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THERMAL MODEL OF CONE DRILL BIT BEARING WITH AIR FLUSHING

Cone drill bits are used in various-purpose drilling in difficult geotechnical conditions, as well as in drilling-and-blasting in open-pit mining. The reliability of the drill bits depends on the rock-breaking source and on the bearing support. One of the causes of low reliability of a drill bit is the failure of the retaining ball bearing due to destruction of roller path and journal of the bearing under high contact and bending stresses. Based on the general provisions of similarity theory, a physical model of the retaining bearing of a roller bit was created. The test data of the bearing on a friction and wear testing machine are presented as the dependences of the bearing temperature, friction moment and clearances on time. The dependence of the bearing temperature on the thermal power is determined. In extreme conditions of lubricant shortage, the contact stresses in the bearing rings and balls and the temperature significantly exceed the permissible values. The methodology is proposed to be used in development of a cone rock bit drilling tool.

The article discusses the method of determining the moment of friction of the bearing of a cone drill bit. The design of bearings without retainers features increased clearances. To determine the moment of friction under these conditions, a simple design of the test bench is proposed, which includes a load and a counterweight connected by a thread thrown over a rolling bearing. The calculations to determine the reduced moment of friction of the rolling bearing in the conditions of the test bench are described.

Keywords: *roller cone bit, drilling bit bearing, thermal conductivity of support elements, heat transfer in leg, ball and roller bearings*

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Introduction

The relatively low penetration rate per drill bit is governed by the difficult operating conditions of the tool as it is to transfer considerable energy to break rocks at bottomhole under essential geometrical constraints. In connection with this, it is rather difficult to provide reliable bearings for drill bits to meet the common standards to ensure the wanted trouble-free life. The main cause of failure of bearing assemblies in drilling in soft and medium-strength rocks is the abrasive wear of roller path as a consequence of bleeding of lubricant because of its heating and due to loss of tightness of bearing cavity. In hard rock drilling, the cause of failures is roller path pitting under high contact stresses. Roller bits are comparatively expensive as they are difficult-to-make, and need special equipment and high-quality constructional materials, including hard alloys. Replacement of drill bits consumes much time and high material inputs. Thus, it is relevant to undertake the research aimed to improve the design of roller bits to ensure their high penetration rate. Many researchers addressed the design development and performance improvement of the roller drill bits [1–6].

S. M. Kuliev, G. G. Gabuzov, B. I. Esman and A. G. Mdivani determined theoretically the temperature of the drill bit bearings [7]. To this effect, they used the heat-balance equation of bottomhole operation of a drill bit. N. N. Zakirov found that the maximum temperature in the friction assemblies of a drill bit may reach 380 °C [8]. These studies show that the heating temperature of drill bits rises with the increasing temperature of the flush fluid, with the higher intensity of friction and with the decreasing thermal conductivity of metal. It is found that the temperature depends on the axial load of a drill bit, its rotation speed, friction factor, physical and mechanical properties of rock, material of the drill bit, rock fracture mechanism (shearing, crushing, cutting), roughness of the contact surfaces and on the bit diameter.

These influence factors of the heating temperature of bearings greatly complicate their analytical correlation. For this reason, it is topical to study thermal burden of a tricone drill bit using an electrothermal analogy.

Table 1. Heat transfer coefficient versus air velocity

No.	Air velocity, m/s	Heat transfer coefficient, W/(m^2 °C)	Average value of α_{ht}	Calculated value of α_{htc}
1	0	15; 18; 22; 25 26; 18	20.5	19.7
2	2	41; 43; 40; 38; 39; 39; 40	40	40.4
3	4	58; 53; 52; 57; 55; 55	55	55.8
4	6	72; 70; 68; 66; 64; 62	68	66.8
5	8	77; 78; 74; 73; 73; 75	72	72.3

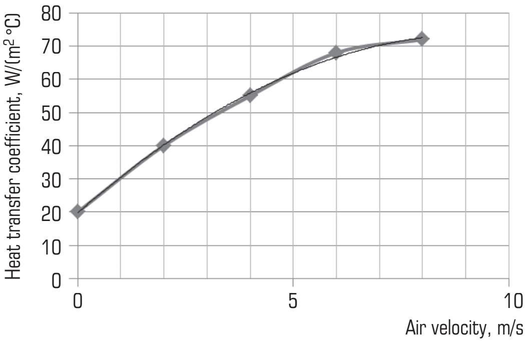

Fig. 1. Heat transfer coefficient versus air velocity *V*

The subject of research is a tricone drill bit. The research aims to obtain data to improve the ball-and-roller bearing design of the drill bits. This study used the methods of mathematical modeling, regression analysis, as well as the common logic techniques (analysis and generalization of scientific and technical information, patent information, induction, synthesis, etc.).

