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INFLUENCE OF GAS BUBBLES ON NONLINEAR DYNAMIC CHARACTERISTICS OF THE OIL FILM OF A TILTING PAD BEARING

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The influence of a comparatively low volume concentration of gas microbubbles contained in oil on nonlinear characteristics describing the behavior of an oil film in the guide gap of a hydrodynamic tilting pad bearing under action of a low-frequency harmonic force is analyzed using a numerical dynamic model of a collar–oil film–pad system. It is shown that bubbles in the oil greatly affect the efficiency of the tilting pad bearing. Results of oil-film-dynamics investigations reported previously (including those of the present author) are generalized.

Introduction. Recently the problems concerned with the substantial growth of alternating axial loads onto hydrodynamic tilting pad bearings in engineering units have become pressing. Under such conditions a comparatively large amount of air bubbles often appears in the lubricant layer, which can cause failure of a tilting pad bearing due to "turning-over" of the oil wedge [1, 2, etc.]. Therefore investigation of the effect of gas bubbles on the behavior of a collar–oil film–pad system is of both practical and theoretical interest.

The subject matter of the present work includes investigation of the nonlinear dynamic characteristics of an oil film with gas microbubbles in the quide gap of a tilting pas bearing and generalization and refinement of the linear model of a similar process described in [3]. In an approximation of a thin lubricant layer calculations are made of the pressure components, deformations, displacements, and temperature field of a tilting pad that are in response to an external alternating harmonic low-frequency force of a comparatively large amplitude. To describe pressure and temperature distributions, we use the unsteady Reynolds equation and the energy equation for an adiabatic lubricant layer. The Rayleigh–Plesset equation is used to determine the radii of oscillating spherical bubbles moving along the stream lines of the carrier medium (here the results reported previously in [3-6, etc.] are slightly modified).

To analyze the influence of layer microbubbles, we calculated the nonlinear coefficients of effective elasticity and damping of the support, which allow for the dynamic response of the bubbles to pressure variation in the film. As might be expected, these characteristics are practically independent of the size of the spherical bubbles, since with small microbubble radii of the order 10^{-6} m (as adopted in the calculations) the frequency of their natural oscillations attains, according to [7], approximately 1 MHz, i.e., differs from the prescribed loading frequency of 75 Hz by several orders. With such a frequency difference, the characteristic time of support loading is lower by approximately several orders as compared with the period of natural oscillations of a bubble. Owing to this fact, the behavior of bubbles in an alternating pressure field, as shown in [7], is quasistatic and described by the degenerate steady-state Rayleigh–Plesset equation. Also, the influence of the volume concentration of bubbles on support efficiency is analyzed.

Mathematical Formulation of the Problem and the Basic Design Formulas. The following system of equations for translational displacement of collar and rotational motion of a pad in tangential and radial directions under the action of an external force, hydrodynamic forces and constraint reactions of rolling friction of a lock bolt of the pad with a caulking ring was presented earlier [4, 6]:

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$$\int\int_{\Omega} p dx dz = N ,$$

$$\int\int_{\Omega} \left[p (x - x_c) + i_1 \left(- \frac{h}{2} \frac{\partial p}{\partial x} + \frac{\eta}{h} \right) \right] dx dz + k_1 \, |N|^{4/3} \operatorname{sgn} \dot{\varphi} = 0 , \tag{1}$$

$$\int\int_{\Omega} \left[p (z - z_c) - i_2 \frac{h}{2} \frac{\partial p}{\partial z} \right] dx dz + k_2 \, |N|^{4/3} \operatorname{sgn} \dot{\psi} = 0$$

at $\dot{\varphi} \neq 0$, $\dot{\psi} \neq 0$ and

$$\left| \int\int_{\Omega} \left[p (x - x_c) + i_1 \left(- \frac{h}{2} \frac{\partial p}{\partial x} + \frac{\eta}{h} \right) \right] dx dz \right| < k_1 \, |N|^{4/3} ,$$

$$\left| \int\int_{\Omega} \left[p (z - z_c) - i_2 \frac{h}{2} \frac{\partial p}{\partial z} \right] dx dz \right| < k_2 \, |N|^{4/3} \tag{2}$$

at $\dot{\varphi} = 0$ and $\dot{\psi} = 0$.

