CALCULATION OF HYDROSTATIC PLASTIC MEDIUM-LUBRICATED THRUST BEARINGS

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A procedure is described to calculate operation parameters of hydrostatic plastic medium-lubricated thrust bearings. A comparison is made of predicted and experimental characteristics of bearing operation for plastic and Newtonian lubrication media.

Hydraulic friction pairs working in a liquid lubrication mode find a wide application in present-day mechanical engineering. Nowdays new plastic lubricants are used in hydrostatic bearings which make it possible to increase a load-carrying capacity of a bearing and to decrease lubricant consumption.

Theoretical solutions are known of the problem on radial flow of a viscoplastic medium obeying the Schvedov–Bingham [1-4] and Herschel–Balkley [5, 6] equations.

It is experimentally established [7] that a rheological equation of the type

$$\tau = \tau_0 + \eta_0 \exp\left(-\frac{\tau - \tau_0}{G_0}\right)\dot{\gamma}. \tag{1}$$

fits the experimental data on the flow of plastic lubricants in a wide range of shear rates.

Consider the plastic lubricant flow adhering to Eq. (1) in a thrust bearing clearance. We neglect rotation of bearing surfaces, considering that they rotate very slowly. A bearing consists of upper 1 and lower 2 disks (Fig. 1). In the central part of the lower disk, resting on fixed base 4, provision is made for hole 3 to supply a lubricant. A tangential stress profile in a lubricant layer has the form of 5 and 6, respectively. Inner and outer radii of the disks are r_2 and r_1, while a clearance between the disks is equal to 2H. A load F is applied to the upper disk.

To solve the problem, we set aside a ring element with a width Δr and a mean radius r_j, as shown in Fig. 1. Assume that within the limits of the element a height of a quasisolid zone h_0 and a pressure gradient are constant. For reasons of the flow symmetry, we solve the problem for the upper part of a radial slot.

An equation for the lubricant motion is of the form

$$\frac{\partial \tau}{\partial z} = -\frac{\partial p}{\partial r}. \tag{2}$$

Having integrated (2), we arrive at

$$\tau = -\frac{\partial p}{\partial r} z + C_1. \tag{3}$$

Using the flow symmetry condition $\tau_{(z=0)} = 0$, we find $C_1 = 0$.

From the condition of quasisolid core equilibrium, we may write the following relation:

$$\frac{\partial p}{\partial r} = \frac{\tau_0}{h_0}, \tag{4}$$

Next, from (3) and (4) it follows that the stress distribution over a section with a width Δr is

$$\tau = \tau_0 \frac{z}{h_0}. \tag{5}$$

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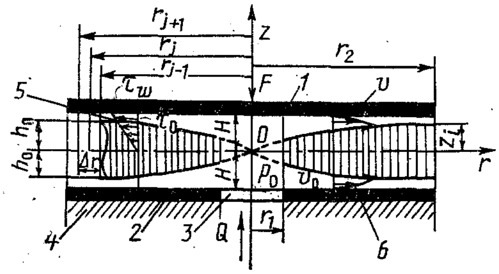

Fig. 1. Schematic of the nonlinear-viscoplastic lubricant flow in a hydrostatic thrust bearing.

Substituting the stress distribution (5) into Eq. (1), we obtain the expression:

$$\frac{dv}{dz} = \frac{\tau_0}{\eta_0}\left(\frac{z}{h_0} - 1\right)\exp\left[\frac{\tau_0\left(\frac{z}{h_0} - 1\right)}{G_0}\right], \tag{6}$$

integration of which yields

$$v = \frac{G_0}{\eta_0}\exp\left[\frac{\tau_0}{G_0}\left(\frac{z}{h_0} - 1\right)\right]\left[h_0\left(1 + \frac{G_0}{\tau_0}\right) - z\right] + C_2. \tag{7}$$

Having determined the integration constant C_2 from the condition of lubricant adherence to the disk $v_{(z=h)} = 0$, we obtain the velocity distribution in a gradient-flow zone ($h_0 \le z \le 1$):

$$v = \frac{G_0}{\eta_0}\left\{\exp\left[\frac{\tau_0}{G_0}\left(\frac{z}{h_0} - 1\right)\right]\left[h_0\left(1 + \frac{G_0}{\tau_0}\right) - z\right] - \right.$$
$$\left. - \exp\left[\frac{\tau_0}{G_0}\left(\frac{h}{h_0} - 1\right)\right]\left[h_0\left(1 + \frac{G_0}{\tau_0}\right) - h\right]\right\}. \tag{8}$$

A quasisolid core velocity v_0 is determined from (8) at $z = h_0$. A dimensionless height h_0 of the core is calculated from the continuity equation

$$Q = 4\pi r\int_0^h vdz = 4\pi r\left(h_0v_0 + \int_{h_0}^h vdz\right) = 4\pi r\left(h_0v_0 + \int_{h_0}^h vz - \int_{h_0}^h zdv\right) = -4\pi r\int_{h_0}^h zdv. \tag{9}$$

Substituting dv from (6) into (9), we obtain

$$Q = -r\int_{h_0}^h z\frac{\tau_0}{\eta_0}\left(\frac{z}{h_0} - 1\right)\exp\left[\frac{\tau_0\left(\frac{z}{h_0} - 1\right)}{G_0}\right]dz. \tag{10}$$

After integration and some transformations, we arrive at a formula for calculation of lubricant consumption

$$Q = 4\pi r\frac{h_0^2}{\eta_0}\frac{G_0^2}{\tau_0}\left\{1 - 2\frac{G_0}{\tau_0} + \exp\left[\frac{\tau_0}{G_0}\left(\frac{h}{h_0} - 1\right)\right] \times \right.$$
$$\left. \times\left[\left(1 - \frac{\tau_0 h}{G_0 h_0}\right)\left(1 - \frac{h}{h_0} + \frac{G_0}{\tau_0}\right) + \frac{G_0}{\tau_0}\right]\right\}. \tag{11}$$

