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LOADING TESTS OF RETAINING TRICONE BIT BEARING USING PHYSICAL MODEL

Tricone bits are currently less commonly used as compared with PDC bits. Tricone bits are mainly used in drilling in difficult geotechnical conditions and in blasthole drilling in open pit mines. The main disadvantage of tricone bits is the relatively short life. The main cause of the age limit is failure of the retaining ball bearing due to destruction of raceway and journal shoulder under high contact and bending stresses. Based on the provisions of the similarity theory, the physical model of the retaining drill bit bearing has been manufactured. The paper presents the data of the retaining bearing testing using a friction and wear testing machine, namely, the temporal variations in temperature, frictional torque and clearances of bearings. The temperature and the heat rating of the bearing are related. In the extreme conditions of operation at deficiency of lubrications, the contact stresses in the rings and balls of bearing, as well as their temperatures exceed greatly the allowable values. The authors propose using the procedure in engineering of a roller drill bit.

Keywords: *roller drill bit, bit bearing, physical model, wear testing, triboengineering, ball bearing*

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Introduction

Reliability improvement of roller cone bits is a challenging problem in view of high energy transmission through the drilling tool for rock breaking in geometrically limited zone at bottom hole. In such conditions, it is difficult to create a bearing conformable with designs standards of loading, cooling, lubrication and safety at the required strength reserve and sufficient roller bit cutting structure. In soft and medium-hard abrasive rock drilling, the main cause of failure of bearing mount assembly is the abrasive wear of raceways. In hard rock drilling, the failure cause is pitting of raceways, which leads to an essential change in the stress concentration factor and to the loss in fatigue strength of shoulder of retaining drill bit bearing. Roller drill bits feature complex manufacturing and, thus, high cost. Replacement of drill bits in deep drilling takes much time. On this basis, it is of the current concern to actuate studies aimed at design improvement of roller drill bits. Many researchers have addressed the design improvement of roller drill bits and their availability enhancement [1–6].

It is very difficult to judge on the work environment of a drill bit when it operates at a depth from tens meters to a few kilometers. One of the earliest attempts to perform theoretical research of the running regime of drill bits was undertaken by S. M. Kuliev, G. G. Gabuzov, B. I. Esman and A. G. Mdivani [7]. The authors analyzed the heat-balance equation of blade drill bit at bottom hole. N. N. Zakirov found that the maximum temperature in friction assemblies of drill bits reaches 270–379 °C [8]. It is suggested that the temperature field expands in the body of roller drill bits wavily, with wave peak at the middle crown of the roller bit, which occurs in the best conditions of cooling provided by side waterway [9]. The implemented research shows that the bit temperature increases with growing temperature of circulating fluid and intensity of friction and with decreasing thermal conductivity of metal. The temperature depends on the axial load of the bit, its rotation speed and friction factor, on physical and mechanical properties of rocks and the bit material, on the failure mechanism of rocks (shearing, crushing, cutting), roughness of contact surfaces and the bit diameter.

The numerous factors to condition the burden and temperature of bearings complicate the relevant analytical dependence to be found. Thus, physical simulation of burden on bearings of a tricone bit is of interest at this time.

Results and discussion

Operating practice of roller drill bits revealed a number of causes of their failure: 1) wear of drill bit cutting structure, which essentially decelerates drill penetration rate; 2) pitting of raceways and balls [7]; 3) abrasive wear of bearing components, which leads to their jamming under conditions of semi-dry and dry friction [10]. Eventually, emergency failure of whole drill bit takes place [6]. With this end in view, it is proposed to equip roller drill bits with circulation-system lubrication of bearings [11]. Still, this is an incomplete solution of the problem. Heating of bearings in operation and insufficient cooling reduces viscosity of lubricants, changes the size of clearances in gaskets and, consequently, ends with leakage of lubricants. Dry friction arises between balls, which essentially increases the rate of failures and, accordingly, decreases service life of bearings. Such failure scenario is most typical of roller drill bits with open bearings, meant for drilling with air cleaning of bottom hole.