The coefficients i_1 and i_2 from (1) and (2), which take into account the hydrodynamic friction forces applied to the pad are calculated by the formulas

$$i_1 = H_1 H_{pc} / L^2 , \quad i_2 = H_1 H_{pc} / B^2 ,$$

while the dimensionless coefficients k_1 and k_2, which allow for rolling friction of the lock bolt, are described by the relations

$$k_1 = \delta_1 \delta_2 / L , \quad k_2 = \delta_1 \delta_2 / B ,$$

where

$$\delta_1 = (0.29 - 0.58) \cdot 10^4 \sqrt[3]{\left(R_\delta / E' \right)} ; \quad \delta_2 = 10^{-4} \sqrt[3]{\left(Q / (z_p \Lambda_0) \right)} ;$$

$$E' = 2 E_1' E_2' / (E_1' + E_2') ; \quad \Lambda_0 = \int_\Omega \int_0 p_0 dx dz ;$$

$$E_k' = E_k / (1 - \nu_k^2) \quad (k = 1 , \ 2) .$$

The dimensionless layer thickness is determined by the formula

$$h = h_c - (x - x_c) \varphi - (z - z_c) \psi + \Delta ,$$

in which the dimensionless deformations Δ of bodies under friction are approximated, according to [4, 8, 9], by the expression

$$\Delta = \chi \, (kN + \delta_t) ,$$

wherein the form function χ is assumed to be dependent only on the coordinates of the bearing surface of the pad and equal to zero beneath the supporting point of the pad and to unity at the points of intersection of the coordinates $x = x_c$ and $z = z_c$ with the pad contour.

The coefficient k of force deformation in the case of point supporting of the pad is calculated by the formula [8]

$$k = \frac{\delta_0 Q}{z_p H_p E H_1 \Lambda_0} ,$$

and the coefficient of thermal deformation δ_t is described by the relation

$$\delta_t = \frac{\alpha_{\exp} L^2}{8H_p} \left[0.5 + \frac{2(1+\nu)}{5 + 3\nu + \dfrac{1-\nu}{\lambda^2}} \right] \frac{\Delta T_{m0}}{H_1},$$

which follows from [9].

The pressure and temperature distributions in the oil film of the guide gap of the tilting pad bearing are determined by simultaneous solution of the unsteady-state Reynolds equations obtained in [3] for a two-phase medium and energy transfer for an adiabatic lubricant layer [10]:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \lambda^2 \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right) = \frac{1}{2}\frac{\partial(\rho h)}{\partial x} + \frac{\partial(\rho h)}{\partial \tau}, \tag{3}$$

$$h\frac{\partial\theta}{\partial\tau} + q_x\frac{\partial\theta}{\partial x} + \lambda q_z\frac{\partial\theta}{\partial z} = \frac{h^3}{12\eta\rho}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\lambda\frac{\partial p}{\partial z}\right)^2\right] + \frac{\eta}{h\rho}, \tag{4}$$

where

$$q_x = \frac{h}{2} - \frac{h^3}{12\eta}\frac{\partial p}{\partial x}; \quad q_z = -\lambda\frac{h^3}{12\eta}\frac{\partial p}{\partial z};$$

$$\theta = (T - T_{\text{oil}})/T_1; \quad T_1 = \mu_{\text{oil}}UL/(\text{P}C_pH_1^2).$$

The boundary conditions for these equations are the initial oil temperature T_{oil} at the inlet of the lubricant layer and equality of the pressures at the boundary of the carrier part of the layer to zero. It is pertinent to note that the indicated boundary can fail to coincide with the pad contour in the case of layer separation. The initial conditions for the temperatures are sought by solution of the relevant steady-state problem.

