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RECONSTRUCTION OF THE FRICTION MOMENT IN A MOBILE CYLINDRICAL COUPLING FROM MEASUREMENTS OF TEMPERATURE

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For slide bearings with different conditions of rotation of the shaft mathematical thermal models (MTM) are developed that involve superpositions of both two-dimensional and two- and three-dimensional elements. Based on the solution of a boundary-value inverse problem by the method of iteration regularization, a technique is suggested for reconstructing the friction moment from measurements of temperature.

In tests of friction elements on experimental rigs and during their service, the facilities available to measure directly the mechanical energy expended for friction do not yield a quantitative estimate of its magnitude. First of all, this is associated with the compact form of real elements and the impossibility of finding a place in them for rather large mechanical facilities. The characteristic of friction losses in slide bearings is the friction moment. This parameter is usually determined by different kinds of torsiometers and beam strain gauges, with the placement of the latter in the operating couplings being virtually impossible. In view of this, in [1-3] the method of thermal diagnosis of friction (TDF) was justified theoretically and verified experimentally. It is based on the fact that almost all of the energy expended for friction is converted into heat [4]. In TDF the friction moment (heat liberation) in the zone of frictional contact is reconstructed by solving a boundary-value inverse problem with the use of MTM by the method of iteration regularization [5]. The use of TDF substantially increases the informativeness of tribotechnical tests, since temperature measurement is an integral part of such tests.

In [1-3] a simplified MTM representing the superposition of one- and two-dimensional temperature fields (model (1×2)) was considered. Such a simplification was adopted due to a substantial difference (up to two orders of magnitude) between the thermophysical characteristics of the metal-polymer pair and also due to a small shaft diameter-to-length ratio.

In the present work this restriction is removed, and this makes it possible to reconstruct the friction moment for a wider class of slide bearings. Moreover, we consider cylindrical couplings with alternating motion of the shaft with a small, as compared to the zone of contact, amplitude of oscillations, for which the temperature field distribution has a number of special features.

I. Simulation of Unsteady Heat Transfer in Slide Bearings (SB) with Rotary Motion (Model (2×2)). We consider a radial slide bearing (Fig. 1). Shaft 1 contacts bushing 2 rigidly fixed in housing 3. In the zone of contact ($r = r_1$, $-\varphi_0 \le \varphi \le \varphi_0$) frictional heat is liberated with intensity $Q(t)$. We adopt the following simplifications: 1) the speed of the shaft is rather high (> 5 rad/sec), the heat flux is distributed uniformly within the arc of contact; 2) heat liberation from the end faces of the bushing and the housing can be neglected.

We write the heat conduction equation for the bushing and the race

$$\frac{\partial T}{\partial t} = A(r)\frac{\partial^2 T}{\partial r^2} + B(r)\frac{\partial T}{\partial r} + C(r)\frac{\partial^2 T}{\partial \varphi^2}, \qquad (1)$$

$$r_2 < r < r_4, \quad 0 < \varphi < \pi, \quad 0 < t \le t_m.$$

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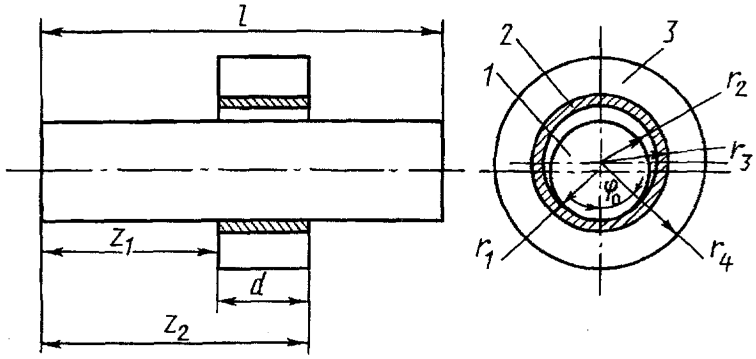


Fig. 1. Design sketch of a friction element: 1) shaft; 2) bushing; 3) housing; φ_0 is the half-angle of contact of a shaft with a bushing.

On the outer edge of the race and also from the bushing surface outside the zone of contact heat transfer is prescribed by the Newton law

$$\lambda \frac{\partial T}{\partial r} = \alpha (T(r, \varphi, t) - T_0(t)). \tag{2}$$

Within the arc of contact frictional heat liberation takes place:

$$2\varphi_0 \lambda_2 r_1 \int_{z_1}^{z_2} \frac{\partial u(r_1, z, t)}{\partial r} dz - 2r_1 d\lambda_1 \int_0^{\varphi_0} \frac{\partial T}{\partial r}\bigg|_{r=r_2} d\varphi = Q(t); \tag{3}$$

$$T(r_2, \varphi, t) = u(r_1, z, t). \tag{4}$$

The temperature distribution in the shaft is found from the heat conduction equation

$$c_2 \rho_2 \frac{\partial u}{\partial t} = \lambda_2 \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u}{\partial r} \right) + \frac{\partial^2 u}{\partial z^2} \right], \tag{5}$$

$$0 < r < r_1, \quad 0 < z < l, \quad 0 < t \le t_m,$$

with the following boundary conditions:

$$u(r, 0, t) = T_1(t); \tag{6}$$

$$\frac{\partial u(r, l, t)}{\partial r} = 0; \tag{7}$$

$$\lambda_2 \frac{\partial u(r_1, z, t)}{\partial r} = -\alpha [u(r_1, z, t) - T_0(t)]; \tag{8}$$

$$\frac{\partial u(0, z, t)}{\partial r} = 0. \tag{9}$$

Moreover, it is assumed that initially ($t = 0$) the distributions of the temperature field of the shaft and the bearing elements are known and there is a condition of symmetry with respect to φ.

