Quality of Dynamics of Gas-static Thrust Bearing with Movable Carrying Circle on Elastic Suspension

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A B S T R A C T

The aim of the work is to present a construction, a mathematical model and a method for calculating the indicators quality dynamics of a gas-static thrust bearing with a carrying center on an elastic suspension. It is shown that the applying this improvement completely eliminates the significant shortcomings of the quality dynamics of a thrust bearing with self-compensation. It turns the design into a dynamic system with optimal dynamic characteristics - high stability indicators, aperiodic nature of transients, and oscillatory index values which are specific for ideally damped dynamical systems.

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1. INTRODUCTION

It is known that gas-static bearings with self-compensation (with "annular diaphragms") are absolutely stable [1 - 3], but the quality of their dynamics has a number of serious drawbacks. These include low speed, increased oscillatory transients, big amplitudes of the resonance frequency response of the transfer function of dynamic compliance. This is due to the negative effect of the volume of compressed gas in gap on the thrust bearing damping.

It is possible to improve the dynamic characteristics of the thrust bearing by means of improvements aimed at reducing the volume of

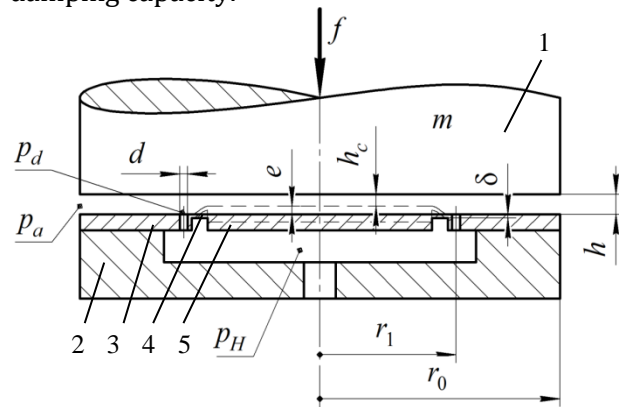the bearing lubricating film and increasing its damping capacity.

Fig. 1. The design scheme of the thrust bearing

Figure 1 shows a scheme of a gas-static thrust bearing (GB) with shaft 1 and base 2, which is connected to the supporting disc 3 of radius r_0. GB uses lubrication from a source of compressed gas through openings of small diameter d, which are evenly located on the disk 3 around a circle of radius r_1. The central movable carrying disc 5 is supported by the elastic suspension 4 in the form of a thin ring of thickness δ providing the required amount of deformation e of the suspension material.

During operation of the thrust bearing, the lubricant is fed into it under pressure of p_H through the hole in the base 2, then through the annular diaphragms into the carrier gap and then flows out of it into the environment. Under the action of the pressure difference $p_H - p_d > 0$ on the surface of the center 5, the ring 4 is deformed, as a result of this center is displaced by the value of e in the direction of the shaft 1. Compared with the usual thrust bearing ($e = 0$), its carrier gap will be less, moreover, due to the deformation of the material of the ring 4, the nature of oscillations of the carrier gap of lubricant will change, which may contribute to the improvement of the design dynamics.

2. MATH MODELING

The study of the quality of the GB dynamics was carried out in dimensionless form. Dimensional values of the mathematical model are indicated by lower case letters, dimensionless – upper case. The scales of dimensionless quantities are taken: outer radius r_0 – for radii; the corresponding static load f_0 (stationary mode of the so called state of a "starting point") thickness h_0 of the lubricant gap on the outer ring of the disk 3 – for displacements, pressure p_a of the environment – for pressures, $\pi r_0^2 p_a$ – for forces.

The mathematical model describes the movement of compressed gas in the areas of the lubricating gap formed by the surfaces of shaft 1 and center 5 (central region $0 \le r \le r_1$) and the outer ring of disk 3 (annular region $r_1 \le r \le r_0$). The areas are in contact around a circle of radius r_1, on which the annular diaphragms are located. The pressure function in the lubricant gap of these regions obeys the system (1) – (2) boundary value problems for the Reynolds differential equation [7]

$$\begin{cases} \dfrac{\partial}{\partial R}\left(R H_c^3 P_c \dfrac{\partial P_c}{\partial R} \right) = \sigma R \dfrac{\partial \left(P_c H_c \right)}{\partial \tau}, \\ \dfrac{\partial P_c}{\partial R} = 0, P_c(R_1, \tau) = P_d(\tau), P_c(R, 0) = 1, \end{cases} \quad (1)$$