The density and viscosity distributions of the film containing gas bubbles in a thermodynamic polytropic process in a bubble are described by the formulas [3]:

$$\rho = 1/(1 + \beta_a r^3), \quad \eta = (1 - \beta)\exp(-\alpha T_1\theta),$$

where

$$\beta = \left(\frac{p_a + 2}{p_a + k_s p + \dfrac{2}{r}}\right)^{1/\gamma}\beta_a; \quad p_a = R_a P_a/\sigma;$$

$$k_s = \mu_{\text{oil}}ULR_a/(H_1^2\sigma),$$

while the dimensionless radius of a spherical bubble is determined from the Rayleigh–Plesset equation [11]:

$$\frac{1}{r^{3\gamma}}(p_a + 2) - p_a - \frac{2}{r} = k_s p, \tag{5}$$

in which the inertia and viscosity terms are dropped based on the work [1].

By analogy with [4, 5], a solution of Reynolds Eq. (3) is sought in the form

$$p = p_1 + p_2\dot{h}_c + p_3\dot{\varphi} + p_4\dot{\psi} + p_5 k\dot{N}, \tag{6}$$

while the functions p_j are determined by the elimination method [12] from the system of equations

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p_j}{\partial x}\right)+\lambda^2\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p_j}{\partial z}\right)=f_j,$$

where

$$f_1=\frac{1}{2}\frac{\partial(\rho h)}{\partial x}+h\frac{\partial\rho}{\partial\tau};\quad f_2=\rho;\quad f_3=(x_c-x)\rho;$$

$$f_4=(z_c-z)\rho;\quad f_5=\chi\rho.$$

In this case, an external dimensionless load that counterbalancies the hydrodynamic reaction of the layer between the collar and the pad is described by

$$N=\Lambda_0(1+\alpha_F\sin\omega\tau).$$

To solve energy Eq. (4), we used the implicit running-count finite-difference scheme [13], while for Rayleigh–Plesset Eq. (5) the method of dividing in half was adopted.

Using relation (6), the system of Eqs. (1) and inequalities (2) is reduced to the form

$$a_{11}\dot{h}_c+a_{12}\dot{\varphi}+a_{13}\dot{\psi}=N-a_{14}k\dot{N}-\Lambda,$$

$$a_{21}\dot{h}_c+a_{22}\dot{\varphi}+a_{23}\dot{\psi}=b_1-a_{24}k\dot{N}-k_1|N|^{4/3}\operatorname{sgn}\dot{\varphi},$$

$$a_{31}\dot{h}_c+a_{32}\dot{\varphi}+a_{33}\dot{\psi}=b_2-a_{34}k\dot{N}-k_2|N|^{4/3}\operatorname{sgn}\dot{\psi},$$

where

$$|a_{21}\dot{h}_c+a_{23}\dot{\psi}+a_{24}k\dot{N}-b_1|<k_1|N|^{4/3},\quad |a_{31}\dot{h}_c+a_{32}\dot{\varphi}+a_{34}k\dot{N}-b_2|<k_2|N|^{4/3},$$

$$a_{1m}=\iint_\Omega p_{m+1}dxdz;\quad \Lambda=\iint_\Omega p_1dxdz;\quad b_1=\iint_\Omega\left[p_1(x_c-x)+i_1\left(\frac{h}{2}\frac{\partial p_1}{\partial x}-\frac{\eta}{h}\right)\right]dxdz;$$

$$b_2=\iint_\Omega\left[p_1(z_c-z)+i_2\frac{h}{2}\frac{\partial p_1}{\partial z}\right]dxdz;$$

$$a_{2m}=\iint_\Omega\left[p_{m+1}(x-x_c)-i_1\frac{h}{2}\frac{\partial p_{m+1}}{\partial x}\right]dxdz;$$

$$a_{3m}=\iint_\Omega\left[p_{m+1}(z-z_c)-i_2\frac{h}{2}\frac{\partial p_{m+1}}{\partial z}\right]dxdz\quad(m=1,2,3,4),$$

is then solved by the iteration technique. Counting is continued on each time layer until a relative error of the pressures, differing from zero, at nodes of the finite-difference network becomes smaller in two subsequent iterations than a specified comparatively small number. In each iteration, system of Eqs. (7) is solved by the Runge–Kutta method [14].

