NOTATION

k_r, recombination rate constant for oxygen atoms; $K_r(M)$, oxygen recombination rate constant with particle M playing role of catalyst; [O], $[O]_2$, [M], concentrations of O, O_2, and M particles; P_1, initial gas pressure in shock tube; P_5, pressure in reflected shock wave; V_s, shock wave velocity; T_5, gas temperature in reflected shock wave.

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EXPERIMENTAL EVALUATION OF THE EFFICIENCY OF TEMPERATURE DIAGNOSIS OF FRICTION

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UDC 621.89:536.24

Results are reported from an experimental evaluation of the efficiency of determining the moment of friction in a cylindrical coupling from measurements of temperature inside the bushing.

In [1], we attempted to solve the problem of identifying heat release in a cylindrical coupling from measurements of the temperature of the rubbing bodies. Determination of this quantity allowed us to determine the friction work of the sliding contact, since it is known that most of this work is converted into heat [2, 3].

To evaluate the efficiency of the method employed in this case, we experimentally checked the determination of the friction moment as a function of time from temperature measurements. The results of this investigation are reported here.

In [1], we used an idealized planar thermal model. To construct the thermal model of the test element, it is necessary to consider heat removal along the axis of the shaft (Fig. 1). Here, the following simplifications are made: the temperatures in the cross sections of the shaft and along the bushing are uniform; heat exchange with the ends of the bushing is not considered. Also, in formulating the boundary conditions, we used the measurement of shaft temperature far from the friction zone. Then the heat conduction equation for the shaft is written in the form:

$$c_2\rho_2 \frac{\partial u}{\partial t} = \lambda_2 \frac{\partial^2 u}{\partial z^2} - \frac{P}{S}\alpha_S(u - T_0) + \left[Q(t) + 2r_2\lambda_1 d \int_0^{\varphi_0} \frac{\partial T(r_2, \varphi, t)}{\partial r} d\varphi\right] \frac{\theta(z)}{Sd}, \quad (1)$$

Institute of Physicotechnical Problems of the North, Yakutsk Branch, Siberian Department of the Academy of Sciences of the USSR, Yakutsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 53, No. 3, pp. 442-446, September, 1987. Original article submitted June 30, 1986.

0022-0841/87/5303-1057$12.50

$$0<z<l,\ 0<t\leqslant t_m;$$

$$\theta(z)=\begin{cases}1, & z\in A_c,\\ 0, & z\notin A_c,\end{cases}$$

where A_c is the set of points of the shaft in contact with the bushing, with the boundary conditions

$$T(r_2,\ \varphi,\ t)=u(z_k,\ t);\ z_k\in A_c;\ |\varphi|\leqslant\varphi_0. \tag{2}$$

The measured temperature is given on the left end of the shaft:

$$u(0,\ t)=\overline{u}(t); \tag{3}$$

while convection occurs on the free end:

$$\lambda_2\frac{\partial u(l,\ t)}{\partial z}=-\alpha_s(u(l,\ t)-T_0). \tag{4}$$

With allowance for conditions (1) and (2), the temperature distribution $T(r,\varphi,\ t)$ in the bushing is determined from the system [1]:

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

$$r_2<r<r_4;\ 0<\varphi<\pi;\ 0<t\leqslant t_m, \tag{5}$$

$$A(r)=\frac{\lambda(r)}{c(r)\rho(r)};\ B(r)=\frac{A(r)}{r};\ C(r)=\frac{A(r)}{r^2}.$$