Design calculations of drill bits are presented in [8, 9, 12–14]. The known procedures for roller bearing neglect the factor of temperature while it essentially limits contact endurance and reduces loading cycles. On the other hand, the experimental check of these procedures was partly unsuccessful.

This paper authors tried to attempt to eliminate these arrears of research. Based on the main provisions of similarity theory, the physical model of retaining roller drill bit bearing has been constructed. A full-scale specimen was chosen as tricone bit III-244,5K-PV (9 5/8 IADC 742) with roller–ball–roller bearing. The axial load per one bit, with regard to

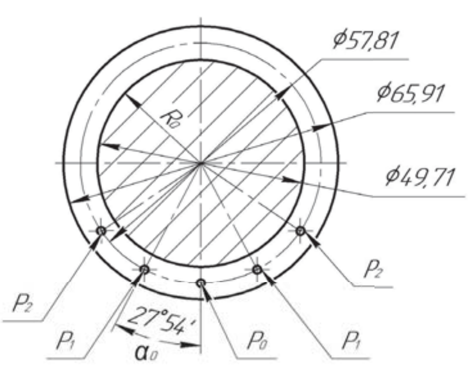

Fig. 1. Loads applied by balls to raceway of bearing journal

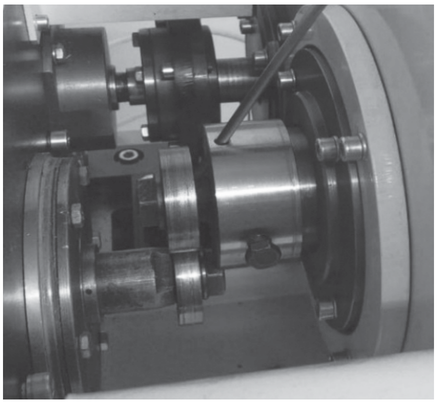

Fig. 2. Friction and wear testing machine II 5018

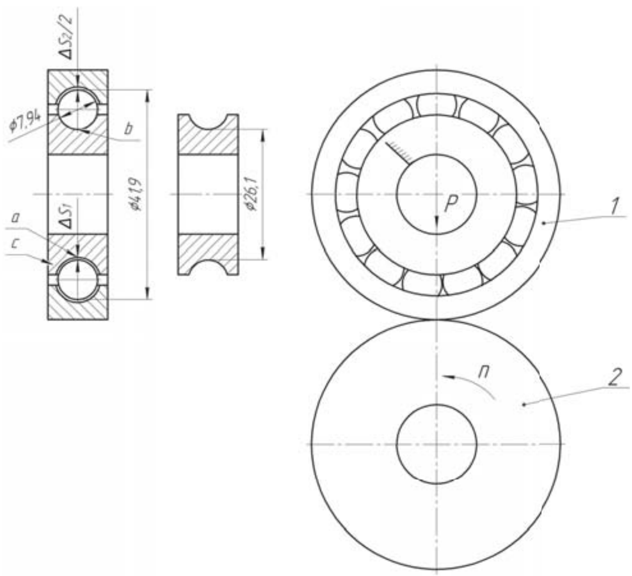

Fig. 3. Model bearing set on friction testing machine II 5018
a – model bearing (a, b, c – hardness measurement points); *b* – installation diagram (*1* – model bearing, *2* – roller)

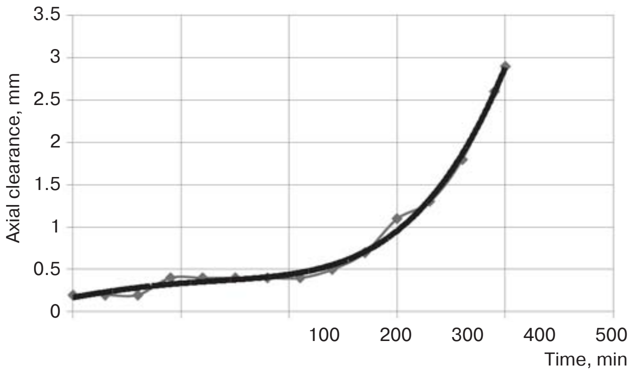

Fig. 4. Axial clearance – time curve

dynamic force, was F_{ax} = 83 kN; geometrics of the bit components was taken from specifications.