In numerical solution of the problem, we determined the region Ω of the carrier part of the layer from the values of the nonnegative pressures at the nodes of the finite-difference network, the dynamic parameters h_c, φ, and ψ of the collar–film with bubbles–pad system, the maximum temperature of oil superheating ΔT_m, and the minimum layer thickness h_m along the mean pad radius and at the lubricant outlet from the gap as functions of the dimensionless time τ as well as the nonlinear coefficients of effective elasticity and damping of the pad by the formulas

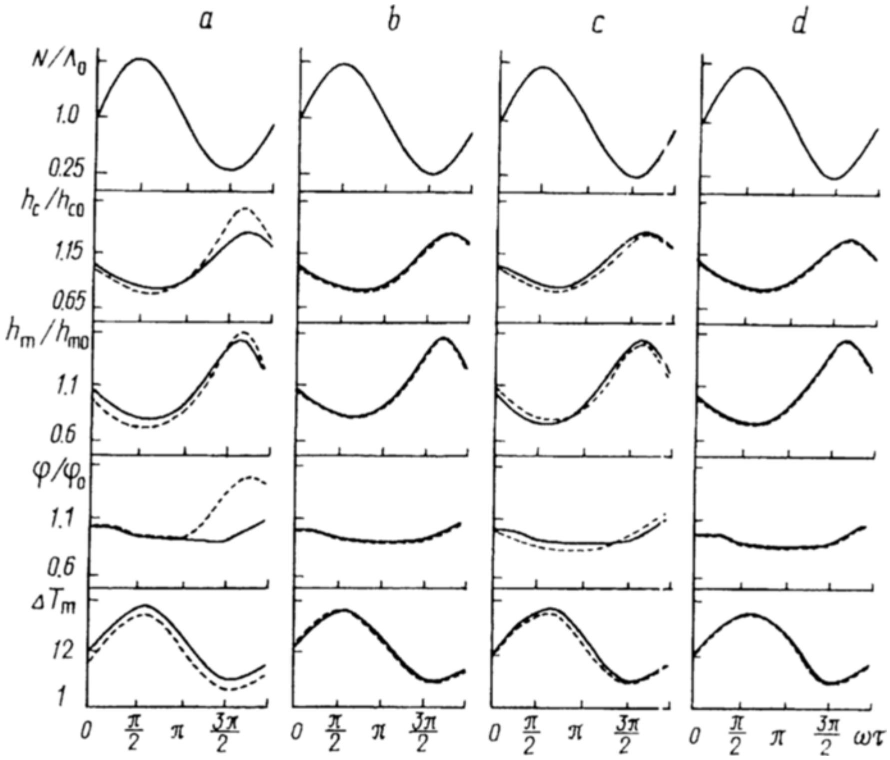

Fig. 1. Relations describing layer behavior under unsteady operating conditions of tilting pad bearing: a) β_a effect: $R_a = 0.5\ \mu$m; $\delta_1 = 0.23$; $z_c = 0.5$; dashed curves: $\beta_a = 0$, $x_c = 0.6088$; solid curves: $\beta_a = 0.2$, $x_c = 0.6069$; b) R_a effect: $\beta_a = 0.2$; $\delta_1 = 0.23$; $x_c = 0.6069$; $z_c = 0.5$; solid curves: $R_a = 0.5\ \mu$m; dashed curves: $R_a = 5\ \mu$m; c) δ_1 effect: $R_a = 0.5\ \mu$m; $\beta_a = 0.2$; $x_c = 0.6069$; $z_c = 0.5$; dashed curves: $\delta_1 = 0$; solid curves $\delta_1 = 0.23$; d) z_c effect: $R_a = 0.5\ \mu$m; $\beta_a = 0.2$; $x_c = 0.6069$; dashed curves: $z_c = 0.5$ solid curves: $z_c = 0.515$. ΔT_{oil}, ^{o}C.