At the prescribed intensity of heat liberation the direct problem is solved numerically with the help of the algorithm from [6]. Comparison of the experimental and calculated values of temperature obtained by means of model (1 × 2) shows that their discrepancy does not exceed 8-10%. Comparison of the calculated distributions of temperature obtained by means of models (1 × 2) and (2 × 2) indicates a high degree of agreement. Thus, one can come to recognize the sufficient efficiency of the algorithm that implements the model (2 × 2).

II. Reconstruction of the Friction Moment in a Crankpin Bearing of an Internal Combustion Engine (ICE). The MTM s given in Section I describe heat transfer in slide bearings intended for tribotesting on serial SMT-1 machines. The high efficiency of the algorithms developed for numerical solution of direct heat conduction problems motivates their use in TDF of actual friction units, in particular in the crankpin bearing of the crankshaft of an ICE. Let us modify the MTM for this case. Due to the high speed of shaft rotation and interaction of the neck with the bushing along the entire contour, the circumferential temperature distribution around the contact is assumed uniform. Since the oil film is rather thin, the contact of the crankshaft neck with the bushing can be regarded as ideal. Suppose the temperature of the lubricant entering the bearing is equal to $T_{en}(t)$.

Taking into account convective removal of heat by the lubricant and the above assumptions, we obtain

$$2\pi \lambda_2 r_1 \int_{z_1}^{z_2} \frac{\partial u(r_1, z, t)}{\partial r} dz - 2r_2 d\lambda_1 \int_0^{\pi} \frac{\partial T(r_2, \varphi, t)}{\partial r} d\varphi = \tag{10}$$

$$= Q(t) - c_{lub} \rho_{lub} V(t)(T(r_2, \varphi, t) - T_{en}(t)) .$$

Since there is heat transfer between the connecting rod and the piston system, it is necessary to take into account the change in the temperature of the former. Therefore, we assume that in its lower part ($r = r_4$, $|\varphi| \le \overline{\varphi}_0$) the temperature is determined from measurements:

$$T(r_4, \varphi, t) = T_{sh}(t) . \tag{11}$$

Conditions (6) and (7) on the ends of the shaft remain invariant. Thus, the MTM for a crankpin and a dry slide bearing differ due to the mixed conditions of heat transfer at the outer edge and to frictional heat liberation (10) with the convective term.

Suppose that at $r = R$ and $\varphi = 0$, additional temperature information is available:

$$T(R, 0, t) = T^e(t) . \tag{12}$$

The boundary-value inverse problem in extremal statement is formulated as follows: it is necessary to find the intensity of heat liberation $Q(t)$ and the friction moment $M_{fr}(t)$, interrelated by the formula

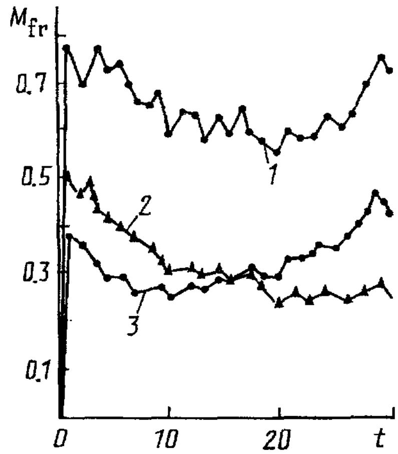
$$M_{fr}(t) = Q(t) r_1 / v_{sh} ,$$

from the condition of a minimum of the functional

$$J[Q(t)] = \int_0^{t_m} [T(R, 0, t) - T^e(t)]^2 dt \tag{13}$$

for solutions of the system of equations that describe heat transfer in a crankpin bearing.

The problem under consideration was solved by the method of iteration regularization on the basis of the gradient minimization of the functional [5]. The nontraditional character of condition (10) somewhat complicates the equations of the conjugate problem used for finding the gradient of the functional. Omitting intermediate calculations, we write boundary conditions for the temperature increment that are obtained by introducing the heat flux coefficient $\alpha_{h.f}$:



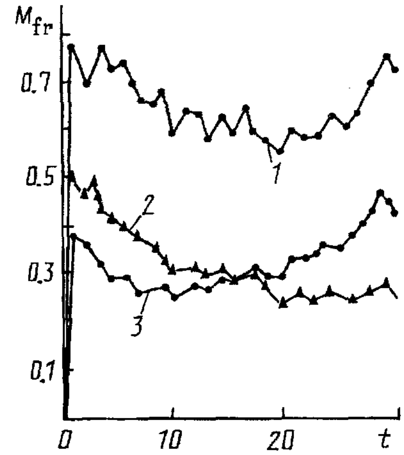


Fig. 2. Friction moment M_{fr} (N·m) vs time t (min) for different discharges of the lubricant V: 1) $V = 0.5$ dm^3/min; 2) $0.5(\sin \frac{t\pi}{30} - \frac{\pi}{2}) + 1$; 3) 0.15 dm^3/min.