Results and discussion

The operating practice of roller drill bits revealed a few causes of their failures and, consequently, of their penetration rate drop: 1) wear and damage of cutting structure; 2) pitting of roller paths [1]; 3) abrasive wear of balls, rollers, and raceways of bearings, resulting in growth of clearances, bleeding of lubricants and jamming [1, 6], causing emergency failure of the tool [6].

The studies into designs of roller drill bits [9–15] neglect the temperature factor while it essentially decreases the trouble-free life of roller and roller paths. The authors made an attempt to correlate temperature and load on drill bit model III-244.5K-PV (9 5/8 IADC 742) with a ball-and-roller bearing design.

Drill bit bearings lack cooling air while operating. The bearings intensively emit heat dependent on the equivalent moment of rolling friction. The drill bit bearings have no retainers, and, because of mutual friction of rolling bodies, the friction moment in such bearings is higher than in some other type bearings. The present study authors experimentally found that the equivalent coefficient of rolling friction in bearings without retainers and with lubricant ranged as 0.55–0.65 mm. In case of loss of the lubricant, the coefficient of rolling friction doubles, which stimulates heat emission accordingly. In this manner, the moment of friction in rolling bearings of roller drill bits is given by:

$$M_i = aN_i, \qquad (1)$$

where N_i is the certain bearing reaction, N; a is the equivalent coefficient of rolling friction at a certain bearing, m.

The steady temperature of the drill bit bearings depends on their heat transfer to air. The temperature of the bearing mostly depends on the heat transfer of the flat surface of the bearing journal.

Many studies [16–18] find out that, all other things being equal, the heat transfer coefficient α_{ht} of a heated body in air depends on the velocity of air flow around this body and on its typical size. For the Reynolds number less than 10^5, the heat transfer coefficient is determined using a Nusselt number. With this end in view, we made a thermal model of a bearing leg as a flat steel wall supplied with heat from one side. From the other side, the wall is insulated. Knowing the incoming heat per unit time (q, W), the wall area (S, m^2), the steady temperature of the wall (t_w) and the air temperature (t_a), it is possible to find the heat transfer coefficient:

$$\alpha_{ht} = q/[S(t_w - t_a)]. \qquad (2)$$

Each test was repeated several times.

The implemented experiments allowed correlating the heat transfer coefficient of a flat vertical surface and the air velocity (**Table 1**). The ambient temperature in the tests was 20 °C.

Figure 1 depicts the heat transfer coefficient as function of the air velocity. This relationship agrees with the other research findings [16–18].

The regression equation is given by:

$$\alpha_{ht} = -0.607V^2 + 11.46V + 19.7. \qquad (3)$$

The relatively high correlation ratio $R^2 = 0.99$ is reflective of an essential connection between the test parameters.

Equation (3) adequately represents a real-life process as the air velocity changes from 0 to 10 m/s.

For a drill bit equipped with three main parallel heat-emitting components—three radial bearings, the thermal conditions conform with the diagram in **Fig. 2**.

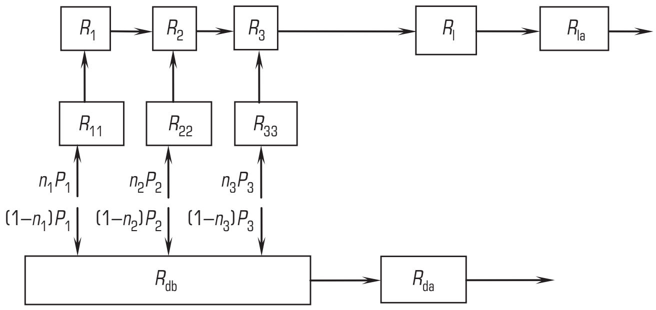

Fig. 2. Equivalent diagram of heat flow propagation in roller drill bit

In Fig. 2: $(1-n_1)P_1$, $(1-n_2)P_2$, $(1-n_3)P_3$ are, respectively, the heat flows from the first radial bearing, retaining bearing and the second radial bearing to the drill bit and to the environment; n_1P_1, n_2P_2, n_3P_3 are the heat flows from the first radial bearing, retaining bearing and the second radial bearing to the journal, leg and to the environment.

The equivalent calculation model of the heat flow in a drill bit uses the known method of the electrothermal analogy and includes some assumptions, namely:

1) the roller bit body and the journal are assumed as a uniform and isotropic body with the constant heat transfer coefficient at all points;

2) the heat transfer coefficient of the roller drill bit and leg to the cooling medium is assumed to be constant.

