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Research Paper

The use of a displacement compensator for improve the static characteristics of a step hydrostatic thrust bearing

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ABSTRACT: *The paper proposes an improved technical solution of a step hydrostatic thrust bearing with an active displacement compensator using an external support ring on an elastic suspension. The results of mathematical modeling and theoretical research of stationary operating modes of the structure presented. The possibility of improving its static characteristics by reducing the compliance shown, which allows eliminating the main disadvantage of stepped bearings – high compliance. It is shown that with a decrease in compliance, the lubricant flow rate also decreases, which indicates an increase in the efficiency of a thrust bearing with a movement compensator.*

KEYWORDS: *stepped hydrostatic thrust bearing, displacement compensator, load characteristics, compliance, lubricant flow rate.*

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I. INTRODUCTION

The advantage of step hydrostatic bearings is that, in comparison with similar throttle bearings, they are structurally simpler and have a higher bearing capacity [1, 2, 3]. However, at the same time, they characterized by increased compliance [4].

To reduce compliance in bearings, active compensation of lubricant flow rate used, where membrane regulators of the nozzle-flap type [5-8] or Laub elastic orifices [9-11] used as inlet flow compensators in aerostatic bearings. The use of membrane or elastic regulators allows the static compliance to be reduced to zero or even negative values (in the latter case, the load and gap increments have the same signs).

Bearings with diaphragm or resilient adjusters have at least three disadvantages. The first is that they are too energy intensive due to the need to significantly increase the lubricant flow rate using the input regulators to reduce compliance. Such bearings have insufficiently stable characteristics of bearing capacity, when low compliance provided only in a narrow range of loads [9, 10]. The third drawback arises from the first – the large gain of the regulators has an increased negative effect on the bearing dynamics when it is difficult to ensure low compliance [11].

The preferred means of reducing compliance is the use of compensators for the movement of the movable element (shaft) [12-14]. Such bearings also allow for a decrease in compliance to zero and negative values, which makes it possible to use them not only as supports, but also as automatic compensators for deformation of the technological system of machine tools in order to reduce the time and increase the accuracy of metalworking.

The article discusses the results of a theoretical study of the static characteristics of a hydrostatic thrust bearing with a compensator of this type.

Figure 1 shows a schematic of an advanced hydrostatic center thrust bearing.

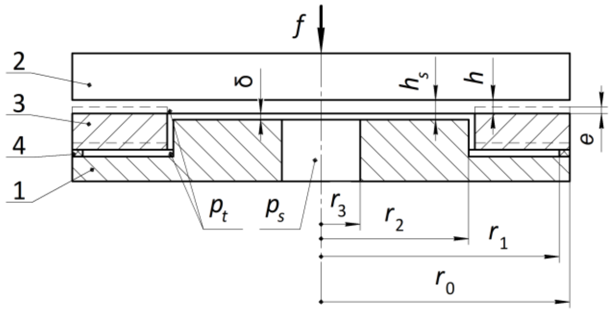

Figure 1. Calculation scheme of the step thrust bearing

The structure has a shaft 2 and a base 1, which hermetically connected to a rigid ring 3 of the inner radius r_2 and the outer radius r_0 by means of an elastic ring 4 of the inner radius r_1. The thrust bearing powered by a lubricant injection source under pressure p_s through an opening of radius r_3. In the absence of pressure between the surfaces of the base 1 and ring 3, a step of height δ formed. In working condition, a lubricant film of thickness h created between the surfaces of ring 3 and shaft 2. Between the base 1 and the ring 3 there is a radial annular gap, through which the grease without resistance enters the blind cavity 5 with a pressure $p_k < p_s$, which is formed as a result of overcoming the hydraulic resistance to the flow in the gap of thickness hs on the ring $r_3 < r < r_2$. The size of this gap determined by the sum

$$h_s = h + \delta + h_e, \tag{1}$$

where h_e is the deformation of the elastic ring 4 caused by the action of the difference in hydrostatic forces on the working surfaces of the ring 3.

With a rigid ring 4, the bearing works like a conventional thrust bearing. If the ring is elastic and has axial compliance $k_e > 0$, then under the influence of hydrostatic forces on the surface of the disk 3, the deformation of the disk 4 occurs, and at certain values of the radius of the structure, the displacement of the disk 3 in the direction opposite to the action of the external force f is ensured. As a result, the gap h_s increases, which entails a decrease in the compliance of the thrust bearing.

