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MODELING OF LAWS GOVERNING HEAT TRANSFER IN SLIDING BEARINGS

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A technique for determining temperatures arising in sliding bearings during operation is presented. Results of the calculation of the level of temperatures in the bearing by the suggested mathematical model depending on the operating conditions and results of their experimental verification are given. The difference between them is slight.

An analysis of heat-transfer conditions and calculations of temperatures in units of sliding friction play an important role in investigation of the serviceability of machines. The reliability of operation of machines decreases as a result of heat liberation and origination of thermal deformations in parts and units of them, since clearances and tightness in the joints of parts and conditions of lubrication change, wear of friction pairs increases, and the possibility of jamming occurs. It is often necessary to constantly monitor the thermal state of sliding bearings, especially in heavy machines, whose repair requires much time and labor and is very expensive. For example, due to the breakdown of an exhaust fan of a central heating-and-power plant, the power of a unit decreases by 60 MW, which greatly affects the total power of the plant. Additional financial costs are related directly to repair (the time for replacement of bearings is up to about 8 h), operation of the central heating-and-power plant within the Integrated Power Supply System, and system losses, which reach one-third of the cost of 1 MW.

Under these conditions, it is worth using the potentials of computer modeling to predict the thermal state of sliding bearings and make a decision on the necessity of monitoring temperatures in the joint by measurement devices.

The heat-flux power originating due to friction in the bearing is

$$W_0 = 2\pi M n \, . \tag{1}$$

Here M is the moment of friction in the bearing, which according to the technique of calculations [1] recommended by the IOS (International Organization for Standardization) is the function

$$M = f(\mu, n, d, l, e, \delta) \, . \tag{2}$$

The amount of heat liberated due to friction in the bearing is removed to the shaft, lining, and lubricant, i.e., the heat balance has the form

$$W_0 = W_{\mathrm{sh}} + W_{\mathrm{h}} + W_{\mathrm{lub}} \, . \tag{3}$$

The temperature on the shaft surface is

$$T_{\mathrm{sh}} = \frac{2q_{\mathrm{sh}}\sqrt{\omega_{\mathrm{sh}}}}{\lambda_{\mathrm{sh}}\sqrt{\pi}} \sqrt{\tau}\, L_{\mathrm{sh}} \, ; \quad \omega_{\mathrm{sh}} = \frac{\lambda_{\mathrm{sh}}}{c_{\mathrm{sh}}\,\rho_{\mathrm{sh}}} \, , \tag{4}$$

and on the surface of the lining it is

$$T_{\mathrm{h}} = \frac{2q_{\mathrm{h}}\sqrt{\omega_{\mathrm{eq}}}}{\lambda_{\mathrm{eq}}\sqrt{\pi}} \sqrt{\tau}\, L_{\mathrm{h}} \, . \tag{5}$$

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According to the boundary condition of the IVth kind, the temperatures of the contacting friction surfaces of two bodies are the same, i.e., $T_{sh} = T_h = T$.

In Eqs. (4) and (5), the shape factor is

$$L_h = \frac{r_{out.lin}}{\Delta}\left|\ln\left(1 - \frac{r_{out.lin}}{r_{in.lin}}\right)\right| , \tag{6}$$

and the coefficient L_{sh} is [2]

$$L_{sh} = c\,\mathrm{Fo}^m = c\left(\frac{\lambda_{sh}}{c_{sh}\,\rho_{sh} d^2}\right)^m . \tag{7}$$

The density of the heat flux to the shaft is

$$q_{sh} = b^* \frac{W - W_{lub}}{\pi dl} ; \tag{8}$$

$$b^* = \frac{1}{\left(1 + \frac{\lambda_{eq}}{\lambda_{sh}}\frac{L_{sh}}{L_h}\sqrt{\frac{\omega_{sh}}{\omega_{eq}}}\right)} ; \tag{9}$$

$$\lambda_{eq} = \frac{\ln\left(\frac{r_{out.lin}}{r_{in.lin}}\right)}{\sum\limits_{i=1}^{m}\left(\frac{1}{\lambda_i}\left|\ln\left(1 - \frac{\Delta_i}{r_i}\right)\right|\right)} ; \tag{10}$$

$$\omega_{eq} = \frac{\lambda_{eq}}{(c\rho)_{eq}} ; \quad (c\rho)_{eq} = \frac{\sum\limits_{i=1}^{m} c_i\,\rho_i V_i}{\sum\limits_{i=1}^{m} V_i} . \tag{11}$$

The actual power of the heat flux in the seat with account for changes in the viscosity of oil in heating can be calculated as

$$W = W_0\,(1 - kT) . \tag{12}$$

The power of the heat flux which enters the lubricant escaping from the bearing is

$$W_{lub} = (c\rho)_{lub}\,TQ . \tag{13}$$

Finally, the temperature in the joint of the sliding bearing is

$$T = \frac{W_0\sqrt{\tau}}{\left(\frac{\sqrt{\pi^3}}{2}\,\frac{\lambda_{sh}}{\sqrt{\omega_{sh}}}\,\frac{dl}{b^* L_{sh}}\right) + [kW_0 + (c\rho)_{lub}Q]\sqrt{\tau}} .$$

