EFFECT OF INERTIA IN A LUBRICANT ON THE THERMOPHYSICAL PROCESSES IN THE RADIAL SLIP BEARINGS OF TURBINE-DRIVEN MACHINERY

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The results from a numerical solution of the spatial nonisothermal problem of lubricant flow and heat transfer in a radial bearing are presented in conjunction with consideration of the lubricant inertia.

The contemporary level of development in turbine-drive machinery is characterized by a rise in the radial velocities of the shafts in slip bearings as well as by the transition to low-viscosity lubricants. This trend is enhanced through the increasing appearance of inertial effects in lubricant flows. The well-known theoretical work [1-3] into the effect of inertial forces on the functioning of slip bearings was done in the assumption that the lubricant flow was isothermal. Nevertheless, it has been demonstrated in numerous papers, particularly in [4-6], that the thermal processes in the stress-bearing lubricant layers of slip bearings play a decisive role in the distribution of the pressures and velocities of the lubricant in the working spaces. On the other hand, with consideration of the lubricant inertia, the hydrodynamic problem becomes nonlinear and its solution becomes more complex. The various methods of linearizing the original equations, used in analyzing the effects of inertia, sometimes result not only in quantitatively, but also in qualitatively, divergent results.

Unlike the estimation approaches to the analysis of inertial effects, the latter based on the utilization of various methods for the linearization of the isothermal equations of motion, the authors have also worked out a mathematical model, numerically generated, for the nonisothermal flow of the lubricant, as well as for the associated transfer of heat in the radial slip bearing, said model based on the direct solution of truncated Navier–Stokes equations written with consideration of the convection inertia terms, together with the differential equations of heat transfer in the lubricant and in those structural elements which contain it. The system of differential equations written in dimensionless quantities in this case has the form:

the equations of motion

$$\mathrm{Re}\left[u\frac{\partial u}{\partial x}+\left(\frac{v}{h}-u\frac{y}{h}\frac{\partial h}{\partial x}-w\frac{y}{h}\frac{\partial h}{\partial z}\right)\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right]=-\frac{\partial p}{\partial x}+\frac{1}{h^2}\frac{\partial}{\partial y}\left(\bar{\mu}\frac{\partial u}{\partial y}\right); \quad (1)$$

$$\mathrm{Re}\left[u\frac{\partial w}{\partial x}+\left(\frac{v}{h}-u\frac{y}{h}\frac{\partial h}{\partial x}-w\frac{y}{h}\frac{\partial h}{\partial z}\right)\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right]=-\frac{\partial p}{\partial z}+\frac{1}{h^2}\frac{\partial}{\partial y}\left(\bar{\mu}\frac{\partial w}{\partial y}\right); \quad (2)$$

the equation for the distribution of lubricant pressure

$$\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial z^2}=\frac{1}{h^2}\left[\frac{\partial u}{\partial y}\left(\frac{\partial\bar{\mu}}{\partial x}-\frac{y}{h}\frac{\partial h}{\partial x}\frac{\partial\bar{\mu}}{\partial y}\right)+\bar{\mu}\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial x}-\right.\right.$$
$$\left.-\frac{y}{h}\frac{\partial h}{\partial x}\frac{\partial u}{\partial y}\right)+\frac{\partial w}{\partial y}\left(\frac{\partial\bar{\mu}}{\partial z}-\frac{y}{h}\frac{\partial h}{\partial z}\frac{\partial\bar{\mu}}{\partial y}\right)+\bar{\mu}\frac{\partial}{\partial y}\left(\frac{\partial w}{\partial z}-\right. \quad (3)$$

