/m; Ω – spin rate of roller, 1/s; τ_1, τ_2 – shearing stresses in lubricant, Pa [6, 12]; R; R_1; R_2 – average radius and inside and outside radii between the first and second concentric surfaces of labyrinth seal, respectively; $L_{1,2}$ – width of lubricant layer between the first and second concentric surfaces of labyrinth seal; labyrinth seal is a set of two rings forming a non-contact twisting clearance filled with plastic lubricant to protect bearing from aggressive external environment and to extend its service life; cartridge–side cover of roller and bearing body; bearing – bearing assembly as a part of roller support of shaft of a preset rigidity, it fixates spatial position and ensures rotation of roller at the least resistance; shaft – roller rotation axis

the whole labyrinth seal, the overall moment on the roller because of the viscous resistance is:

$$M_{lab} = 4\mu\pi\Omega\left\{\frac{L_{1,2}R_1^2}{1-\left(\frac{R_1}{R_2}\right)^2} + \frac{L_{3,4}R_3^2}{1-\left(\frac{R_3}{R_4}\right)^2} + \ldots + \frac{L_{2n-1}R_{2n}^2}{1-\left(\frac{R_{2n-1}}{R_{2n}}\right)^2}\right\}, \quad (2)$$

where n is the number of the concentric surfaces of the labyrinth seal; $L_{3,4}$ is the width of the lubricant layer between the third and fourth concentric surfaces of the labyrinth seal; R_3; R_4 are the inside and outside radii between the third and fourth concentric surfaces of the labyrinth seal.

The roller rolling force F_{lab}, N, in the labyrinth seal is given by:

$$F_{lab} = \frac{4M_{lab}}{D}, \quad (3)$$

where D is the roller diameter, m.

Friction appears in ball and roller bearings in the zone of contacts of the bearing members as a result of hysteresis as a consequence of low rotation speeds and preliminary sealing of bearings with a plastic lubricant to extend their service life. This lubricant acts as a load transmission that prevents metal-to-metal contact between rolling members of a bearing.

The overall moment of friction in a roller bearing is a sum of moments in the bearing without loading and under loading. The friction loss in a bearing with a plastic lubricant during roller rolling is rather difficult to predict exactly, however, it is proposed to determine approximately the friction moment M_0 without regard to the bearing loading [23]:

$$M_0 = f_0 \cdot 10^{-1} d_m^3 \sqrt[3]{(\nu s)^2}, \quad (4)$$

where f_0 is the coefficient dependent on the bearing design and lubrication method, for single-row ball bearings with a deep groove, this coefficient is 1.5–2.0; d_m is the average diameter of the bearing, found from the equation:

$$\nu s \geq 2000, \quad (5)$$

where ν is the kinematic viscosity of the lubricant (base oil viscosity), usually expressed in centistokes, where 1 CS = $1 \cdot 10^{-6}$ m^2/s; s is the spin rate of the bearing, min^{-1}; and the ratio of the dynamic viscosity μ and kinematic viscosity ν is:

$$\mu = \nu\rho, \quad (6)$$

where ρ is the lubricant density, kg/m^3.

The average diameter of the bearing is found from the equation:

$$d_m = 0.5(d_i + d_0), \quad (7)$$

where d_i is the inside diameter of the bearing and d_0 is the outside diameter of the bearing. The load-dependent friction moment M_1, governed by the sliding resistance and hysteresis resistance of the bearing members, is given by [17]:

$$M_1 = f_1 F_r d_m, \quad (8)$$

where F_r is the radial component of the dynamic load of the bearing, N. The friction coefficient f_1, dependent on the load F is found as follows:

$$f_1 = 0.0008\left(\frac{X_0 F}{C_0}\right)^{0.55}, \quad (9)$$

for the ball bearing with a deep groove of series 62 and as follows:

$$f_1 = 0.0009\left(\frac{X_0 F}{C_0}\right)^{0.55}, \quad (10)$$

for the ball bearing with a deep groove of series 63. Here, C_0 is the nominal static load of the bearing from the manufacturer's specifications, N; X_0 is the radial coefficient dependent on the bearing design, for the single-row ball bearing with a deep groove, $X_0 = 0.6$.

The overall friction moment for each bearing, M_{brg}, can be found by summing up the load-free moment M_0 and loaded bearing moment M_1:

$$M_{brg} = M_0 + M_1. \quad (11)$$

The resistance force of the roller bearing is given by:

$$F_{brg} = \frac{4M_{brg}}{D}. \quad (12)$$

Rolling resistance of roller with lip seal

The outside lip seal generates a barrier for external contaminants to enter the labyrinth seal, while the inside lip seal (if any) contains grease to prevent entry of external contaminants inside the bearing. Most of lip seals for conveyor rollers are made of nitrile rubber or similar mixtures. The outside lip seals are usually attached to the roller axis and contact the outward surface of the labyrinth seals without any lubricant.