Figure 1 shows the load distribution on retaining roller drill bit bearing. The maximal load is taken by the bottom ball – P_0. The loads on the other balls are calculated from the formulas:

$$P_1 = P_0[\cos(\alpha_0)]^{3/2};\ P_2 = P_0[\cos(2\alpha_0)]^{3/2}, \quad (1)$$

where α_0 is the central angle of neighbor balls. These forces are also random values.

Aiming to prove relevance of simulation model [15], we designed and manufactured physical model of retaining drill bit bearing.

The basic condition of similarity between the physical model and full-scale specimen is the equality between elasticity moduli of their materials at the same surface and core hardnesses, as well as the equality of stresses and, accordingly, endurances of parts. Using Hertz's formula [16], it is possible to correlated geometric parameters of bearing (radii of raceway and balls) and load of balls.

Thus, at the known axial load per on roller bit, with regard to dynamic force and in compliance with specifications of geometric sizes of components within tricone bit III-244,5K-PV, the model parameters are: N_{mo} = 2.0 kN; r_{shm} = 3.97 mm; r_{sh2m} = 4.1 mm; r_{sh1m} = 13.05 mm. The rings of the model bearing are made of steel 20KHN3MA. Heat treatment processes is temper hardening: balls – HRC = 38÷40; bearing rings – HRC = 57÷62. Given such parameters, the maximal contact stresses in the raceway and balls will be equal in the model and at full scale.

Figure 2 shows friction and wear testing machine II 5018. The machine allows recording revolutions of samples, loads, temperature and friction moments.

The bearing model (**Fig. 3a**) is arranged on an immobile shaft meant to transmit the force *P* (**Fig. 3b**). The external ring was rotated via roller 2 mounted on the main shaft. The test conditions fitted the parametric model: force *P* = 2 kN, revolution number *n* = 1000 min^{-1} of a sample conformed with the rotation speeds of 100 min^{-1} of drill bits. During the tests, we recorded the friction moment on the main shaft, radial load on the bearing, temperature of the internal ring of the bearing, axial ΔR and radial ΔS clearances between the ball and the ring of the bearing. For the clearance measurements, the main shaft was stopped every 60 min. there was no abrasive, and lubricant was not refilled.

Figure 4 depicts the found time dependence of the axial clearance.

The regression equation is given by:

$$\Delta R = 2E{-}10x^4 - 6E{-}08x^3 - 2E{-}06x^2 + 0.0022x + 0.17. \quad (2)$$

Coefficient of determination R^2 = 0.99 proves essential connection between the axial clearance and operation time.