At $|\varphi|>\varphi_0$

$$\lambda_1\frac{\partial T(r_2,\ \varphi,\ t)}{\partial r}=\alpha(T(r_2,\ \varphi,\ t)-T_0); \tag{6}$$

$$\lambda_2\frac{\partial T(r_4,\ \varphi,\ t)}{\partial r}=-\alpha(T(T_4,\ \varphi,\ t)-T_0),\ 0<\varphi<\pi; \tag{7}$$

$$\frac{\partial T(r,\ 0,\ t)}{\partial\varphi}=\frac{\partial T(r,\ \pi,\ t)}{\partial\varphi}=0; \tag{8}$$

$$\lambda_1\frac{\partial T(r_3-0,\ \varphi,\ t)}{\partial r}=\lambda_2\frac{\partial T(r_3+0,\ \varphi,\ t)}{\partial r};\ T(r_3-0,\ \varphi,\ t)=T(r_3+0,\ \varphi,\ t); \tag{9}$$

$$T(r,\ \varphi,\ 0)=u(z,\ 0)=0. \tag{10}$$

The numerical solution of system (1-10) is obtained by the trial-run method, as in [4]. Using additional measurements of temperature in the bushing with a fixed R

$$T(R,\ \varphi_j,\ t)=T_j^e(t),\ r_2<R<r_3,\ j=\overline{1,M}, \tag{11}$$

the intensity of heat release Q(t) and the friction moment $M_{fr}(t)$, connected to each other through the formula

$$M_{fr}=\frac{Q(t)r_s}{v}, \tag{12}$$

are determined from the condition of the minimum of the functional:

$$J[Q(t)]=\sum_{j=1}^{M}\int_0^{t_m}[T(R,\ \varphi_j,\ t)-T_j^e(t)]^2dt, \tag{13}$$

which is a measure of the deviation of the temperatures calculated from system (1-10) from the measured temperatures.

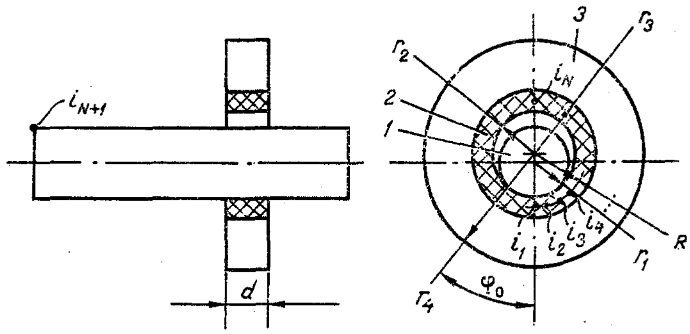

Fig. 1. Design sketch of friction element: 1) shaft; 2) bushing; 3) housing; i_k) points of temperature measurement.

To calculate the gradient of functional (13), it is necessary to examine the following problem when using gradient methods of minimization:

$$-c_2\rho_2\frac{\partial\chi}{\partial t}=\lambda_2\frac{\partial^2\chi}{\partial z^2}-\frac{P}{S}\alpha_s\chi+\frac{2\theta(z)r_2\lambda_1 d}{Sd}\int_0^{\varphi_0}\frac{\partial\psi(r_2,\ \varphi,\ t)}{\partial r}d\varphi,\ 0<z<l,\ 0<t\leqslant t_m; \tag{14}$$

$$\psi(r_2,\ \varphi,\ t)=\chi(z_k,\ t),\ |\varphi|\leqslant\varphi_0; \tag{15}$$

$$\chi(0,\ t)=0; \tag{16}$$

$$\lambda_2\frac{\partial\chi(l,\ t)}{\partial z}=-\alpha_s\chi(l,\ t); \tag{17}$$

$$-\frac{\partial\psi}{\partial t}=A(r)\frac{\partial^2\psi}{\partial r^2}+B(r)\frac{\partial\psi}{\partial r}+C(r)\frac{\partial^2\psi}{\partial\varphi^2}+\frac{2A(r)}{r}[T(R,\ \varphi,\ t)-T_j^e(t)]\,\delta(r-R)\,\delta(\varphi-\varphi_j);\ j=\overline{1,M}, \tag{18}$$

$$r_2<r<r_4;\ 0<\varphi<\pi;\ 0<t\leqslant t_m,$$

where

$$\delta(x)=\begin{cases}1, & x=0,\\ 0, & x\neq 0.\end{cases}$$

At $|\varphi|>\varphi_0$

$$\lambda_1\frac{\partial\psi(r_2,\ \varphi,\ t)}{\partial r}=\alpha\psi(r_2,\ \varphi,\ t); \tag{19}$$