The approximation of the rolling resistance of the lip seal uses an equation based on the graphic representation of capacity loss for the contact lip seals [26]. The force F_{ls} resulting from the rolling resistance of the lip seal is:

$$F_{ls} = \frac{0.3e^{2Sd_{ls}}}{D\pi}, \quad (13)$$

where d_{ls} is the lip seal diameter, m; S is the spin rate of the bearing, min^{-1}.

Equation (13) is to be only considered approximately because of a huge variety of designs and configurations of lip seals, to balance the lack of a temperature correction factor taking into account dry or lubricated contact surfaces of the seal [27, 28].

It should be mentioned that the rolling resistance of a conveyor roller is greatly influenced by the type and structural parameters of conveyor belts, especially energy-saving and aramid belts which bring a positive effect [29, 30].

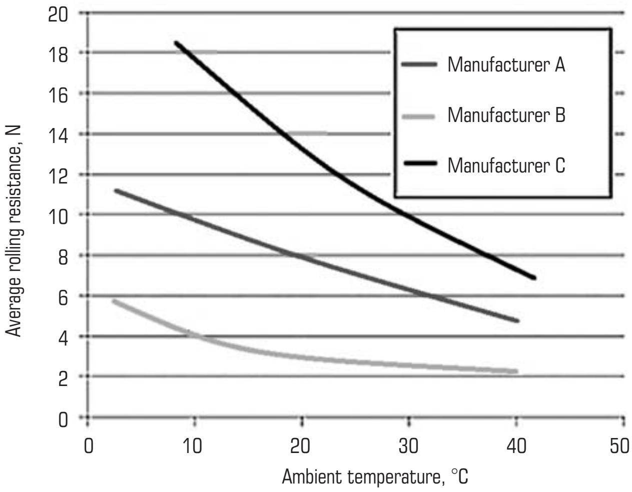

Fig. 4. Average rolling resistance of rollers versus ambient temperature per three different manufacturers

Proposed approaches to design of seals for conveyor rollers

Using the above-presented equations, it is possible to make some recommendations on reducing rolling resistance of conveyor rollers.

Figure 4 depicts the dependence between the average rolling resistance of conveyor roller (without actual load) and the ambient temperature per different manufacturers, and **Fig. 5** demonstrates contribution of each component to the overall rolling resistance.

Lip seal friction and bearing friction, dependent on loading, are assumed to be independent of temperature, while the labyrinth seal and bearing seal without loading depend on dynamic viscosity of lubricant and, consequently, on temperature [5, 6]. The decrease in the rolling resistance with the growing temperature is also evident in Fig. 2.

Figures 5*b* and 5*c* illustrate the relationship between the rolling resistance and roller speed with and without loading; it is seen that the rolling resistance of the lip seal and roller bearing are independent of the roller speed, while the rolling resistance of the labyrinth seal in case of no loading of the bearing depends on the dynamic viscosity of a lubricant and on the roller speed. The load-dependent friction of bearings is the only component influenced by the radial loading given that the roller axis deflections are not excessive [31].

It is seen in Figs. 4 and 5 that, considering possible designs of roller seals, their testing should take into account ambient temperature (in a range of the winter to summer operating conditions), roller speed and actual loading.

If the data on the design variables are unavailable, then the curves in Figs. 4 and 5, and the above-presented equations allow determining presumable design variable. For example, with the available data on loading only (usually smaller than the test data by SANS and Din standards), then the rolling resistance for higher loads can be extrapolated by calculating additional friction dependent on radial loading of bearing from Eq. (8).

Prediction of the effect of the roller speed and ambient temperature is a more complex problem as the base friction in the lip seal and the friction in the bearing without loading are unknown. Therefore, it is necessary to have data on rolling resistance of rollers at the preset speed and with regard to operational temperatures [32].

Conclusions

The theoretical dependences presented in this article are the theoretical framework for the calculation of rolling resistance of conveyor rollers and for the justification of rolling resistance minimization at the design stage.