An expression to calculate a load-carrying capacity is of the form

$$F = 2\pi\sum_{j=1}^{k-1}\int_{r_{j-1}}^{r_{j+1}}\left[p_{j+1} + A_j(r_{j+1} - r)\right]rdr. \tag{12}$$

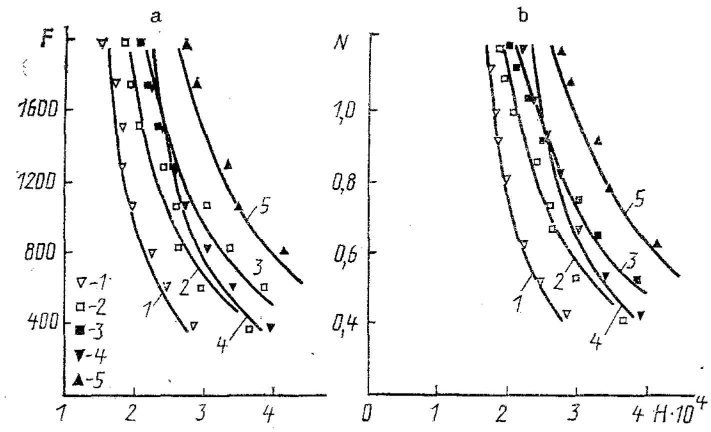

Fig. 2. Calculated curves and experimental points of the load-carrying capacity of the bearing (a) and power consumption for lubricant pumping (b) as a function of the working clearance: 1) IGP-30; 2) IGP-30 + 2.5% of ceresin-65; 3) IGP-30 + 5% of ceresin-65; 4) IRP-75; 5) IRP-75 + 2.5% of ceresin-65. F, N; H, m; N, W.

After integration we obtain

$$F = 2\pi \left[\sum_{j=1}^{k-1} p_{j+1} \frac{r_{j+1}^2 - r_{j-1}^2}{2} + \right.$$

$$\left. + \sum_{j=1}^{k-1} \frac{A_j}{6} (3r_{j+1}^3 - 3r_{j-1}^2 r_{j+1} - 2r_{j+1}^3 + 2r_{j-1}^3) \right]. \tag{13}$$

Load-carrying characteristics of the bearing have been calculated for lubricants listed in Table 1.

Rheological constants for plastic lubricants have been determined by a rotary viscosimeter "Rheotest RV," while for industrial oils – on a capllary viscosimeter VPZh-3 at a temperature of 31°C. The constants values are listed in Table 1.

In order to experimentally check the applicability of the rheological equation (1) for calculation of hydrostatic thrust bearings, an experimental set-up has been designed to include a hydrostatic thrust bearing with the central chamber, having the radius $r_1 = 0.035$ m and outer radius $r_2 = 0.1$ m, a lubrication system, and a temperature-control system. Loading on the bearing was created by calibrated loads via a loading lever system.

Radial pressure distribution was fixed at four points on the radius by standard pressure gauges of the accuracy class 0.6, the lubricant temperature was measured by chromel–copel thermocouples at five points on the radius. The temperature of the lubricant was 31°C and maintained constant.

Figure 2a represents the calculated and experimental data on the load-carrying capacity of the bearing as a function of a working clearance for tested lubricants at their constant flow rate of $1.51 \cdot 10^{-5}$ m^3/sec. With the working bearing load and lubricant consumption being constants, the application of plastic lubricants ensures a 2 to 2.5-fold increase of the working clearance of the bearing that allows reduction of technological requirements for supporting surfaces to be manufactured. A divergence between calculated and experimental data does not exceed 10%.

Figure 2b displays the calculated and experimental data on the power consumed for lubricant pumping via the bearing as a function of a working clearance at the same lubricant flow rate. A divergence between the calculated and experimental values also does not exceed 10%.

If one compares the power consumed for lubricant pumping of all grades at the same load-carrying capacity of the bearing then a conclusion may be drawn from the analysis of Fig. 2 that the application of plastic lubricants allows reduction of power consumption by 7-10% as compared to unthickened oils. This is attributed to a more uniform pressure distribution over the radial surface of the bearing in the radial direction for plastic lubricants. Such a character of the pressure distribution is confirmed by experimental data.

Thus, Eq. (1) describes well the rheological properties of plastic lubricants and may be recommended for rheodynamic calculations of hydrostatic thrust bearings.

TABLE 1. Rheological Constants of Tested Lubricants

Lubricant	Composition	Rheological constants			
		G_0, Pa	η_0, Pa·sec	τ_0, Pa	μ, Pa·sec
Industrial oil	IGP-30				0,0189
Industrial oil	IRP-75				0,0503
Plastic lubricant	IGP-30+2,5% ceresin-65	288,7	0,0463	12,0	
Plastic lubricant	IGP-30+5% ceresin-65	238,0	0,0774	18,0	
Plastic lubricant	IRP-75+2,5% ceresin-65	222,1	0,1782	13,0	

NOTATION

τ_0, η_0, G_0, rheological constants; τ, $\dot{\gamma}$, stress and shear rate, respectively; p, pressure; z, axial coordinate; r, radial coordinate; r_{j-1}, r_j, r_{j+1}, inner, mean and outer radii of the j-th ring; $A_j = \partial P/\partial r$, pressure gradient on the section numbers j; v, velocity; Q, flow rate; h_0, half-height of the quasi-solid core on the section numbered j; h, half-height of the bearing clearance; k, quantity of ring areas into which the bearing base is subdivided during calculations; F, load-carrying capacity of the bearing; μ, lubricant viscosity.

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