EFFECT OF THE RHEOLOGICAL PROPERTIES OF A LUBRICANT ON THE CARRYING CAPACITY OF HYDROSTATIC BEARINGS

V. O. Yablonskii, N. V. Tyabin, and V. M. Yashchuk

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On the basis of computational results an analysis is performed of the influence of the rheological properties of plastic lubricants on the carrying capacity of a hydrostatic bearing.

To describe the rheological properties of plastic lubricants, being structurized disperse systems, until recently use was made of either the Shvedov–Bingham equation [1] or the Herschel–Bulkley one [2]. An approximation method for flow curves of plastic lubricants, proposed by Tyabin and Vinogradov [1], makes it possible to represent a flow curve by a broken line and to describe the rheological properties by a set of constants. However in this case it is difficult to analyze the effect of the rheological properties of a lubricant on the main parameters of a lubricant layer.

For plastic lubricants A. I. Makorchevskii and V. A. Skurchinskii [3] recommended the following rheological equation

$$\tau = \tau_0 + \eta_0 \exp\left[-(T - \tau_0)/G_0\right] \gamma, \tag{1}$$

which for simple shift may be represented in dimensionless form as

$$\bar{\tau} = 1 + \exp\left[-(\bar{\tau} - 1)/L\right] \bar{\gamma}. \tag{2}$$

The values of the rheological constants of this equation for some plastic lubricants at a temperature of 30°C are given in Table 1.

In engineering practice, for estimating the pressure distribution over the hydrostatic bearing surface it is convenient to use the load capacity coefficient [4], calculated by the formula

$$\omega_f = \frac{F}{\pi p_{\mathrm{ch}} r_{11}^2} \tag{3}$$

An increase of the load capacity coefficient corresponds to a decrease of power expenditures for lubricant pumping-through at a constant load on the bearing.

Works [5, 6] give computational results on the carrying capacity coefficient of hydrostatic bearings for lubricants whose rheological properties are described by the Herschel–Bulkley equation.

However, in view of the fact that the flow of plastic lubricants in hydrostatic bearings within the mode of slow displacements is characterized by low shear velocities, it is to calculate the carrying capacity coefficient for the rheological equation (1).

Figure 1 presents the design diagram of a hydrostatic thrust bearing, which consists of upper 1 and lower 2 disks. The lower disk at the central part is provided with hole 3 (to supply lubricants) and rests upon fixed base 4. The figure also shows the velocity profile in lubricant layer 5, stress distribution sheet 6, and quasisolid core boundary 7. The load is applied to the upper disk. We neglect rotation of the bearing surfaces, assuming that we have a slowly rotating bearing.

The lubricant rotation equation takes the form

$$\frac{\partial \tau}{\partial z} = -\frac{\partial p}{\partial r}. \tag{4}$$

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TABLE 1. Rheological Constants of Plastic Lubricants

Name of lubricant	τ_0, Pa	G_0, Pa	η_0, Pa·sec	L
GOI-54p	45.3	702.4	1.13	15.5
TsIATIM-203	112.4	905.3	5.39	8.1
Solid oil C	195.0	1120.6	12.79	5.8
I-13	597.5	1212.8	125.1	2.0
Lithol-24	780.3	933.2	10.65	1.2

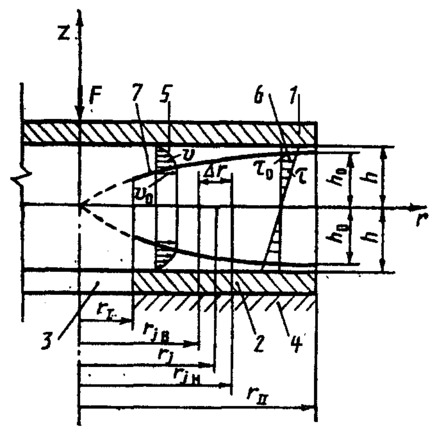

Fig. 1. Design diagram of lubricant flow in a hydrostatic thrust bearing.

The pressure gradient is determined from the equilibrium condition of the quasisolid core

$$\frac{\partial p}{\partial r} = \frac{\tau_0}{h_0}. \tag{5}$$

Upon integrating (4), using the flow symmetry condition $\tau_{(z=0)} = 0$ and taking into account relation (5), we find the stress distribution

$$\bar{\tau} = \frac{\bar{z}}{\bar{h}_0}, \tag{6}$$

and substituting it into rheological equation (2), we obtain a differential equation which describes the velocity distribution in the gradient flow region. Performing integration of it under the boundary condition of lubricant adhesion to the disks $v_{(z=1)} = 0$, we determine the velocity distribution

$$\bar{v} = L\left\{\exp\left[\frac{1}{L}\left(\frac{\bar{z}}{\bar{h}_0} - 1\right)\right][\bar{h}_0(1+L) - \bar{z}] - \exp\left[\frac{1}{L}\left(\frac{1}{\bar{h}_0} - 1\right)\right][\bar{h}_0(1+L) - 1]\right\}. \tag{7}$$

The quasisolid core velocity is calculated by formula (7) for

$$\bar{z} = \bar{h}_0 : \bar{v}_0 = \bar{v}_{(\bar{z}=\bar{h}_0)}.$$

The dimensionless lubricant consumption is defined in the following manner:

$$\bar{Q} = \bar{r}\int_0^1 \bar{v}d\bar{z} = \bar{r}\left(\bar{v}_0\bar{h}_0 + \int_{\bar{h}_0}^1 \bar{v}d\bar{z}\right). \tag{8}$$