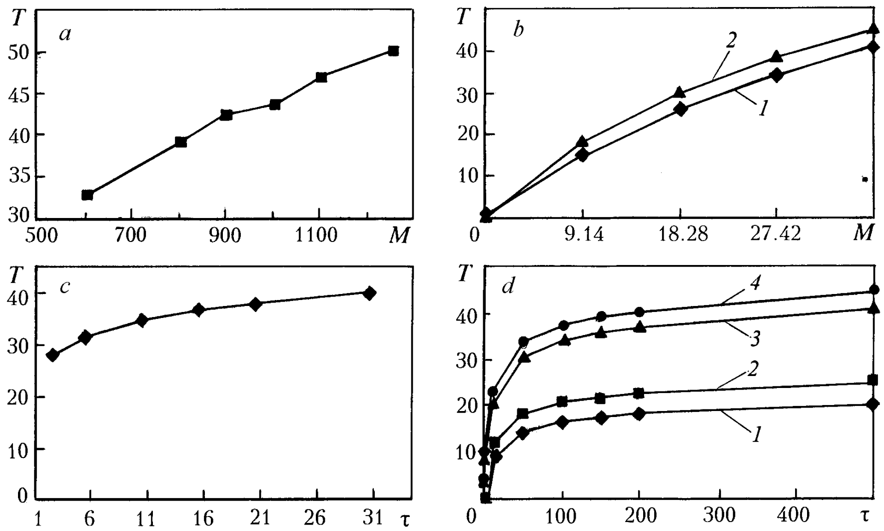

Fig. 1. Dependence of the temperature in the bearings on the frictional moment (a, b) and time of operation (c, d): a) fan, time of operation 6400 h; b) conveyer, time of operation 600 (1) and 6000 h (2); c) fan, torque 900 N·m, rotational speed 12.3 revolutions/sec, oil flow rate 30 liters/min; d) conveyer, rotational speed of the shaft: 1) n = 0.1 revolutions/sec; 2) 0.133; 3) 0.266; 4) 0.3. T, °C; M, N·m; τ, min.

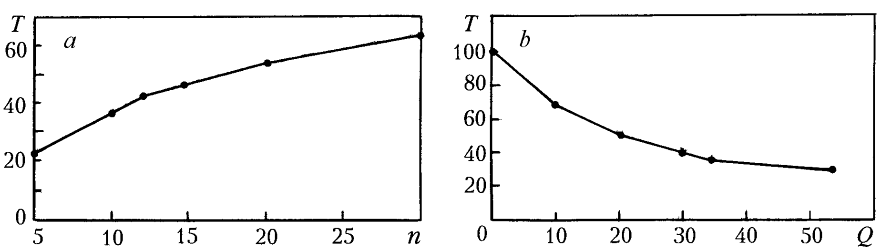

Fig. 2. Dependence of temperature in the bearings of the fan on rotational speed (a) and oil flow rate (time of operation 6400 h) (b). T, °C; n, rpsec; Q, liter/min.

Based on the above-given algorithm, we developed a program for calculation of temperatures arising under different operating conditions of bearings. Two typical regimes of operation of bearings — with high velocity and oil lubrication and with low velocity and plastic lubrication — were considered. In the first case, operating conditions of a suction two-sided fan were analyzed. The material of the fan shaft is steel 45. The weight of the shaft with the blades is 9560 kg, the diameter of the bearing neck 190 mm, the torque 9550 N·m, the frictional moment ~900 N·m, and the rotational speed 740 rpm. The unit was lubricated by AN-46Z machine oil (viscosity 0.17 Pa·sec) at a pressure of 0.1 MPa and with a flow rate of $0.5 \cdot 10^{-3}$ m^3/sec (30 liters/min). The material of the bearing lining is CuSn8Pb15Ni brass, the working surface — bearing alloy SnSb8Cu4, and the material of the bearing case — steel casting.

In the second case, the operating conditions of a scraper conveyer were considered. The material of the drive shaft is steel 45. The force in the sliding bases changes within 4–5 kN, the diameter of the bearing neck is 90 mm, and the rotational speed is 8 rpm. Plastic lubricant LT-4S2 is used in the bearing. The material of the bearing lining is CuSn8Zn5Pb5 brass (OTsS 5-5-5), and the material of the case is SCh25 cast iron.

Results of the calculation for different operating conditions of the mechanism under study are given in Figs. 1 and 2.