The paper considers a mathematical model of the stationary state of the support and calculates the static characteristics of the thrust bearing – load capacity, compliance and flow rate.

II. MODEL OF THE STATIONARY STATE OF THE BEARING

The study of the characteristics of the step thrust bearing was carried out in a dimensionless form. The scales of values are taken: p_s for pressures, r_0 for radial dimensions, gap h_0 for gaps, step size and deformation of ring 4, $\dfrac{h_0^3 p_s}{6\mu}$ for volumetric lubricant flow rates, $2\pi r_0^2 p_s$ for axial forces, where h_0 corresponds to the gap h in the thrust bearing, which perceives the calculated load f_0, μ is lubricant viscosity. Further, dimensionless quantities are designated by attributed Latin letters.

The static model will include the equation for the balance of flow rates at the input and output of the stage

$$Q_1 - Q_2 = 0, \tag{2}$$

the equations of the power balance of the shaft

$$W = F, \tag{3}$$

ring 3

$$H_e = K_e W_e \tag{4}$$

and dimensionless analogue (1) for the total gap

$$H_s = H + \Delta + H_e. \tag{5}$$

Here flow rates

$$Q_1 = \lim_{R \to R_2 - 0} H_s^3 R \frac{dP}{dR}, Q_2 = \lim_{R \to R_2 + 0} H^3 R \frac{dP}{dR}, \tag{6}$$

hydrostatic forces

$$\begin{gathered} W_1 = \int_0^{R_3} RPdR, W_2 = \int_{R_3}^{R_2} RPdR, \\ W_3 = \int_{R_2}^{R_1} RPdR, W_4 = \int_{R_2}^{1} RPdR, \end{gathered} \tag{7}$$

$$W = W_1 + W_2 + W_4, \tag{8}$$

the difference in hydrostatic forces acting on the surface of the ring 3

$$W_e = W_3 - W_4. \tag{9}$$

The pressure distribution function in the gaps obeys the stationary Reynolds differential equation

$$\frac{d}{dR}\left(R\frac{dP}{dR}\right) = 0 \tag{10}$$

with boundary conditions

$$P(R_3) = 1, P(R_2) = P_t, P(1) = 0, \tag{11}$$

where P_t is the pressure at the junction of the step and the ledge ($R = R_2$).

Substituting the solution to problem (10), (11) into integrals (7), we obtain

$$\begin{gathered} W_1 = A_{0,}\ W_2 = A_1 + A_2 P_t, \\ W_3 = A_3 P_t, W_4 = A_4 P_t, \end{gathered} \tag{12}$$

where

$$A_0 = \frac{R_3^2}{2}, A_1 = R_3^2\left[2Z^2 + \frac{\left(1 - Z^2\right)}{4\text{Ln}Z}\right],$$

$$A_3 = \frac{R_1^2 - R_2^2}{2}, A_4 = \frac{\left(R_2^2 - 1\right)}{4\text{Ln}R_2} - \frac{R_2^2}{2}, Z = \frac{R_2}{R_3}.$$

Substituting (12) into (8) and (9), we find

$$W = A_5 + A_6 P_t, W_e = A_7 P_t, \tag{13}$$

where

$$A_5 = A_0 + A_2, A_6 = A_1 + A_4, A_7 = A_3 - A_4.$$

Equation (2) takes the form

$$H_s^3(1 - P_t) = A_8 H^3 P_t, \tag{14}$$

where $A_8 = \dfrac{\text{Ln}\left(R_3 / R_2\right)}{\text{Ln}R_2}$.

III. DEFINITION OF THE PARAMETERS OF THE «DESIGN POINT»

The «design point»" means the point on the load curve $H_s(F)$ for the gap $H = 1$, which corresponds to the dimensional design gap $h = h_0$. For the point, we will adjust the pressure at the junction of the step and the protrusion using the normalized parameter $\chi \in [0,1]$ according to the formula $P_t = \chi.$. As input parameters, we will also use all the radii and the coefficient of axial compliance K_e of the elastic ring 4. Using (14) taking into account (4), (5), we will determine the calculated deformation H_{e0}, design gap H_{s0}, step height and bearing capacity W.

$$H_{e0} = A_7\chi, H_{s0} = \frac{A_9\chi}{(1-\chi)}, \Delta = H_{s0} - 1 - H_{e0}, W_0 = A_5 + A_6\chi. \tag{15}$$

where $A_9 = \sqrt[3]{A_8}$.