$$\begin{cases} \dfrac{\partial}{\partial R}\left(R H^3 P_r \dfrac{\partial P_r}{\partial R} \right) = \sigma R \dfrac{\partial \left(P_r H \right)}{\partial \tau}, \\ P_r(R_1, \tau) = P_d(\tau), P_r(1, \tau) = 1, P_r(R, 0) = 1, \end{cases} \quad (2)$$

where $P_c(R, \tau)$ and $P_r(R, \tau)$ – pressure functions in lubricant gaps of areas; $H_c(\tau)$ and $H(\tau)$ are functions of the thickness of the gaps in these areas; $P_d(\tau)$ is the pressure function at the outlet of the annular diaphragms; R, τ – radius and current time.
Here

$$\sigma = 12 \mu r_0^2 / p_a h_0^2 t_0 \quad (3)$$

– "number of compression" of the gas gap [8], where μ is the dynamic viscosity of the gas lubricant, t_0 is the current time scale.

To determine the unknown pressure $P_d(\tau)$, the continuity equation of the lubricant flow was used

$$Q_r(\tau) - Q_c(\tau) = Q_d(\tau), \quad (4)$$

where

$$Q_r = -R H^3 \left.\dfrac{\partial P_r^2}{\partial R}\right|_{R=R_1}, Q_c = -R H_c^3 \left.\dfrac{\partial P_c^2}{\partial R}\right|_{R=R_1},$$

$$Q_d = A_d H \Psi\left(P_H, P_d\right) \quad (5)$$

– functions of the mass flow rate of gas at the entrance to the gaps of the corresponding regions and at the exit from the annular diaphragms, where A_d is the similarity criterion of the feeding holes in the Prandtl outflow function [8].

The force balance equation of shaft 1 was represented as

$$W(\tau) - F_{in}(\tau) = F(\tau), \quad (6)$$

where F is the external force, $W = W_r + W_c$ is the bearing capacity of the thrust bearing,

$$W_r(\tau) = 2 \int_{R_1}^{1} R \left(P_r - 1 \right) dR,$$

$$W_c(\tau) = 2 \int_{0}^{R_1} R \left(P_c - 1 \right) dR,$$

$$F_{in}(\tau) = M \frac{d^2 H(\tau)}{d^2 \tau} \qquad (7)$$

– components of the carrying capacity of the areas of the gap and the inertia force of the shaft 1, where M – its mass.

The displacement ε (τ) of center 5 and the function of the gap Hc (τ) were found by the formulas

$$W_H = R_1^2 (P_H - 1), \varepsilon = K_m (W_H - W_c),$$
$$H_c = H - \varepsilon, \qquad (8)$$

where K_m is the elasticity of the elastic ring 4.

The study of the dynamics of the thrust bearing was carried out for small oscillations of the shaft 1 in the vicinity of the aforementioned state of a "starting point" using a specialized computer simulation environment, calculation and research of gas-static bearings (SIGP environment) [8] by methods of the theory of linear dynamic systems [8, 9]. The solution of boundary value problems (1), (2) for the linearized and Laplace-transformed Reynolds equations is obtained by numerical method [7], which guarantees the specified accuracy of the calculation of complex coefficients with integro-differential images of the generalized coordinates of the dynamic model (1) - (8).
For a quantitative assessment of the stability and speed of the GB as a dynamic system, the degree of stability η was used [8]. The stability margin of the GB was estimated using the oscillatory index β [7] of the amplitude-frequency characteristic of the transfer function of the dynamic compliance of the thrust bearing $K(s) = \overline{\Delta H}(s) / \overline{\Delta F}(s)$, where $\overline{\Delta H}, \overline{\Delta F}$ are the Laplace transform of small deviations of the corresponding functions from their stationary values, s is the variable of the Laplace transform [8, 9].

The following parameters were used as input: supply pressure P_H, radius R_1, "squeeze number" σ and ε_0 – static shift of center 5and normalized diaphragm resistance adjustment factor $\chi = (P_{d0}^2 - 1)/(P_H^2 - 1) \in [0,1]$, where P_{d0} is the static lubricant pressure at the outlet annular diaphragms with a static gap thickness $H_0 = 1$ (state of a "starting point").

The scale t_0 was determined from the condition

$$M = \frac{m h_0}{\pi r_0^2 p_a t_0^2} = 1, \qquad (9)$$

where m is the mass of the shaft 1.