$$\lambda_2\frac{\partial\psi(r_4,\ \varphi,\ t)}{\partial r}=-\alpha\psi(r_4,\ \varphi,\ t);\ 0<\varphi<\pi; \tag{20}$$

$$\frac{\partial\psi(r,\ 0,\ t)}{\partial\varphi}=\frac{\partial\psi(r,\ \pi,\ t)}{\partial\varphi}=0; \tag{21}$$

$$\lambda_1\frac{\partial\psi(r_3-0,\ \varphi,\ t)}{\partial r}=\lambda_2\frac{\partial\psi(r_3+0,\ \varphi,\ t)}{\partial r}; \tag{22}$$

$$\psi(r_3-0,\ \varphi,\ t)=\psi(r_3+0,\ \varphi,\ t); \tag{23}$$

$$\psi(r,\ \varphi,\ t_m)=\chi(z,\ t_m)=0. \tag{24}$$

Equation (14) is obtained from the necessary condition of the extremum through the solution of a system of integrodifferential equations [1].

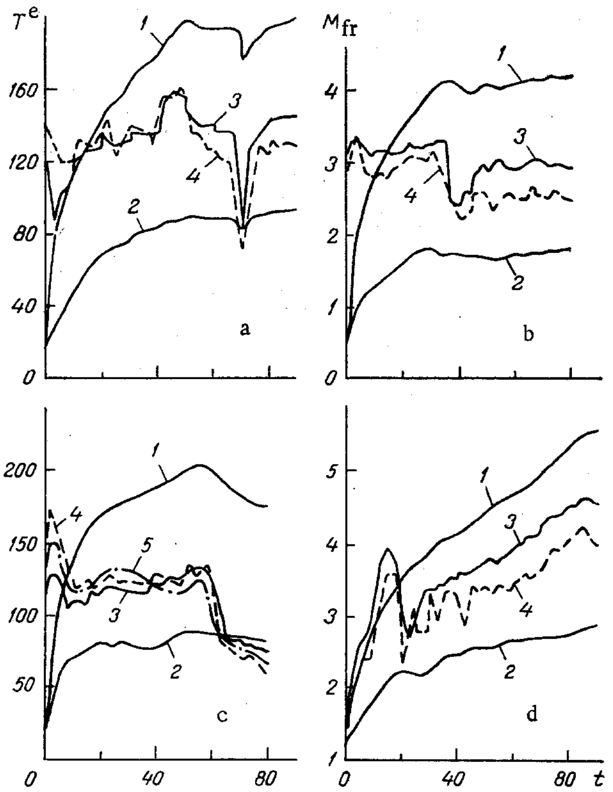

Fig. 2. Comparison of calculzted and experimental curves of the friction moment: 1, 2) measurements in the bushing at $\varphi = 0$ and at the end of the shaft i_{N+1}; 3, 4) experimental and calculated friction-moment relations for loads which were constant (b, c) and variable (a, d) over time; 5) result of determination from the self-controlling algorithm [5], T^e, °C; M_{fr}, N•m; t, min.

The gradient of functional (13) is expressed through the function $\psi(r, \varphi, t)$ as follows:

$$J'[Q(t)] = \frac{\psi(r_2, 0, t) S}{r_2 \lambda_1}. \tag{25}$$

An algorithm which was realized in FORTRAN constructs iterative approximations of the sought function by the method of conjugate gradients, with the condition of stopping for the increment of the functional [5, 6]. This algorithm was used to determine the friction moment in slide bearings.