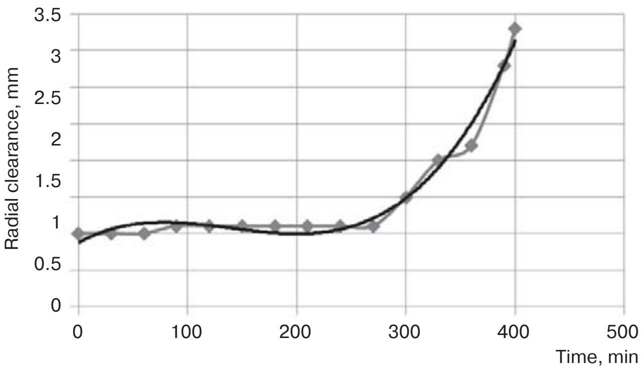

Fig. 5. Radial clearance–time curve

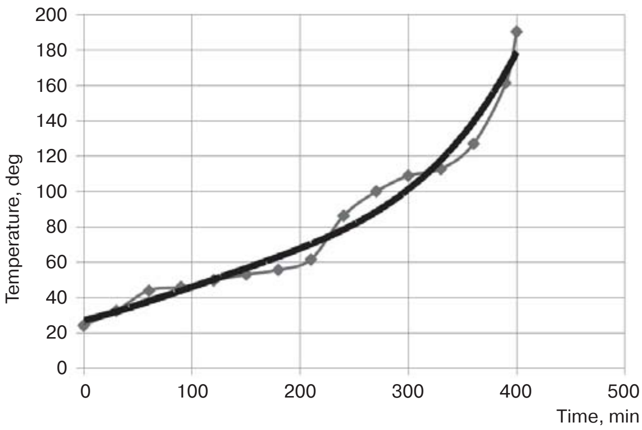

Fig. 6. Time–temperature dependence in bearing

The time dependence of the radial clearance is depicted in **Fig. 5.** The growing clearance with an increase in wear is contributed to by the outside ring wear ΔS_{out}, inside ring wear ΔS_{in} and ball wear ΔS_b. The wears ΔS_{out} and ΔS_b are symmetrical, while ΔS_{in} is one-sided, localized on the loaded (bottom) side of the ring (Fig. 3a).

As per State Standard GOST 20692-2003 Roller Drill Bits. Specifications, for the type of drill bit under analysis, the radial motion variation of roller bits relative to the axis of thread is not more than 1.2 mm.

The regression equation is given by:

$$\Delta S = 2E - 07x^3 - 7E - 05x^2 + 0.0081x + 0.876. \quad (3)$$

Coefficient of determination $R^2 = 0.99$ proves essential connection between the axial clearance and operation time.

Figure 6 illustrates the time–temperature dependence of bearing parts.

The regression equation is:

$$t = -4E{-}09x^4 + 1E{-}05x^3 - 0.0039x^2 + 0.56x + 22.1. \quad (4)$$

Sufficiently high coefficient of determination $R^2 = 0.97$ proves the stable time–temperature connection.

Figure 7 demonstrates the time history of the moment on bearing.

The regression equation is:

$$M = 1E{-}07x^3 - 5E{-}05x^2 + 0{,}0095x + 0{,}19. \quad (5)$$

Sufficiently high coefficient of determination $R^2 = 0.96$ proves the connection between the time and moment.

Hardness was measured at the points *a*, *b*, *c* (Fig. 3, **Table**). It is found that the hardness lowers from the initial value of 61 HRC to 41 HRC on the worn surface of raceway of the bearing cone, on the loaded outer side, which implies wear of the carburized case.

After analysis of Figs. 4–6, we have drawn some conclusions as follows

1. The shapes of the curves coincide, which proves interconnection of wear (radial and axial) and temperature with operation time.

Hardness measurements

Point	HRC
a – after tests, wear of bearing cone	41.5
b – before and after tests, unloaded side of inner ring	61.7
c – at section, inner ring core hardness	44.9

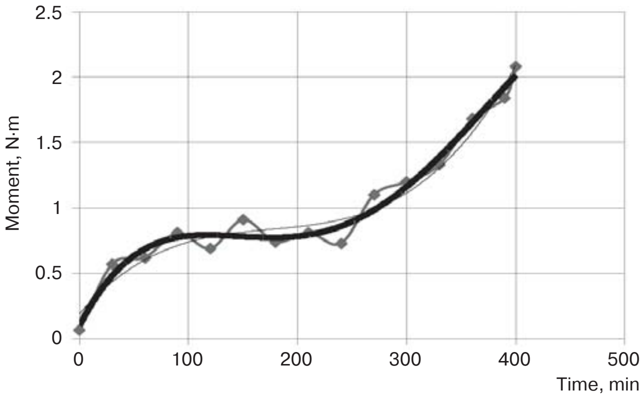

Fig. 7. Variation in moment on bearing shaft as function of time

2. The curves have the classical time-related branches: first—breaking-in of the bearing parts; second—normal operation at unvaried wear and temperature; third—intolerable (catastrophic) increase in wear and temperature. The bearing fails as a result.