A necessary condition for reducing the compliance K of the thrust bearing is the positiveness of the hydraulic force W_e. From (9), (12) it follows that this takes place when $A_3 > A_4$. Using it, we can calculate the smallest value of the radius $R_{1,\min} = \sqrt{\dfrac{R_2^2 - 1}{2\text{Ln}R_2}}$, at which the specified condition will be met.

To determine the compliance K of the bearing, we take into account that

$$K = -\frac{dH_s}{dW} = -\frac{dH_s}{dP_t} / \frac{dW}{dP_t}. \tag{16}$$

Performing differentiation (14) and (13) and substituting its results into (16), we find a formula for calculating the compliance

$$K = \frac{3A_7A_8K_eP_t - H\left(A_8 + A_{10}\right)}{3A_6\left[A_{11}\left(1 - P_t\right) - A_8P_t\right]}, \tag{17}$$

where $A_{10}=\dfrac{A_8 P_t}{1-P_t}, A_{11}=A_{10}{}^{2/3}$.

Using (17), one can find the coefficient of elasticity at which the thrust bearing will have zero compliance K = 0 at the design point. Substituting H = 1 and $P_t = \chi$ in (17) and equating the numerator of the expression to zero, after simple transformations we find

$$K_{e0}=\frac{4}{3}\left[\left(2R_1^2+\frac{1-R_2^2}{\mathrm{Ln}R_2}\right)\chi(1-\chi)\right]^{-1}. \tag{18}$$

Formula (18) is convenient to use for calculating the static characteristics of the bearing capacity, compliance and lubricant consumption, setting the elasticity coefficient K_e in proportion to the value of K_{e0}. So, for $K_e = K_{e0}$, the thrust bearing at the design point will have zero compliance K = 0, for $0 \le K_e < K_{e0}$ – positive, and for $K_e > K_{e0}$ – negative.

IV. STATIC CHARACTERISTICS OF THE BEARING

After determining the parameters of the «design point», we can start calculating the characteristics of the thrust bearing. It is convenient to carry out calculations in parametric form, taking the pressure P_t as a parameter. For the current pressure, using formulas (8) and (12), one can find the bearing capacity W and the hydraulic force W_e. Using formula (4), one can find the deformation of H_e. Using (14), one can find the gap

$$H=\frac{\Delta+H_e}{A_{12}-1}, \tag{19}$$

where $A_{12}=\sqrt[3]{\dfrac{P_t \mathrm{Ln}Z}{(1-P_t)\mathrm{Ln}R_2}}$.

Now we can determine the gap H_s and the flow rate Q

$$H_s=H+\Delta+H_e, Q=-\frac{H^3 P_t}{\mathrm{Ln}R_2}.$$

When calculating the characteristics, it is necessary to take into account that the denominator (19) can vanish. This takes place at the minimum operating pressure $P_{t,\min}=\dfrac{\mathrm{Ln}R_2}{\mathrm{Ln}R_3}$. Consequently, when calculating the characteristics, the pressure P_t, as a parameter, should be varied in the range $P_{t,\min}<P_t<1$.

Figure 2 shows the graphs of load characteristics for different values of the coefficient of elasticity K_e. Parameters χ = 0.7 are fixed; R_1 = 0.95; R_2 = 0.85; R_3 = 0.1.

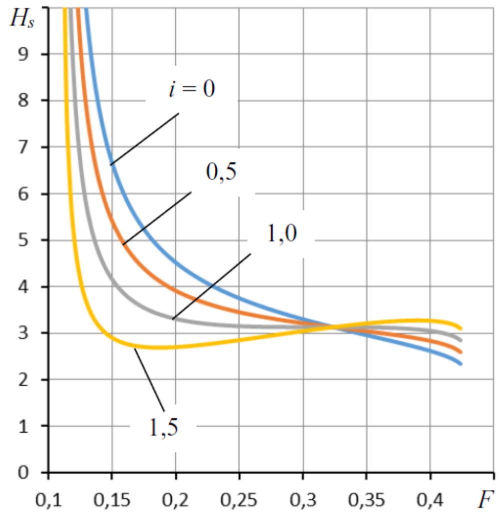

Figure 2. Load characteristics of the thrust bearing at different values of the coefficient of elasticity $K_e = i\ K_{e0}$.