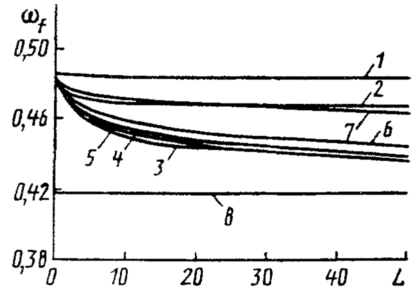

Fig. 2. Dependence of the bearing carrying capacity coefficient ω_f on parameter L at different lubricant consumptions Q for the bearing when $r_I = 0.035$ m, $r_{II} = 0.1$ m, $h = 1.5 \cdot 10^{-4}$ m: 1) Q = 0.01; 2) 0.1; 3) 1.0; 4) 3.0; 5) 5.0; 6) 10.0; 7) 100.0.

Substituting v and v_0 from (7) into (8) and integrating, we finally obtain

$$\bar{Q} = \bar{r}L^2\bar{h}_0^2\left\{1 - 2L + \exp\left[\frac{1}{L}\left(\frac{1}{\bar{h}_0} - 1\right)\right] \times \left[\left(1 - \frac{1}{L\bar{h}_0}\right)\left(1 - \frac{1}{\bar{h}_0} + L\right) + L\right]\right\}. \tag{9}$$

In order to carry out the calculations, we subdivide the whole bearing surface into ring elements of width Δr with mean radius τ_j, as shown in Fig. 1. Within the element limits the quasisolid region height h_0 and the pressure gradient are taken to be constant. The quasisolid core height for each ring element with number j is determined by the numerical solution of Eq. (9) at the prescribed value of dimensionless consumption Q.

The pressure distribution over the bearing surface is calculated by the formula

$$\bar{p}_{j_\mathrm{i}} = \bar{p}_{j_\mathrm{o}} + \bar{A}_j\Delta\bar{r}, \tag{10}$$

where the dimensionless pressure gradient $\bar{A}_j = 1/h_0$.

We find the mean pressure on the bearing surface in the following fashions:

$$\bar{p}_\mathrm{m} = \bar{p}_\mathrm{ch}\bar{r}_\mathrm{I}^2 + \sum_{j=1}^{N} \bar{p}_j(\bar{r}_{j\mathrm{o}}^2 - \bar{r}_{j\mathrm{i}}^2). \tag{11}$$

and calculate the bearing carrying capacity coefficient by the formula

$$\omega_f = \frac{\bar{p}_\mathrm{m}}{\bar{p}_\mathrm{ch}}. \tag{12}$$

Figure 2 illustrates the calculation results for the carrying capacity coefficient, depending on L, for different values of consumption Q as applied to a bearing with the geometrical parameters $r_I = 0.035$ m, $r_{II} = 0.1$ m and $h = 1.5 \cdot 10^{-4}$ m.

A decrease in parameter L corresponds to an increase in τ_0 and to a reduction in G_0, i.e., to the anomalous amplification of the non-Newtonian lubricant properties. From Fig. 2 we can see that with variation of L from 50 to 0.3 the value of ω_f changed from 0.43 to 0.48, i.e., increased by $\approx 12\%$; this corresponds to an equivalent reduction of power expenditures for lubricant pumping-through with an unchanged bearing carrying capacity.

When applying a non-Newtonian lubricant to a bearing of the given geometry, the carrying capacity coefficient is equal to 0.42 and is independent of lubricant viscosity (curve 8 in Fig. 2). Consequently, use of non-Newtonian lubricants in hydrostatic bearings provides an increase of the carrying capacity coefficient, and therefore it is necessary to employ lubricants with a more pronounced anomaly; this fact allows us to decrease the power expenditures for lubricant pumping-through by 10-15%.

NOTATION

τ_0, η_0, G_0, rheological constants; τ, γ, stress and shear velocity, respectively; $\tau = \tau/\tau_0$, $\gamma = \gamma\eta_0/\tau_0$, dimensionless tangential stress and shear velocity; T, intensity of stresses; ω_f, bearing carrying capacity coefficient; F, load; p_{ch}, pressure in the central chamber of the bearing; p_m, mean pressure at the bearing surface; r_{II}, outer radius of the disks; r_I, central chamber radius; $r = r/r_{II}$, dimensionless radial coordinate; $L = G_0/\tau_0$, $s = h/r_{II}$, dimensionless parameters; h, half-height of the gap between the disks; h_0, half-height of the quasisolid core; h_0, dimensionless half-height of the quasisolid core; z, axial coordinate; $z = z/h$, dimensionless axial coordinate; v, velocity; $v = v\eta_0/(\tau_0 h)$, dimensionless velocity; Q, lubricant consumption; $Q = Q\eta_0/(4\pi\tau_0 h^2 r_{II})$, dimensionless flow rate; j, number of the ring element; p, pressure; $p = p(s/\tau_0)$, dimensionless pressure; $\bar{A}_j = \partial p/\partial r$, dimensionless pressure gradient in the j-th region; N, number of ring elements; $r_{j\,o}$, $r_{j\,i}$, outer and inner radii of the ring element, respectively.

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