3. The normal operation period is clearly deficient.

All this is explained by the difficult work environment of bearings, first of all, by high loading which leads to relatively intense heat liberation. The latter induce degradation of lubrication properties, and lubricants start leaking from the bearing through the gasket. The bearing retaining goes to an operation mode without lubricant, which increases the friction force and, consequently, heat liberation. The bearing transfers to the phase of catastrophic wear. In real-life conditions, it is also possible that an abrasive (chips) enter inside the bearing, which essentially accelerates wear.

Figure 8 offers post-testing pictures of bearing parts.

The pictures in Fig. 8 show the traces of pitting, which means the contact stresses have greatly exceeded the allowable value. After testing, some balls have sites of wear, which implies dry friction between the rolling bodies.

The measured temperatures agree with the calculations [17]. It is experimentally found that the bearing temperature depends on the heat rating. **Figure 9** depicts the variation in the temperature as function of the heat rating.

The regression equation is given by:

$$t = 0.0022x^2 + 0.32x + 15.8 \quad R^2 = 0.95. \quad (6)$$

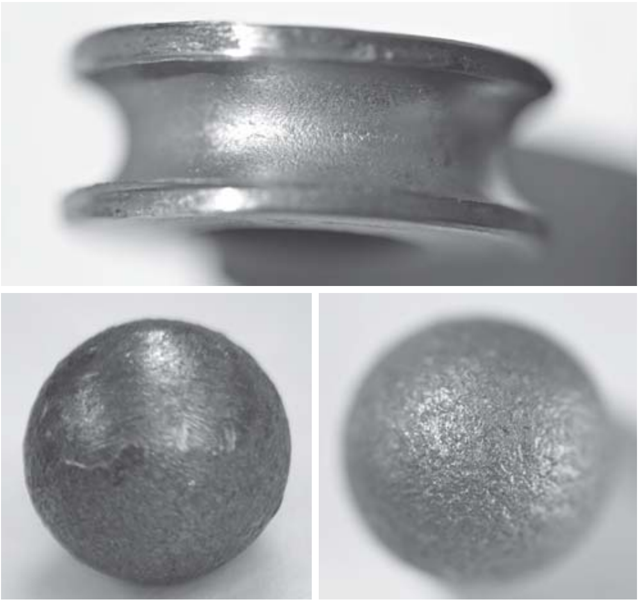

Fig. 8. Inner ring and balls of bearing after testing

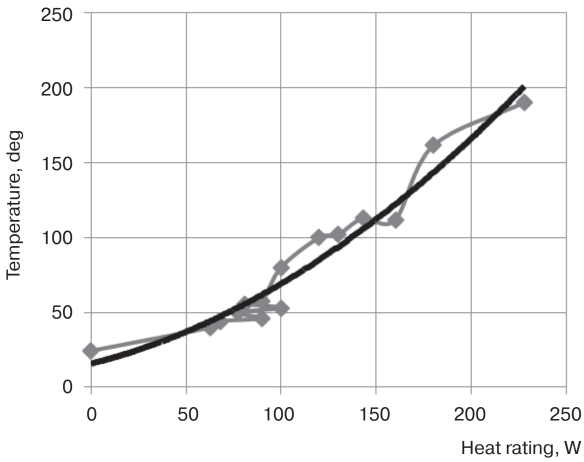

Fig. 9. Temperature of bearing versus heat rating

The determination coefficient R^2 of 0.95 proves an essential connection between the temperature and heat raring.

The connection between the steady-state temperature and heat liberation rate was found in [18]. We have calculated the heat liberation rate (specific energy flow) on the inner ring of the model bearing. The specific energy flow on the inner ring of the bearing is found from the formula:

$$q = N_1 S^{-1}, \quad (7)$$

where N_1 is the energy emitted on the inner ring of the bearing per unit time, W; S^{-1} is the area of heat liberation on the inner ring, mm^2.