$$K_e = \frac{Q}{L\Lambda_0} \sqrt{\left(\frac{Q}{\mu_{oil}\, z_p U B \Lambda_0}\right)}\ C \cos \varphi_{load}\,,$$

$$K_d = \frac{Q}{\omega' L\Lambda_0} \sqrt{\left(\frac{Q}{\mu_{oil}\, z_p U B \Lambda_0}\right)}\ C \sin \varphi_{load}\,.$$

Results of Numerical Simulation and Discussion. Numerical calculations were performed for a hydrodynamic tilting pad bearing with the following initial data: $L = B = 7.9$ cm; $\mu_{oil} = 0.043$ Pa·sec; $\rho C_p = 1.76$ MJ/(m^3·K); $P_a = 0.1$ MPa; $\sigma = 0.003$ N/m; $\alpha = 0.0336$ 1/K; the polytropic index $\gamma = 1$; number of pads $z_p = 8$; the mean pad radius is 13.5 cm; the pad thickness is $H_p = 2.5$ cm; the distance from the point of contact of the lock bolt to the bearing surface of the pad is $H_{pc} = 3.5$ cm; the curvature radius of the bearing bolt is $R_\delta = 20$ mm; the speed of collar rotation is 4500 rpm; the static load is $Q = 0.09$ MN; Young's modulus is $E = 2.1 \cdot 10^5$ MPa; the Poisson coefficient is $\nu = 0.3$; the temperature coefficient of linear expansion is $\alpha_{exp} = 0.12 \cdot 10^{-4}$ 1/K. The function χ is described by the relation $\chi = (1 - 2x)^2 + (1 - 2z)^2$, and the coefficient of force deformation δ_0 was assumed to be equal to 0.25, according to [8].

All calculations were performed using a 9 × 9 finite-difference network with time step $\Delta\tau = \pi/(60\omega)$ at the prescribed dimensionless amplitude $\alpha_F = 0.75$ and a harmonic-strength frequency of 75 Hz and different coefficients of rolling friction of the lock bolt k_1 and k_2 corresponding to $\delta_1 = 0$ and $\delta_1 = 0.23$.

To study the influence of gas bubbles on the dynamic parameters of the layer h_c, h_m, φ, ψ, and ΔT_m the concentration coefficient β_a was assumed to be equal to 0 and 0.2.

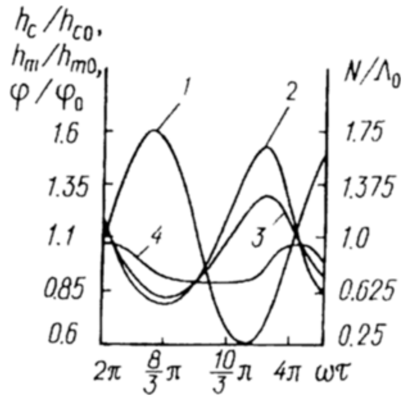

Fig. 2. Relations describing layer behavior under steady operating of tilting pad bearing: 1) N/Λ_0; 2) h_m/h_{m0}; 3) h_c/h_{c0}; 4) φ/φ_0; $x_c = 0.6069$; $z_c = 0.5$; $\beta_a = 0.2$.

In numerical calculations, the dimensionless coordinates x_c and z_c of point support of the pad were maintained with an accuracy of $2 \cdot 10^{-3}$.

Figure 1 illustrates the behavior of an oil film with bubbles in the case of unsteady functioning of the tilting pad bearing. An analysis of the curves in this figure reveals a substantial change in the dimensionless parameters of the layer $h_c \neq h_{c0}$, h_m/h_{m0} and φ/φ_0 with air bubbles in the oil (see Fig. 1a). However, the shape of the curves is practically independent of bubble radius (see Fig. 1b). It should be also noted that friction in the support also influences the dynamic characteristics of a lubricant film [6] but to a considerably lesser degree than the volume content of air bubbles (Fig. 1c).

From Fig. 1d we can judge the influence of friction in a tilting pad bearing installed with small eccentricity in the radial direction. In this case, the form of the curves remains unchanged; $\psi = 0.25$ and is independent of time.