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$$- \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial w}{\partial y} \Big) \Big|_0^1 - \mathrm{Re} \int_0^1 \Big[\Big(\frac{\partial u}{\partial x} - \frac{y}{h} \frac{\partial h}{\partial x} \frac{\partial u}{\partial y} \Big)^2 +$$

$$+ \frac{1}{h^2} \Big(\frac{\partial v}{\partial y} \Big)^2 + \Big(\frac{\partial w}{\partial z} - \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial w}{\partial y} \Big)^2 + \frac{2}{h} \frac{\partial u}{\partial y} \Big(\frac{\partial v}{\partial x} - \tag{3}$$

$$- \frac{y}{h} \frac{\partial h}{\partial x} \frac{\partial v}{\partial y} \Big) + 2 \Big(\frac{\partial w}{\partial x} - \frac{y}{h} \frac{\partial h}{\partial x} \frac{\partial w}{\partial y} \Big) \Big(\frac{\partial u}{\partial z} - \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial u}{\partial y} \Big) + \frac{2}{h} \frac{\partial w}{\partial y} \Big(\frac{\partial v}{\partial z} - \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial v}{\partial y} \Big) \Big] dy;$$

the continuity equation

$$\frac{\partial^2 v}{\partial y^2} = - h \frac{\partial}{\partial y} \Big(\frac{\partial u}{\partial x} - \frac{y}{h} \frac{\partial h}{\partial x} \frac{\partial u}{\partial y} + \frac{\partial w}{\partial z} - \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial w}{\partial y} \Big); \tag{4}$$

the equation of lubricant energy

$$u \Big(\frac{\partial t}{\partial x} - \frac{y}{h} \frac{\partial h}{\partial x} \frac{\partial t}{\partial y} \Big) + \frac{v}{h} \frac{\partial t}{\partial y} + w \Big(\frac{\partial t}{\partial z} - \frac{y}{h} \frac{\partial h}{\partial z} \frac{\partial t}{\partial y} \Big) =$$

$$= \frac{\bar{\mu}}{h} \frac{\mathrm{Ec}}{\mathrm{Re}} \Big[\Big(\frac{\partial u}{\partial y} \Big)^2 + \Big(\frac{\partial w}{\partial y} \Big)^2 \Big] + \frac{1}{\mathrm{Pe}\, h^2} \frac{\partial^2 t}{\partial y^2}; \tag{5}$$

the equation of shaft heat conduction

$$\frac{1}{r} \frac{\partial}{\partial r} \Big(r \frac{\partial t_w}{\partial r} \Big) + \frac{\partial^2 t_w}{\partial z^2} = 0, \tag{6}$$

the equation of bearing heat conduction

$$\frac{1}{r} \frac{\partial}{\partial r} \Big(r \frac{\partial t_p}{\partial r} \Big) + \frac{1}{r^2} \frac{\partial^2 t_p}{\partial \theta^2} + \frac{\partial^2 t_p}{\partial z^2} = 0. \tag{7}$$

Equation (3) has been derived by differentiation of (1) and (2) with respect to x and y, respectively, followed by their subsequent addition and integration from 0 to h with consideration of the condition $\partial p/\partial y = 0$. Equation (4) is a continuity equation differentiated with respect to the variable y. In its solution, this makes it possible to find the distribution of the velocity v which satisfies the boundary conditions y = 0 and y = 1 at the surfaces.

In solution of Eqs. (1)-(4) for the velocities and pressures we use the following conditions:

$$\begin{aligned}
&\text{when} \quad x = 0 \quad p = p_0, \quad w = 0, \quad \int_0^l \int_0^1 uhdzdy = \bar{G}; \\
&\text{when} \quad x = x_k \quad p = 0, \quad w = 0, \quad \frac{\partial u}{\partial x}\Big|_{x_k} = \frac{\partial u}{\partial x}\Big|_{x_k - \Delta x}; \\
&\text{when} \quad y = 0 \quad u = 1, \quad v = w = 0; \\
&\text{when} \quad y = 1 \quad u = v = w = 0; \\
&\text{when} \quad z = 0 \text{ and } z = l \quad p = 0, \quad \frac{\partial}{\partial x} \int_0^l \int_0^1 uhdydz = h \int_0^1 wdy.
\end{aligned} \tag{8}$$