Fig. 3. Dependences of the temperature at the connecting rod T_{sh} (1), of the lubricant T_{en} (2), at one end of the shaft T_1 (3), and in the bearing T^e (4) (^{o}C) on time [7].

$$2\pi \lambda_2 r_1 \int_{z_1}^{z_2} \frac{\partial w (r_1, z, t)}{\partial r} dz = \alpha_{h.f} (\Delta Q (t) - c_{lub} \rho_{lub} V(t) w (r_2, \varphi, t)) ;$$

$$- 2r_2 d\lambda_1 \int_0^{\pi} \frac{\partial v (r, \varphi, t)}{\partial r} d\varphi = (1 - \alpha_{h.f}) (\Delta Q (t) - c_{lub} \rho_{lub} V(t) v (r_2, \varphi, t)) .$$

By applying the main theorem of variational calculus, eliminating $\alpha_{h.f}$, and employing substitution of variables, we arrive at a boundary condition of the form (10).

In Fig. 2 the results of calculations of the friction moment in a crankpin bearing of an automobile engine for the shaft rotation speed ν_{sh} = 1500 rev/min are given. The temperature data (Fig. 3) and the geometric dimensions are borrowed from [7]. Missing thermophysical characteristics are taken from domestic publications. Some of the parameters (volume discharge of lubricant, coordinates of the points at which thermocouples are imbedded) were not used in [7]. Thus, some of the parameters differ somewhat from those used in that work. In view of this, the aim of the present work was to perform a qualitative reconstruction of the change in the friction moment. The results obtained reproduce qualitatively the behavior of the parameter of interest, indicating the possible practical application of TDF for reconstructing the friction moment in a crankpin bearing of an ICE. In this case the error in temperature measurements should not exceed 0.5^{o}C, since the temperatures T_1, T_{en}, and T_{sh} differ insignificantly.

III. Reconstruction of the Friction Moment in Bearings with Alternating Motion . Bearings with alternating motion with a small oscillation amplitude are used extensively for damping various vibrations. The special features of the temperature field of such supports were investigated in [8]. In that work the conclusion was drawn that calculations of heat transfer in a coupling should take into account the circumferential temperature nonuniformity in the shaft. For this purpose a (3 × 2) model was developed. For the shaft, the three-dimensional heat conduction equation in cylindrical coordinates

$$c_2 \rho_2 \frac{\partial u}{\partial t} = \lambda_2 \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 u}{\partial \varphi^2} + \frac{\partial^2 u}{\partial z^2} \right] , \tag{14}$$

was written, which is connected with the two-dimensional equation for a housed bushing via the condition on the contact

$$\lambda_2 \frac{\partial u}{\partial r}\Big|_{r=r_1-0} - \lambda_1 \frac{\partial T}{\partial r}\Big|_{r=r_1+0} = \overline{Q}(\varphi, t). \qquad (15)$$

In contrast to a bearing with rotary motion, the intensity of heat liberation is represented by a function of two variables. In this case the temperature should be measured along the circumference in the vicinity of the contact zone

$$T(R, \varphi, t) = T^{e}(\varphi, t).$$

The minimized functional has the form

$$J[Q(\varphi, t)] = \int_0^{t_m} \int_0^{\varphi_0} [T(R, \varphi, t) - T^{e}(\varphi, t)]^2 \, d\varphi dt. \qquad (16)$$

Algorithms for numerical solution of direct and inverse problems were constructed. The calculations showed that one iteration on a (11 × 6, 10 × 16 × 20, 30) grid requires about 15 min (IBM PC/AT), and the moment is reconstructed in for 6-7 iterations. The attained accuracy of the reconstruction of the friction moment, which is commensurable with the error in the experimental data, as well as insignificant expenditures of computer time indicate the suitability of TDF in bearings with alternating motion.

NOTATION

T, u, temperature of the housed bushing and the shaft, respectively; t, running time; z, coordinate along the shaft; r, φ, polar coordinates; t_m, time of testing; ρ_1, c_1, λ_1, density, specific heat, and thermal conductivity of the bushing material; ρ_2, c_2, λ_2, density, specific heat, and thermal conductivity of the shaft and housing material; α, coefficient of heat transfer from free surfaces to the environment; T_0, ambient temperature; w, v, increment in the temperature of the shaft and the housed bushing; Q, intensity of heat liberation; ΔQ, increment in heat liberation intensity; $\overline{Q}$, specific heat liberation intensity; J, discrepancy functional; φ_0, half-angle of contact; $\overline{\varphi}_0$, half-angle at which the connecting rod temperature is prescribed; ρ_{lub}, c_{lub}, V, density, specific heat, and discharge of the lubricant; v_{sh}, speed of shaft rotation.

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