Figure 2 shows dimensionless relations that describe the behavior of a layer with gas bubbles after cessation of the first period of oscillations. They correspond to practically steady operation of the tilting pad bearing. Using these relations, the nonlinear coefficients of effective elasticity K_e and damping K_d of the lubricant film were found to be $K_e = 0.168 \cdot 10^{10}$ N/m and $K_d = 0.274 \cdot 10^7$ N·sec/m. For comparison, we give the values of these coefficients calculated by the linear model [3]: $K_e = 0.170 \cdot 10^{10}$ N/m and $K_d = 0.270 \cdot 10^7$ N·sec/m. The insignificant difference between their values is explained by the quasisteadiness of the lubrication process adopted in the models under low-frequency loading and by the small variables of force deformations of the pad as compared with static thermal deformations (for the given tilting pad bearing their ratio at the lubricant exit from the gap and the mean radius of the pad does not exceed 0.22).

SUMMARY

1. A generalized model was constructed to study the dynamic characteristics of an oil film containing gas microbubbles.

2. It was shown that the presence of microbubbles in a lubricant layer greatly affects the law of variation of the angle of pad turning in the tangential direction and the thickness of the layer beneath the supporting point and at the point of lubricant exit from the gap.

3. The moment of rolling friction of the lock bolt of the pad im[airs the support efficiency to a substantially lesser degree than does the volume content of gas microbubbles in the layer.

NOTATION

$x = X/L$, $z = Z/B$, dimensionless coordinates; X, Z, coordinates; L and B, pad length and width; λ, pad length-to-width ratio; H_1, characteristic thickness of the film; $p = PH_1^2/(\mu_{oil} = UL)$, dimensionless manometric pressure; P, manometric pressure; U, circumferential velocity of the collar over the mean pad radius; $\tau = Ut/L$, dimensionless time; t, time; $\beta = Q/Q_1$ and $\eta = \mu/m_{oil}$, dimensionless true density and dynamic viscosity coefficient of the film; Q, C_p, μ, density, specific heat, dynamic viscosity coefficient of the film; Q_1, true density of the pure oil; $\bar{\theta}$, dimensionless excess temperature; T, temperature; T_1, characteristic temperature; ΔT_{oil}, maximum superheating of the oil over the mean pad radius; $r = R/R_a$, dimensionless bubble radius; R, bubble radius; σ, surface tension; γ, polytropic index; β, ratio of the gas volume in the form of bubbles to the oil volume without bubbles; α and α_{exp}, temperature coefficients of viscosity and linear expansion of the pad base, respectively; $\varphi = L\Phi/H_1$, $\psi = B\Psi/H_1$, dimensionless angles of pad turning; Φ, Ψ, angles of pad turning; N, dimensionless harmonic force balancing the hydrodynamic reaction of the layer; α_F, dimensionless amplitude of the harmonic load; Q, static load on the tilting pad bearing; z_p, quantity of pads; H_p, pad thickness; H_{pc}, distance from the bearing plane of the pad to the point of intersection of the symmetry axis of the lock bolt with its spherical surface; R_δ, curvature radius of the lock bolt; E_1 and ν_1, E_2 and ν_2, E and ν, Young's modulus and Poisson coefficient of the lock bolt, caulking ring of the lock bolt and material of the pad, respectively; δ_0, coefficient of force deformation (determined as in [8]); C, ratio of the dimensionless amplitude of the harmonic load balancing the hydrodynamic reaction of the layer between the collar and the pad to the dimensionless amplitude of periodic oscillations of the collar; φ_{load}, angle of phase shift between acting harmonic load and steady periodic oscillations of the collar; $\omega = \omega' L/U$, dimensionless angular frequency of periodic oscillations; ω', angular frequency of periodic oscillations; Ω, region covered by the film. Subscripts: c, quantities referred to the supporting point of the pad; oil, quantities referred to the oil without bubbles at the gap inlet; a, quantities at atmospheric pressure; m, quantities for the lubricant at the gap outlet and over the mean pad radius; 0, steady-state values; $\cdot$, derivative with respect to time.

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