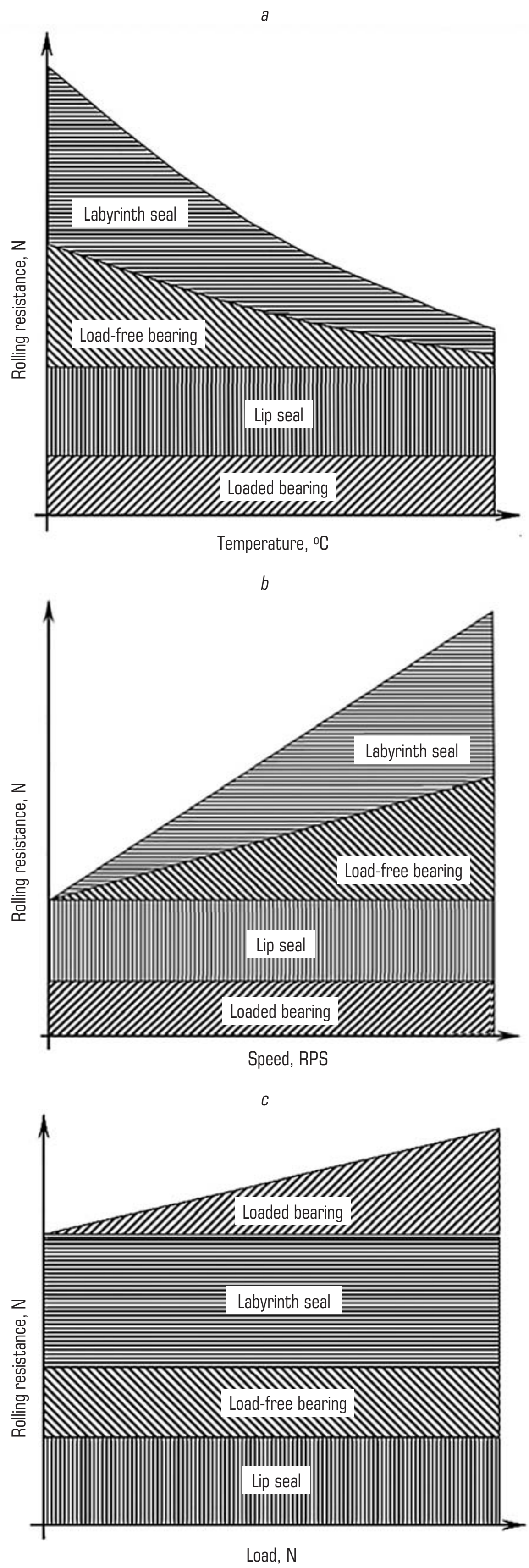

Fig. 5. Rolling resistance of roller as function of: type of bearing seal and temperature (*a*); roller speed with and without loading of bearing (*b*); radial loading and type of seal (*c*)

The main components to influence the rolling resistance of conveyor rollers include: the viscous resistance dependent on shearing of lubricant inside a labyrinth seal, F_{lab}; the rolling friction of bearings depending on actual loading of rollers, F_{brg}; the rolling resistance of lip seals of rollers, F_{ls}.

The authors present a typical labyrinth seal of a conveyor roller for the rolling resistance analysis with regard to the type of a lubricant and the actual load on the roller bearing in a preset range of temperatures in reliable operation of belt conveyor rollers.

The friction force appears in the ball and roller bearings in the contact zones of their members as a result of hysteresis, low spin rates and preliminary sealing of the bearings with a plastic lubricant, as a rule, which extends the service life of the bearing and acts as a load transmission that prevents metal-to-metal contact between the bearing parts.

Since the viscous resistance of labyrinth seals is calculated approximately as it is difficult to take into account velocity fields, the presented theoretical analysis is efficient in design of the labyrinth seals for conveyor rollers. For instance, it is possible to design the labyrinth seals to prevent entry of contaminants at the concurrent decrease in the viscous resistance owing to minimal gaps between immobile and rotating surfaces of the seals, which are arranged closer to the roller axis to reduce the torque.

Operation of belt conveyors can benefit from the use of rollers of larger diameter, which leads to reduction of their speed and rolling resistance in addition to the increased fatigue life of bearings.

It is important to reduce rolling resistance of conveyor rollers as it affects their reliable operation. It is also recommended to determine efficiency of roller seals by the parameter of contamination, which can initiate preliminary failure of bearings.

The value of the rolling resistance of conveyor rollers is greatly influenced by the type and structural parameters of conveyor belts, especially energy-saving and aramid belts which provide a positive effect.

Owing to the comprehensive analysis of rolling resistance of conveyor rollers in the study, the theoretical models in combination with the theoretical dependences can be used in prediction of the overall rolling resistance of rollers of energy-efficient belt conveyors.

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