The boundary conditions for the temperatures were chosen from an analysis of the heat-transfer processes occurring at the surfaces bounding the bearing-lubricant-shaft region. We employed boundary conditions of the 3rd kind on the external surfaces of both the bearing and the shaft:

$$\alpha (T_b - T_s) = - \lambda \frac{\partial T}{\partial n}\Big|_n. \tag{9}$$

Over the contour of the lubricant layer, with 0 < y < 1, we made use of the following conditions:

$$\text{when} \quad x=0 \quad t=1; \quad x=x_k \quad \left.\frac{\partial t}{\partial x}\right|_{x_k}=\left.\frac{\partial t}{\partial x}\right|_{x_k-\Delta x}, \tag{10}$$
$$\text{when} \quad z=0 \quad \left.\frac{\partial t}{\partial z}\right|_{0}=\left.\frac{\partial t}{\partial z}\right|_{\Delta z}; \quad z=l \quad \left.\frac{\partial t}{\partial z}\right|_{l}=\left.\frac{\partial t}{\partial z}\right|_{l-\Delta z}.$$

The temperature values at the boundary between the shaft and the lubricant and between the lubricant and the bearing were derived from the conjugacy conditions. For the bearing these conditions in dimensionless form are written as follows:

$$\text{when} \quad y=1 \quad \text{or} \quad r=1 \quad t=t_p, \quad \frac{1}{h}\left.\frac{\partial t}{\partial y}\right|_{y=1}=\frac{\lambda_p\psi}{\lambda}\left.\frac{\partial t_p}{\partial r}\right|_{r=1}. \tag{11}$$

Taking into consideration the assumption of circumferential isothermicity for the shaft at the boundary between the shaft and the lubricant, we achieved conjugacy conditions of the following form:

$$\text{when} \quad y=0 \quad \text{or} \quad r=1 \quad t=t_w, \quad \frac{1}{x_k}\int_0^{x_k}\frac{1}{h}\left.\frac{\partial t}{\partial y}\right|_{y=0}dx=\frac{\psi\lambda_w}{\lambda}\left.\frac{\partial t_w}{\partial r}\right|_{r=1}. \tag{12}$$

The system of differential equations (1)-(7), together with the boundary conditions (8)-(12), were solved by a finite-difference method which involved the utilization of the implicit locally one-dimensional schemes described in [6, 7]. The solution was accomplished by a run-through method. The additionally introduced conditions for u, w, and t in (8) and (10) have no effect on the physics of the processes, but make it possible to carry out this method. Approximation of the first-order derivative in (1), (2), and (5) was accomplished with utilization of that spatial weight whose values were selected from the conditions for the positive approximation of the equations.

For a comparative analysis of the effect of lubricant inertia, and in order to reduce the time of calculation, we initially solved a problem based on application of the classical nonisothermal Reynolds equation valid for noninertial lubricant flow [6]. The mathematical model was retained in this case for the thermal processes. The resulting distributions of pressure, velocity, and temperature were taken as an initial approximation for the solution of the inertial problem.

We carried out the calculations by the described method for a shaft with a diameter d = 420 mm that is extensively used as a journal bearing in turbine construction. A twice-tapered split bearing with shifting bushings permitted an extension of 130° for the upper bushing, and 140° for the lower. For our original base values, we have taken the following quantities as the regime and geometric parameters: shaft rotation speed n = 3000 rpm; the initial lubricant temperature T = 40°C; bearing width L = 0.335 m (L/d = 0.8); radial clearance δ_h = 0.0005 m; degree of ellipticity δ_v/δ_h = 0.5; the horizontal displacement of the upper half relative to the lower, i.e., $\varepsilon_h = e_h/\delta_h$ = 0.5, where e_h is the absolute magnitude of the horizontal displacement; the relative eccentricity e = 0.75. In these calculations we employed the physical characteristics of "Turbine-22" oil.