The pressure P_{d0}, the criterion A_d and the compliance K_m in the "starting point" mode, for which problems (1) – (2) have an analytical solutions, were determined by the formulas

$$P_{d0} = 1 + \sqrt{\chi (P_H^2 - 1)}, \quad A_d = \frac{1 - P_{d0}^2}{\Psi(P_H, P_{d0}) \ln R_1},$$

$$K_m = \frac{\varepsilon_0}{R_1^2 (P_H - P_{d0})}.$$

3. RESEARCH RESULTS

Figure 2 shows the dependences of the static compliance K_0 of the thrust bearing on the setting factor χ at various values of the supply pressure P_H. As the parameters ε_0 and σ do not affect the static characteristics of the thrust bearing, so the meanings of the coefficient χ are significant and at which the thrust bearing has the least static compliance. The function K_0 (χ) is unimodal and, therefore, has a unique minimum, which, as can be seen from the graphs, corresponds to $\chi \approx 0.45$. The calculation of the dynamic characteristics carried out for this value χ.
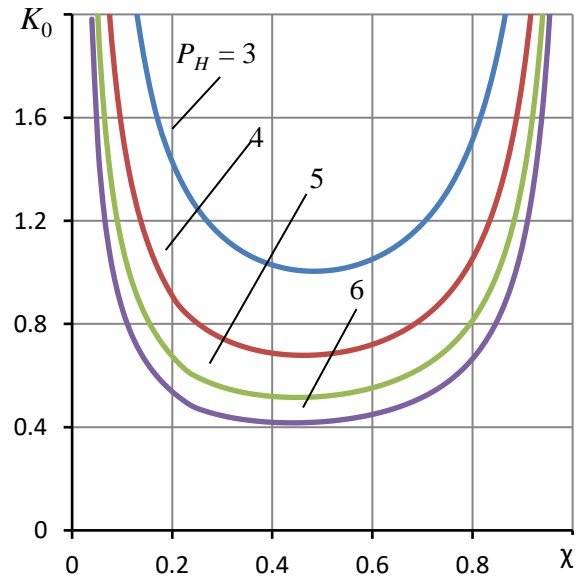

Fig. 2. Curves of static compliance K_0 as a function of the setting factor χ for different values of supply pressure P_H at $R_1 = 0.7$

Figure 3 shows the effect of the static displacement ε_0 of center 5, which is equivalent to the effect of K_m compliance of ring 4, on the character of the transient process $\Delta H(\tau)$ with the input force $\Delta F(\tau)$ as a δ-function of Dirac [7].
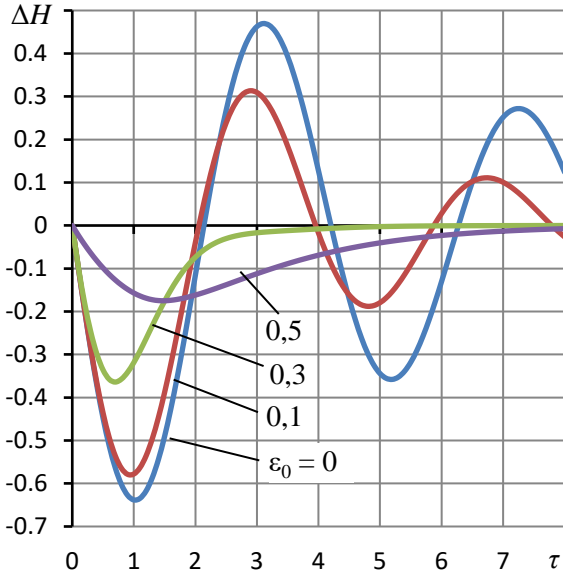

Fig. 3. Curves ΔH of the transition process for different displacements ε_0 of center 5 at $P_H = 5$; $\chi = 0.45$; $R_1 = 0.7$; $\sigma = 20$

With a hard disk 5, when $\varepsilon_0 = 0$ and $K_m = 0$, the transition curve is characterized by distinct oscillation. With an increase in ε_0 and K_m the oscillation of the transition curves decreases, which is reflected in a decrease in the amplitude of the oscillations. Already at $\varepsilon_0 \geq 0.3$, when in a static state, the thickness of the gap in the central region due to the displacement of the center 5 decreases by at least 30% and thereby contributes to a noticeable decrease in the volume of lubricant in the gas gap, the transient process becomes aperiodic. For such modes the duration of transient processes decreases.