It worth mentioning that due to an essential difference between the heat transfer areas, the thermal resistance of the bearing journal and leg is much more than the thermal resistance of the roller drill bit body. The heat sharing ratios n_i along the journal are constant as the bearings are made of the same materials and the heat transfer conditions are also the same. For this reason, the heat balance equation is given by:

$$\theta o = n[R_{11}P_1 + R_1P_1 + R_{22}P_2 + R_2(P_1 + P_2) + R_{33}P_3 + R_3(P_1 + P_2 + P_3) + (R_l + R_{la})(P_1 + P_2 + P_3) + (R_{db} + R_{da})(P_1 + P_2 + P_3) = (R_{db} + R_{da})(P_1 + P_2 + P_3), \qquad (4)$$

where θo is the temperature difference between the most heated part of the journal and the environment, K; P_1, P_2, P_3 are the heat flow capacities

Table 2. Maximal temperatures in roller drill bit bearing

No.	Axial loading, kN	Rotation speed, min^{-1}	Heat transfer coefficient, W/(m^{2}K)	Excessive temperature, deg
1	43	60	40	152
2	80	60	40	300
3	43	90	40	310
4	80	90	40	600
5	43	60	80	87
6	80	60	80	174
7	43	90	80	180
8	80	90	80	347

in the first radial, retaining and the second radial bearings, respectively, W; R_1 is the thermal resistance of the journal between the front and the first radial bearings in axial heat flow; R_{11} is the thermal resistance of the journal in heat flow from the first radial bearing across the axis; R_2 is the thermal resistance of the journal between the first radial and retaining bearings in axial heat flow; R_{22} is the thermal resistance of the journal in heat flow from the retaining radial bearing across the axis; R_3 is the thermal resistance of the journal between the retaining bearing and the leg bedding point in the axial heat flow; R_{33} is the thermal resistance of the journal in heat flow from the second radial bearing across the axis; R_l is the thermal resistance of the leg between the second radial bearing and the hole in the axial heat flow; R_{la} is the thermal resistance in the heat flow from the leg to air; R_{db} is the thermal resistance in the heat flow from the inside to the outside of the drill bit; R_{da} is the thermal resistance in the heat flow from the drill bit outside to cooling air.

In drill bit model III-244.5K-PV (9 5/8 IADC 742) with the roller–ball bearing design, the thermal resistances are, (K/W): $R_{11} = 0.155$; $R_1 = 0.95$; $R_{22} = 0.22$; $R_2 = 0.29$; $R_{33} = 0.18$; $R_3 = 0.36$; $R_l = 0.09$; $R_{li} = 0.96$; $R_{db} = 0.03$; $R_{da} = 0.96$.

The thermal resistances in the journal were determined from the known formulas [16–18].

The lost heat power in the first and second radial bearings:

$$P_r = N_r a_r \omega, \tag{5}$$

where N_r is the radial force in the roller bearing, N; a_r is the equivalent rolling friction coefficient in the roller bearing, m; ω is the angular speed of the bearing, s^{-1}.

The lost heat power in the retaining bearing is:

$$P_{rb} = N_{af} a_{rb} \omega, \tag{6}$$

where N_{af} is the axial force in the retaining bearing, N; a_{rb} is the equivalent rolling friction coefficient in the retaining bearing, m.

Table 2 compiles the calculated data on the maximal temperature in the bearing of tricone roller drill bit under different axial loading, rotation speed and heat transfer coefficient.

The relation between the steady excessive temperature in the bearing, its axial loading, rotation speed and heat transfer coefficient (in terms of coded variables X_1, X_2, X_3) is given by:

$$\Theta_0 = 269 + 86.5X_1 + 90.8X_2 - 72.9X_2 + 27.8X_1X_2 - 22.6X_1X_3 - 24X_2X_3 - 7.7X_1X_2X_3. \tag{7}$$

The coded variables X_1, X_2 and X_3 may assume the values from the range $-1 \le X_i \le +1$.

The calculation results agree with the data presented in [19].

Conclusions

1. The heat transfer coefficients in the bearing and leg depend on the air flow velocity around them and range from 20 to 80 W/(m^{2}K).

2. In air flush of a drillhole, the thermal resistances in the drill bit–air and leg–air systems grow. The change in the heat transfer coefficient induces the change in the heat shearing ratio.

3. The method of the electrothermal analogy allows determining the excessive temperature in the bearings of roller drill bits as function of operating conditions and heat transfer coefficient at design stage using Equation (7).

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