Heat liberation takes place on the site between the balls which take up loads. The length of this site is equal to the length of the arch between the loaded balls, and the maximum width of the site equals the longer axis of the stress ellipsoid. From Hertz's theory, the longer axis a of ellipsoid is evaluated using the formula [17]:

$$a = \{1.5Q/[\pi(\sigma^* k)]\}^{0.5}, \quad (8)$$

where k is the coefficient governed by the ball radius, raceway radius and raceway cross radius.

As a result of the testing and calculation, the specific energy flow on the inner ring of the model bearing is 2.8 W/mm^2 and is 2.9 W/mm^2 in the steady run. In the limit conditions of operation, the specific energy flow is 4.6 and 8 W/mm^2 without and with lubricants, respectively, which agrees with the theoretical data for the inner ring of a bearing in [7] – 2.5 W/mm^2.

Conclusions

Finally, we have drawn some conclusions listed below.

1. The contact stresses in rings and balls of bearing essentially exceed the allowable values.

2. The test value of the specific energy flow agree with the theoretical data in [7].

3. The temperature of bearing essentially exceeds the allowable value, which leads to leakage of lubricants and to operation in dry mode.

4. The studies into durability of bit bearing can determine efficient values of hardness of working faces, rational conditions of loading and adjusted design variables.

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References

1. Huang Z., Li G. Failure analysis of roller cone bit bearing based on mechanics and microstructure. *Journal of Failure Analysis and Prevention*. 2018. Vol. 18, No. 2. pp. 342–349.
2. Pialchenkov V. A. Methods of study of the load degree of equipment and cone drill bit bearings. *Oil and gas studies*. 2015. No. 1. pp. 88–95.
3. Pialchenkov V. A., Pialchenkov D. V., Dolgushin V. V., Kulyabin G. A. Studying load of the cutting structure of rolling cutter bit depending on the errors of its manufacture. *Oil and gas studies*. 2019. No. 1. pp. 113–120.
4. Bogomolov R. M. Analysis of techniques of parts production and rolling-cutter drill bits assembling and their influence on the tool operational capacity. *Equipment and technologies for the oil and gas complex.* 2020. No. 4. pp. 8–12.
5. Serdyuk N. I., Blinkov O. G. Simulation of a slide bearings failure in a drill-bit bearing structure. *Equipment and technologies for oil and gas complex.* 2017. No. 2. pp. 4–7.
6. Pelfrene G., Stab O., Tilleman D., Gallifet T. et al. Model-ling the 3D Bit-Rock Interaction Helps Designing Better PDC Bits. *In SPE/IAD International Drill-ing Conference and Exhibition.* 2019. DOI: 10.2118/194134-MS
7. Zhidovtsev N. A., Kershenbaum V. Ya., Ginzburg E. S., Bikbulatov I. K. et al. Longevity of rolling cutter. Moscow : Nedra, 1992. 272 p.
8. Zakirov N. N. Technikal and technological elements of life cycle of roller bits. Tyumen : TIU, 2018. 140 p.
9. Slipchuk A., Kuk A. Evaluation of the permissible moment in a roller cone drill bit providing the prescribed reliability of work. *Ukrainian Journal of Mechanical Engineering and Materials Science.* 2018. Vol. 4, No. 1. pp. 116–124.