A more complete picture of the duration of the transient characteristics is provided by the curves of Fig. 4, which show the effect of the displacement ε_0 on the speed of the thrust bearing. The graphs show that with increasing ε_0, the degree of stability η quickly increases, reaches its maximum and then decreases, which indicates the extreme nature of the dependence $\eta(\varepsilon_0)$. It was established that the σ values that are to the left of this maximum correspond to oscillatory modes of transient processes, and to

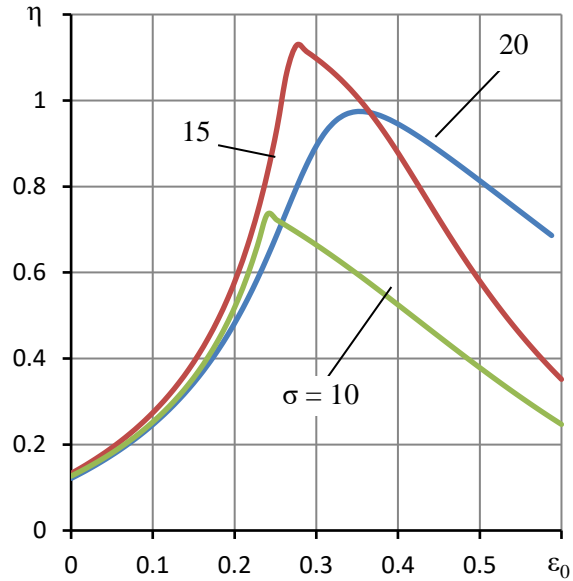the right – aperiodic ones, which is confirmed by the graphs shown in Fig. 3.

Fig. 4. Curves of the degree of stability η from the displacement ε_0 of the movable center 5 for different values of the "compression number" σ at $P_H = 5$; $\chi = 0.45$; $R_1 = 0.7$
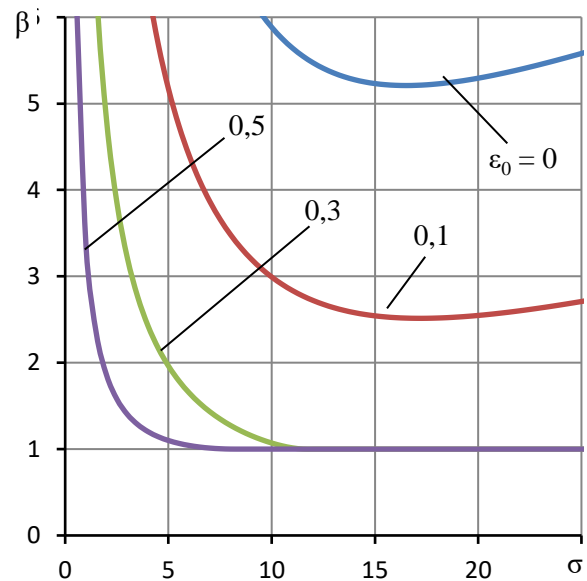

Fig. 5. Curves of the oscillation index β from the "number of compression" σ for different displacements ε_0 of the moving center at $P_H = 5$; $\chi = 0.45$; $R_1 = 0.7$

The analysis of the curves in Fig. 4 indicate that the function $\eta(\sigma)$, and, therefore, the function $\eta(\varepsilon_0, \sigma)$ also has an extremal character and has a unique extremum maximum. It determines the optimal mode of the speed of the thrust bearing. As a result of the optimization of the function

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$\eta(\varepsilon_0, \sigma)$, it has been established that the thrust bearing has a maximum response speed when ε_0 = 0.31 and σ = 13, which corresponds to the extremum η ($\varepsilon0$, σ) = 1.52. A similar thrust bearing with a rigid ring 4 (ε_0 = 0) would have the largest value η = 0.13. From this it follows that the improvement applied in the thrust bearing, with an optimal choice of parameters affecting only the dynamics of the construction, makes it possible to increase its speed more than an order of magnitude.

Figure 5 shows the dependence of the oscillation index β on the "number of compression" σ for various displacements ε_0. To the heel with a rigid ring 4 (ε_0 = 0) there corresponds a curve on $\beta >$ 5, which characterizes the heel as a resonant system of too high oscillation.

With increasing ε_0, the index β decreases and already at $\varepsilon_0 \geq 0.2$, the thrust bearing acquires the properties of a well-damped dynamic system, for which the indicator β should not exceed values of 1.1 – 1.5 [7]. As can be seen from the graphs, when $\varepsilon_0 \geq 0.3$, the dependences β (σ) become resonanceless (β = 1), and the thrust bearing becomes a non-oscillating dynamic system with aperiodic nature of transients, providing the structure with the greatest stability margin.

4. CONCLUSION

The results suggest that the applying this improvement completely eliminates the significant shortcomings of the quality dynamics of a thrust bearing with self-compensation. It turns the design into a dynamic system with optimal dynamic characteristics - high stability indicators and oscillations index values which are specific for ideally damped dynamical systems.

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