The calculations were carried out for an adiabatic bearing, i.e., it was assumed in Eq. (9) that α = 0. The outer bearing radius R_2 = 0.294 m. We modeled the flow of heat to the inside depression of a shaft with a radius R_0 = 0.147 m by assuming that α = 600 W/(m²·deg) and the ambient-medium temperature T_a = 100°C.

The calculations demonstrated that the influence of the inertial effects increases as the speed of shaft rotation rises. When n < 1000 rpm (U < 21 m/sec) the solutions of the noninertial and inertial problems virtually coincide with respect to all indicators (Fig. 1). Subsequently, as the circumferential velocity increases there is a redistribution of the lubricant pressures in the working spaces of the upper and lower bushings. When n = 3000 rpm, consideration of the lubricant inertia leads to an increase in the maximum pressures by 16% within the load-carrying bushing, and to a reduction by 12% in the level of pressures within the working space of the upper bushing (Fig. 1). The diverse nature of the influence exerted by inertial forces in the carrying, as well as in the nonstressed, lubricant layers

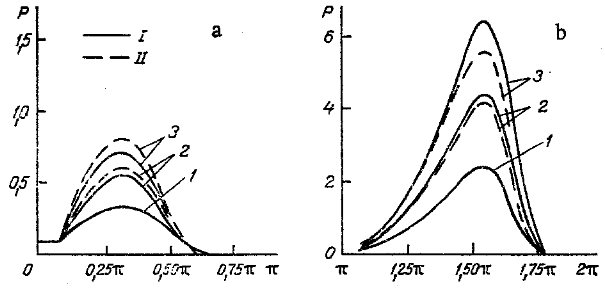

Fig. 1. Circumferential distribution of pressures in the lubricant layer: a) upper half; b) lower half; 1) n = 1000 rpm; 2) 2000 rpm; 3) 3000 rpm; I) with consideration of the inertial effects; II) without consideration of the inertial forces in the lubricant. P, mP; 2π, π, φ/rad.

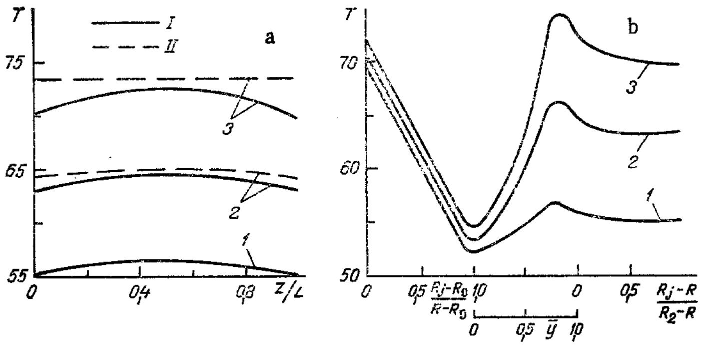

Fig. 2. Temperature distribution in a radial bearing: a) in the axial direction; b) in the radial direction; 1) n = 1000 rpm; 2) 2000 rpm; 3) 3000 rpm; I, II) see Fig. 1. T, °C.

bears witness to the complexity of the processes occuring within them. It has been demonstrated analytically that the fundamental factors determining the degree to which inertial forces make their appearance are the magnitudes of the lubricant thickness and of the pressure gradients. The highest pressure gradients and the lowest lubricant thicknesses are attained in the lower working space. This enhances an increase in pressures as a consequence of the inertial forces. The relationship between the pressures and the effect of the inertial forces in the upper working space is more complex. Here, the lubricant thicknesses are several times greater, and the pressure gradients are severalfold smaller. Moreover, as was demonstrated by calculation, the effect of the inertial forces leads to a reduction in the load angle and to an increase in lubricant outflow at the end. As a result of the total effect of the cited factors, the appearance of inertial forces leads to some reduction in the pressures within the upper space.

Making provision for the inertial forces in the case of constant eccentricity has little effect on the temperature distribution in the lubricant layer: the fundamental effect is associated with some increase in the axial temperature gradient (Fig. 2a).