The common point of the curves corresponds to the «design point» mode. The graphs show that with an increase in the elasticity coefficient K_e, the nature of the curves changes. In this case, in the region of low and moderate pressures, with an increase in this parameter, the compliance K of the thrust bearing decreases, as can be judged from the change in the slope of the curves.

For $K_e = K_{e0}$ (i = 1) at the design point, the thrust bearing acquires zero compliance, and for $K_e > K_{e0}$ (i > 1) – negative.

The observed effect most clearly demonstrated by the curves of the dependence of the compliance on the external force, which are shown in Figure 3. It can be seen that with an increase in K_e, the compliance K decreases in the range of working loads. At $K_e = K_{e0}$, the thrust bearing reaches zero compliance at the design point. At $K_e > K_{e0}$, the compliance curves reach negative values. In the range of loads corresponding to negative compliance, the thrust bearing is able to carry loads and compensate for the elastic deformations of the machine.

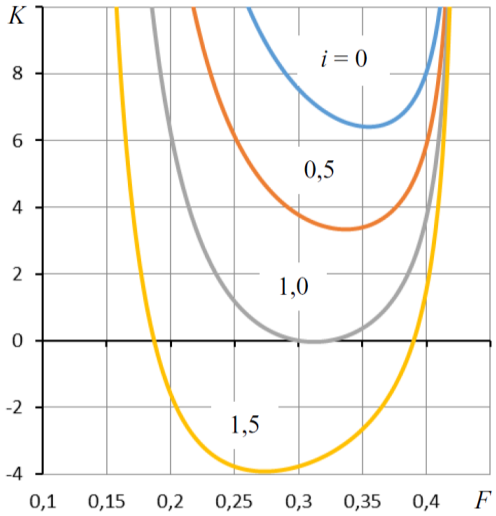

Figure 3. Dependences of compliance K on external force F at different values of the elasticity coefficient $K_e = i\, K_{e0}$.

Figure 4 shows the flow rate characteristics of the step thrust bearing.

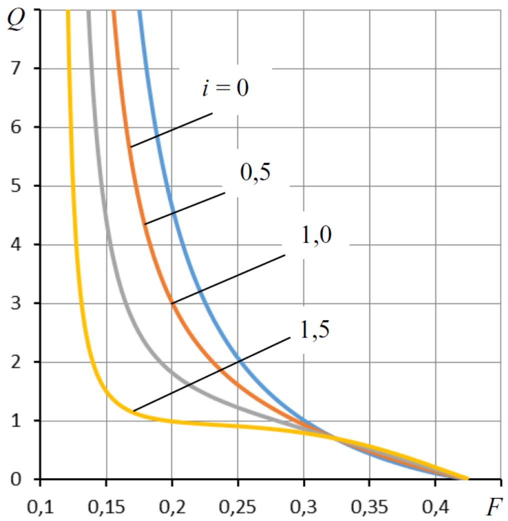

Figure 4. Flow rate characteristics of the thrust bearing at different values of the elasticity coefficient $K_e = i\, K_{e0}$.

It is seen that with an increase in the coefficient of elasticity K_e. In the region of low and moderate loads, the flow rate decreases, which indicates an increase in the efficiency of the structure.

V. CONCLUSION

The paper proposes an improved technical solution of a stepped hydrostatic thrust bearing with active regulation of the lubricant output flow through the use of an external support ring on an elastic suspension. The results of mathematical modeling and theoretical research of stationary operating modes of the structure presented. The possibility of improving its static characteristics by reducing the compliance shown, which allows eliminating the main disadvantage of stepped bearings - high compliance. It shown that with a decrease in compliance, the lubricant flow rate also decreases, which indicates an increase in the efficiency of the thrust bearing with active compensation of movement.

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