10. Shevchenko A. N., Krasnoshtanov S., Ruuz M. V., Perfilev V. Analysis of the roller bit lubrication system. *IOP Conference Series: Earth and Environmental Science.* 2020. Vol. 408, No. 1. 012061.
11. Simisinov D., Afanasev A., Adas V., Simisinov A. Lubrication system of a roller cone bit. *MATEC Web Conf*. 2020. Vol. 329. DOI: 10.1051/matecconf/202032903006
12. Han C., Yu C., Li Y., Yan J. Mechanical performance analysis of hollow cylindrical roller bearing of cone bit by FEM. *Petroleum*. 2015. Vol. 1, No. 4. pp. 388–396.
13. Shigin A. O., Gilev A. V. Methods to calculate fatigue strength as a major factor of roller bit durability. *Vestnik Irkutskogo gosudarstvennogo tekhnicheskogo universiteta*. 2012. No. 3(62). pp. 22–27.
14. Schroder J., Pasquale M. D., Richards A. Bearing innovations extend roller-cone bit life. *Oil and gas journal*. 2016. Vol. 114, No. 6. pp. 50–55.
15. Simisinov D.I., Afanasiev A.I., Shestakov V.S., Valiev N.G. Loading of bearing retaining pin of tricone drill bit. *Gornyi Zhurnal*. 2020. No. 12. pp. 64–66. DOI: 10.17580/gzh.2020.12.14
16. Belyaev N. M., Strength of materials. Moscow : Nauka, 1965. 856 p.
17. Dobrovolsky V. A., Zablonskiy K. I., Mak S. L., Radchik A. S. et al. Parts of machines : Textbook. 7th enlarged and revised edition. Moscow : Mashinostroenie, 1972. 502 p.
18. Afanasev A. I., Kosenko E. A., Suslov D. N. Results and methods of thermal tests of linear motor of a screen-feeder. *Izvestiya vuzov. Gornyi Zhurnal*. 2011. No. 7. 2011. pp. 106–109. EM

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THE MAIN DIRECTIONS OF INCREASING THE OPERATIONAL EFFICIENCY OF HIGH PRODUCTIVE BELT CONVEYORS IN THE MINING INDUSTRY

When transporting bulk cargo at long distances, modern high productive belt conveyors must be equipped with energy-saving conveyor belts, which can reduce the energy consumption of the belt drive by 28%, and reduce the negative impact on the environment, including with ESG principles. Formula is proposed to determine the mass of emissions of harmful substances into the atmosphere - carbon dioxide, nitrous oxide and solid particles depending on the type of fuel burned to generate electricity. The use of supporting rollers on conveyors with a shortened middle roller, as well as with lower resistance coefficients to the rotation of the rollers, allows an additional 5-7% energy savings. The issue of saving energy consumed by the belt conveyor by optimizing the control and monitoring of the belt speed, as well as the processes of starting and braking the conveyor belt, is considered. Installation of gearless drives equipped with synchronous motors at high productive belt conveyors of long length allows to increase the efficiency of the drives, as well as to eliminate the work of gearboxes, starting couplings, which significantly reduces capital and operating costs.

Keywords: *conveyors, energy saving belts, belt cover, energy consumption, carbon dioxide, emissions, atmosphere, rolling resistance, idlers, belt speed, drive, gearless drives, synchronous motors, stator, rotor, efficiency*
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Introduction

Recently, for the transportation of bulk cargo at distances of 1000 m or more, belt conveyors of various types are used: rectilinear, tubular belt, steep incline belt, as well as belt conveyors with curves along the route, which are used most often because of the ability to transport minerals without overloading at distances up to 28 km, in one rate with a capacity of 15.000 t/h, and more [1, 2]. Conveyors of this scale are quite energy-intensive, therefore directly or indirectly they negatively affect the environment.

In accordance with this, the article is devoted to the analysis of the market of developments that make it possible to increase the operational efficiency of high productive conveyor belts through the use of energy-saving conveyor belts, innovative design and technological developments supporting idlers, speed control of the belt, as well as gearless drives with synchronous motors.

Consideration of the issue

The use of energy-saving belts not only reduces the consumption of electricity by the conveyor, and the amount of fuel burned for its production (coal, oil, gas, nuclear fuel), but also reduces the amount of carbon dioxide (CO_2) emissions into the atmosphere, and harmful particles of combustion products – nitrous oxide (NO_x) and solid particles (S_p).

To study the energy-saving properties of the conveyor belt in Germany, a special test and stand were developed, described in the standard [3, 4], representing a method for determining the rolling resistance of conveyor belts by supporting rollers, taking into account the width of the belt, the