It should be noted that inclusion within the region being calculated not only of the lubricant layer, but of the structures which contain the lubricant, as well as consideration of the process of heat transfer between these structures, made it possible to obtain a spatial temperature field for all elements of the "shaft–lubricant–bearing" system. The temperature distribution (Fig. 2b) is both complex and nonlinear in nature, dependent on the great number of geometric and regime parameters, as has been demonstrated by actual investigation.

The redistribution of the pressures in the lubricant layer leads to a change in the integral characteristics of the bearing as a consequence of the effect of the inertial forces.

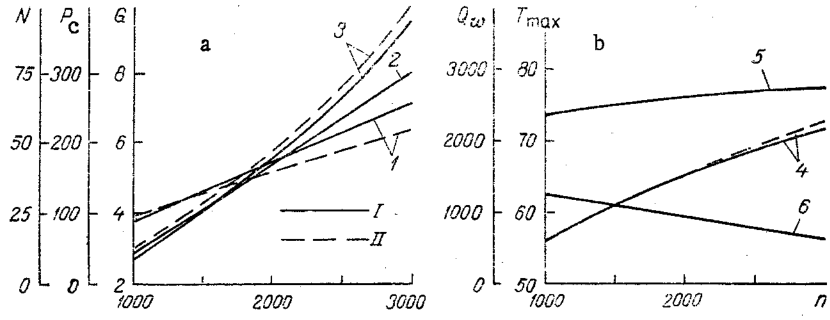

Fig. 3. Change in the integral characteristics of a bearing as a function of shaft rotation speed: 1) stress-carrying capacity of the bearing; 2) lubricant flow rate; 3) power losses; 4) maximum bushing temperature; 5, 6) heat flows at the shaft surface for the upper and lower halves of the bearing, respectively; I, II) see Fig. 1. N, kW; P_c, kN; G, kg-f; Q_w, W; T_{max}, °C; n, rpm.

Most significant is the effect of the inertial forces on the increase in the carrying capacity of the bearing, in particular as the shaft rotation speed increases. When n = 1000 rpm, if the effect of the inertial forces does not result in an increase in the carrying capacity, then when n = 2000 rpm the increase amounts to 8%, reaching 17% when the shaft rotation speed reaches n = 3000 rpm (Fig. 3a).

Consideration of the inertial forces leads to some reduction in power losses resulting from the redistribution of velocities in the lubricant layer. Reduction of the lateral gradients of the circumferential velocity component U (most perceptible at the inlet and outlet segments of the taper, where we find the thickest lubricant layers) leads to a reduction in the force of friction in the bearing.

The thermal characteristics of the bearing for a given magnitude of the eccentricity are virtually independent of the lubricant inertial forces. An increase in the shaft rotation speed leads to an intensification of the transfer of heat within the upper lubricant layer and to a reduction in the transfer of heat in the lower layer. The combined heat flows removed by the lubricant from the shaft are reduced in this case (Fig. 3b).

Table 1 shows the results from our investigation into the extent to which the appearance of lubricant inertia is affected by the magnitude of the radial space, ellipticity, eccentricity (load), horizontal displacement, the width of the bearing, and the initial temperature.

The fundamental conclusion drawn from the results of the analysis of our numerical studies lies in the fact that the effect of the subject parameters on the inertial effects coincides with their influence on the change in the pressure gradients.

Our attention is drawn to that circumstance in which the combination of any given geometric and regime parameters may vary as a function of the velocity boundary for the appearance of inertial effects. If for a bearing with a diameter d = 0.42 m and a relative width L/R = 1.595, the effect of inertia becomes perceptible even at n = 2000 rpm, whereas for a more narrow bearing with a relative width of the stress-carrying surface, i.e., L/R = 0.89 (diameter d = 0.315 m), the boundary at which the inertial forces are taken into consideration shifts toward the values of n = 2800-3000 rpm. The Reynolds numbers $Re = \omega R\delta/\nu$ in this case are, respectively, equal to 611 and 1118. The substantial difference between these two numbers bears witness to the fact that without consideration of the basic geometric and regime parameters, the Re number cannot serve as the solitary criterion of feasibility for taking the inertial forces of the lubricant into consideration. This is confirmed, moreover, by the fact that in the range of changes in the initial temperature of the lubricant from 30° to 50°C the Re number increases by a factor of 2.5, whereas the difference between the maximum and minimum increases in the carrying capacity amounts to less than 30% as a consequence of taking the inertial forces into account, and here the greatest increment occurs at T_0 = 40°C, i.e., when the Re number corresponds to the middle of the interval under consideration.