Experiments were conducted on serial testing machine SMT-1. The machine was equipped with a torsion friction sensor. The tests were conducted at a constant speed of shaft rotation v = 0.39 m/sec in the load range 500–1500 N. We tested cylindrical bushings made of filled fluoroplastic F4K20. The dimensions of the bushings ϕ 32 × ϕ 26 × 20 mm. The diameter of the rotating abradant was 25.5 mm. Temperature was measured with copper–constantan thermocouples with ϕ 0.1 mm. The thermocouples were pressed into the end of the specimen at six points about the circumference r = R = 14.2 mm. The readings of friction moment and the thermocouple readings were recorded automatically by means of an F-799/1 commutator switch matched through a "KAMAK" system with data-computing complex IVK-2, which is based on an SM-4 computer. The thermophysical properties of the specimen and abradant:

$$\text{F4K20: } c\rho = 2{,}67 \cdot 10^6 \text{ J/m}^3 \cdot {}^\circ\text{C}; \ \lambda = 0{,}39 \text{ W/m} \cdot {}^\circ\text{C};$$

$$\text{steel: } c\rho = 3{,}48 \cdot 10^6 \text{ J/m}^3 \cdot {}^\circ\text{C}; \ \lambda = 46 \text{ W/m} \cdot {}^\circ\text{C}.$$

The coefficients of heat transfer with the free surfaces of the components of the friction element were calculated by the methods described in [7, 8].

The reliability of measuring temperature at one point was shown by the determination of friction moment from records of temperature at different bushing points with r = R and variation of the number of points. Figure 2 shows results of determination of the friction moment from thermocouple readings at $\varphi = 0$ for different loads. The theoretical relations satisfactorily (to within 10-15%) describe the experimental data, which is evidence of the possibility of making practical use of the proposed method to diagnose friction in movable couplings.

NOTATION

$Q(t)$, intensity of heat release in the region of the friction contact; T_0, initial temperature; T, temperature of bearing; u, temperature of shaft; t, running time; z, coordinate along the shaft; r, φ, polar coordinates; t_m, test time; S, P, area and perimeter of shaft cross section; ρ, density; c, heat capacity; λ_1, thermal conductivity of the bushing material; λ_2, thermal conductivity of the shaft and housing material; α_s, coefficient of heat transfer from the surface of the shaft; α, heat transfer coefficient of the free surfaces of the bushing and housing; χ, ψ, Lagrangian multipliers.

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NUMERICAL STUDY OF THE EFFECTS OF SLIP AND A TEMPERATURE DISCONTINUITY ON THE SURFACE OF A SPHERE IN A SUPERSONIC FLOW

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UDC 533.6.011.8

This article examines the effect of boundary conditions for slip and temperature discontinuity on drag and heat-transfer characteristics. Numerical solutions of the Navier—Stokes equations are used to analyze the dependence of these effects on the governing parameters of the flow.

Deviations from the continuum model begin to occur with an increase in negative pressure. These deviations can be accounted for in a gas of moderately low density by means of slip and temperature-discontinuity boundary conditions. The effect of these conditions in problems involving supersonic flow about bluff bodies has usually been studied with the use of the Navier—Stokes equations (see [1], for example). The authors of [2] used the example of unidimensional flow on the stagnation line to show that the complete equations need not be used when the above-mentioned effects are negligible. Well-known solutions of the complete two-dimensional Navier—Stokes equations with slip and temperature-discontinuity conditions [3-6] demonstrate the importance of these effects but are not accompanied by an analysis of their dependence on the governing parameters of the flow. Below we present results of a study of the dependence of the effects of slip and temperature discontinuity on the Reynolds number, surface temperature, energy accommodation coefficient, and the properties of the gas.

1. We examine the steady supersonic flow of an ideal gas about a sphere at Reynolds numbers $Re \gtrsim 20$. The Reynolds number Re was calculated from the radius of the sphere and parameters of the gas behind a forward-traveling shock wave. As is known, the use of such a similarity criterion nearly eliminates the dependence of the characteristics of the flow about the sphere on the Mach number of the incoming flow.

A. F. Ioffe Physicotechnical Institute, Leningrad. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 53, No. 3, pp. 446-450, September, 1987. Original article submitted June 16, 1986.

0022-0841/87/5303-1061$12.50