To compare the results of the calculations with the actual level of temperatures in the bearing of the fan we conducted measurements with a Raynger ST^{TM} optical pyrometer at a distance of 0.5 m from the bearing. The accu-

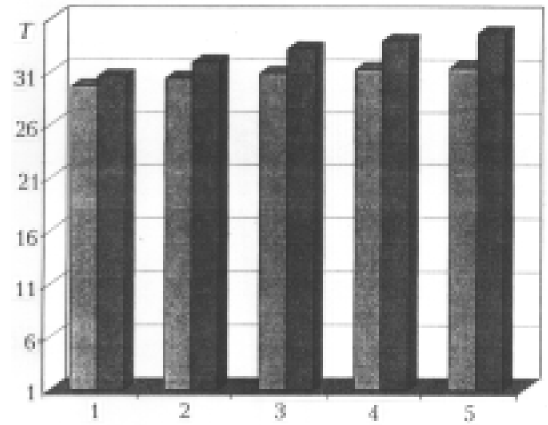

Fig. 3. Comparison of calculated (light field) and measured (dark field) temperatures in the bearing of the conveyer (horizontal axis — time of operation, h). T, ^oC.

racy of measurements for this distance is $\pm 1^oC$. The following values of temperature were obtained from the measurements: temperature of oil in the pipeline in front of the bearing unit — 38^oC; that of the bearing case — 42^oC; that of oil in the pipeline after the bearing unit — 47^oC.

It is easy to see that the temperature difference is 9%, i.e., it is small, especially if we take into account that the computation algorithm disregards the use of heated oil (38^oC instead of room temperature) in the bearing.

The temperature in the bearings of the scraper conveyer was measured by a RAY6LXG pyrometer within each hour of operation under different loading conditions. Comparison of results of the calculation with the actual level of temperatures is shown in Fig. 3. After the first hour of operation the temperatures are virtually the same; then the difference is from 3.5 to 11%.

Thus, as a result of analysis of the conditions of heat transfer in the pairs of sliding friction we developed an algorithm for calculation of the power of heat fluxes and temperatures in sliding bearings. Laws governing the effect of lubrication conditions, applied torques, and time of operation of the studied unit on the level of temperatures in the zone of sliding friction and the level of power of heat fluxes were determined. The data obtained can be used for monitoring the state of bearing units by measurement devices.

NOTATION

W_0, heat-flux power arising due to friction in the bearing, W; M, frictional moment, N·m; n, rotational speed of the shaft, revolutions/sec; μ, coefficient of friction in the bearing; l and d, diameter and length of the bearing, mm; e, eccentricity between axes of the shaft and the hole of the bearing lining, mm; δ, radial clearance in the bearing, mm; W_{sh}, W_h, and W_{lub}, power of heat fluxes entering the shaft, lining with parts adjacent to it, and lubricant, respectively, W; q_{sh} and q_h, densities of heat fluxes directed toward the shaft and the lining hole, respectively, W/mm^2; τ, time of heat propagation, sec; ρ_{sh}, c_{sh}, λ_{sh}, and ω_{sh}, density (kg/m^3), mass heat capacity (J/(kg·^oC)), thermal conductivity (W/(m·^oC)), and thermal diffusivity (m^2/sec) of the shaft material, respectively; λ_{eq} and ω_{eq}, equivalent thermal conductivity (W/(m·^oC)) and thermal diffusivity (m^2/sec), respectively, of the system of solid bodies which include the lining, boss, and the case wall where the bearing is located; L_h and L_{sh}, shape factors of the lining and shaft; $r_{in.lin}$, Δ, and $r_{out.lin}$, inner radius, wall thickness, and outer radius of the lining (multilayer hollow cylinder), respectively, mm; Fo, Fourier number; c and m, coefficients dependent on the level of Fo; λ_i, thermal conductivity of material of the ith layer of the multilayer hollow cylinder "lining–boss–case wall," W/(m·K); Δ_i and r_i, thickness and radius, respectively, of the outer surface of each layer of the multilayer hollow cylinder, mm; $(c\rho)_{eq}$, equivalent bulk heat capacity of the system of solid bodies, J/(m^{3}·^oC); c_i, mass heat capacity of each element of the system "lining–boss–case wall," J/(kg·^oC); ρ_i, density of material of each of these elements, kg/m^3; V_i, volume of each element of the system "lining–boss–case wall," m^3; W, actual power of heat flux in the base, W; $k \approx 0.01$, coefficient which allows for change in lubricant viscosity with increase in temperature; $(c\rho)_{lub}$, bulk heat capacity of lubricant, J/(m^{3}·^oC); T, temperature of lubricant escaping from the bearing, which is equal to the temperature of the friction surface, ^{o}C; Q, flow rate of lubricant, m^3/sec. Subscripts: sh, shaft; in.lin, inner radius of the lining; h, hole of the lining; out.lin, outer radius of the lining; eq, equivalent; i, number of the layer of the multilayer lining; lub, lubricant.

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