TABLE 1. Results from a Parametric Study of a Radial Slip Bearing

Variable parameter	Symbol	φ_c, rad	η_c	P_c, kN	$f_{fr}\cdot 10^2$	N, kW	G, kg-f	T^b_{max}, °C	Q^u_w	Q^l_w	$Q_{w\Sigma}$
									W		
$e=0,6$	R	0,522	0,358	44,2	5,23	95,7	8,16	61,1	1982	1582	3564
	J	0,522	0,37	45,8	4,89	92,6	8,16	61	1983	1584	3567
$e=0,7$	R	0,586	1,26	155,5	1,52	97,9	8,01	67,6	2482	1031	3513
	J	0,564	1,43	176,9	1,29	94,4	8,01	67,3	2471	1044	3515
$e=0,8$	R	0,564	2,46	304,5	0,8	101	8	79,9	3222	130	3351
	J	0,533	2,86	354,3	0,659	96,4	8	79,2	3248	70	3318
$e=0,85$	R	0,507	3,35	414,2	0,585	100	7,87	89,5	3972	865	3107
	J	0,507	7,77	466,4	0,493	95	7,87	89,3	3995	892	3102
$n=1000$	R	0,55	2,23	91,9	1,03	13	2,85	56,1	2401	1250	3650
	J	0,55	2,26	93,3	0,975	12,6	2,85	56,1	2403	1253	3656
$n=1500$	R	0,56	2,1	130,1	1,044	28	4,16	60,7	2530	1063	3593
	J	0,56	2,19	135,7	0,956	26,8	4,16	60,7	2531	1069	3600
$n=2000$	R	0,571	1,99	163,8	1,059	47,7	5,46	65,1	2637	908	3545
	J	0,553	2,15	177,7	0,934	45,7	5,46	64,9	2619	931	3550
$n=2500$	R	0,575	1,874	193,2	1,075	71,5	6,77	69,3	2722	786	3508
	J	0,563	2,115	218,1	0,913	68,6	6,77	69	2714	789	3503
$n=3000$	R	0,585	1,796	222,3	1,08	99,3	8,07	73,1	2783	677	3459
	J	0,552	2,104	260,2	0,888	95,6	8,07	72,4	2800	649	3449
$\varepsilon_h=0,75$	R	0,592	1,374	170	1,376	96,7	8,75	72,9	2352	988	3340
	J	0,554	1,585	196,2	1,142	92,6	8,75	72,3	2359	979	3338
$\varepsilon_h=1$	R	0,644	0,919	113,7	2,042	95,9	9,4	72,5	1948	1283	3231
	J	0,581	1,063	131,6	1,682	91,5	9,4	71,9	1954	1276	3231
$L/d=0,5$	R	0,591	1,09	84,6	1,106	61,6	5,73	71,5	1693	487	2180
	J	0,568	1,245	96,5	0,929	59,1	5,73	71	1690	495	2185
$L/d=0,65$	R	0,586	1,451	146,4	1,086	80,6	7,0	72,3	2236	594	2830
	J	0,578	1,703	169,7	0,887	77,2	7,0	71,9	2244	581	2825
$L/d=0,95$	R	0,58	2,079	306,5	1,114	118,5	9,21	73,7	3349	757	4106
	J	0,554	2,4	353,9	0,931	114,3	9,21	73,1	3362	734	4097
$\delta_v/\delta_h=0,4$	R	0,493	1,317	162,9	1,572	105,8	7,43	73,5	2505	860	3365
	J	0,493	1,472	182,5	1,358	102,2	7,43	73,4	2516	856	3372
$\delta_v/\delta_h=0,6$	R	0,663	2,016	249,4	0,908	93,5	8,72	72,4	2915	596	3511
	J	0,595	2,315	286,4	0,76	90	8,72	71,8	2899	573	3472
$T_0=30°C$	R	0,607	1,445	288,6	1,12	133	7,99	70,2	3273	708	3981
	J	0,534	1,644	328,2	0,929	126	7,99	69,6	3308	642	3950
$T_0=50°C$	R	0,57	2,145	170,9	1,043	73,6	8,23	76,4	2323	597	2920
	J	0,57	2,422	193	0,897	71,5	8,23	76,3	2323	595	2918
$\delta_h=0,3$ $e=0,6$	R	0,534	0,306	105,1	2,953	128,2	4,93	76,7	2248	1207	3455
	J	0,499	0,334	114,9	2,639	125,3	4,93	76,4	2273	1171	3444
$\delta_h=0,4$ $e=0,7$	R	0,596	1,17	226,1	1,204	112,5	6,52	75	2774	708	3482
	J	0,546	1,368	264,5	1,001	109,4	6,52	74,4	2837	628	3465
$\delta_h=0,6$ $e=0,8$	R	0,566	2,287	223,7	0,965	88,2	9,73	71	2677	745	3422
	J	0,566	2,596	256	0,799	83,7	9,73	70,8	2682	747	3430

Note. The symbol R denotes solution of noninertial lubricant flow with utilization of the Reynolds equation; J denotes the laminar inertial flow of the lubricant.

The investigations that have been carried out have demonstrated that the inertial effects promote an increase in the carrying capacity of the bearings, which may be treated as a positive fact for heavily loaded slip bearings. For lightly loaded high-velocity bearings situated near the boundary of dynamic stability, the appearance of inertial forces may become the cause for the transition of these bearings to regimes of unstable operation. In each and every case, the method developed here for slip bearings makes it possible more precisely to account for the actual physical processes occurring within the lubricant layers.

NOTATION

X, Y, Z, circumferential, radial, and axial coordinates; $x = X/R$, $y = Y/H = Y/h\delta$, $z = Z/R$, dimensionless coordinates; U, V, W, circumferential, lateral, and axial velocity components; $u = U/\omega R$; $v = V/\omega\delta$; $w = W/\omega R$, corresponding dimensionless velocity components; P, lubricant pressure; H, lubricant thickness; T, lubricant temperature; $p = P\psi^2/\mu_0\omega$; $h = H/\delta$, $t = T/T_0$, dimensionless values for lubricant pressure, thickness, and temperature; $r = R_j/R$, dimensionless radial coordinate; R_j, radius of the j-th section of the shaft or bearing; $\overline{\mu} = \mu/\mu_0$; $Re = \omega\delta^2/\nu$; $Ec = \omega^2R^2/cT_0$; $\psi = \delta/R$; T_0, lubricant temperature at the inlet to the bearing; R, shaft radius; δ, radial clearance (space); ω, angular speed of shaft rotation; μ, ν, c, dynamic and kinematic viscosity, as well as the specific heat capacity of the lubricant oil; $\ell = L/R$, relative width of the bearing; x_k, coordinate of the point at which the lubrication layer separates from the shaft; α, heat-transfer coefficient; T_a, temperature of the ambient medium; T_s, temperature of the heat-exchange surface; φ_c, load angle; $\eta_c = P_c\psi^2/\mu\omega RL$, dimensionless load factor; P_c, carrying capacity of the bearing; $f_{fr} = N/P_cU$, coefficient of shaft resistance to rotation; N, force of friction; G, lubricant flow rate; T_{max}^b, maximum temperature at the working surface of the bearing; Q_W^u, Q_W^ℓ, $Q_{W\Sigma}$, flows of heat to the shaft surface, respectively, for the upper and lower halves, as well as the total flow.

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