Influence of technological factors on characteristics of hybrid fluid-film bearings

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Abstract. The influence of the parameters of micro- and macro unevenness on the characteristics of a hybrid bearing with slotted throttling is considered in the present paper. The quantitative assumptions of calculation of pressure distribution, load capacity, lubricant flow rate and power loss due to friction in a radial hybrid bearing with slotted throttling are taken into account, considering the shape, dimensions and roughness of the support surfaces inaccuracies. Numerical simulation of processes in the lubricating layer is based on the finite-difference solution of the Reynolds equation using an uneven orthogonal computational grid with adaptive condensation. The results of computational and physical experiments are presented.

1. Introduction
The use of fluid-film bearings in a number of cases is practically non-alternative. However, the selection, calculation and design is done by the designer for each case individually. The fluid-film bearing from the point of view of design is a multifactor complex model that allows to take into account the influence and interaction of design, technological and operational parameters. Accounting technology parameters associated with the study of the effect of macro and micro deviations on the characteristics of bearings [1], as the surface shape is formed during processing. The geometric parameters of the unevenness of the bearing surface depend on the combination of processing modes, rigidity and errors of the “machine-tool-detail” system. These errors can both improve the characteristics of the bearing, and adversely affect its performance. So, the study of the operation of a bearing with a different combination of parameters requires the formation of an individual approach, conceptual and mathematical models based on the joint numerical solution of the Reynolds equation and the radial gap function.

2. Mathematical model development
In some cases, the radial method of supplying lubricant to the fluid-film bearing is impossible or difficult for various reasons. If it is necessary to create a hybrid lift in the lubricating layer in a radial bearing, slit throttling from one side of the bearing can be used. The creation of this type of lifting force during the operation of hydrostatic dynamic bearing with slotted throttling is explained by the Lamakin-Etinger effect [2]. The bearing capacity of the lubricant layer is formed due to the pressure flows formed as a result of a specially created pressure drop at the sides and hydrodynamic effects when the spindle rotates. If the pressure at the sides of the developed bearing becomes equal to
atmospheric pressure, the bearing becomes hydrodynamic. Bearings with this method of creating of load capacity are used in machines for pumping liquids, lubrication is possible with a processed medium under pressure in the working area. For this bearing design, it is necessary to provide a device for discharging the used lubricating fluid from the opposite side inlet. To determine the type of mathematical dependencies describing the processes in the lubricating layer of a plain bearing, the design diagram of which is shown in the Figure 1, we introduce a number of assumptions [3]:

1) the lubricant is assumed to be a Newtonian fluid;
2) the flow of the lubricant under steady-state operation is assumed to be laminar, which will be provided by such a combination of operating parameters of the rotor-support system that
\[ Re \leq 41.3 \left( \frac{D_{\text{nom}}}{2h_0} \right)^{3.41} \]
3) the lubricant completely adheres to the support surfaces and during the operation of the system;
4) the lubricant is considered incompressible;
5) the bearing is completely filled with lubricant, in the loaded area the rupture of the lubricating film is not allowed;
6) there is no misalignment of shaft and bearing axes;
7) the motion along the longitudinal axis of the bearing is absent;
8) the isothermal flow of the lubricant in the gap is considered, as a result of which the viscosity of the lubricant is constant for the constant temperature.

![Figure 1](image)

Figure 1. Calculation diagram of a fluid-film bearing with slotted throttling.

On the basis of the assumptions above and the principle of operation of the fluid-film bearing, the determination of the pressure distribution in the lubricating layer is carried out by numerically solving the Reynolds equation in an isothermal formulation, which has the form:

\[
\frac{\partial}{\partial x} \left[ h^3 \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[ h^3 \frac{\partial p}{\partial z} \right] = 6 \mu \omega R \frac{\partial h}{\partial x}
\]  

(1)

To obtain the correct solution of (1), let us set the following boundary conditions, starting from the calculation scheme and the experience of numerical determination of pressures in the lubricating layer [2, 3]:

boundary conditions of the sides of the bearing:

\[ p(x, 0) = p_1; \quad p(x, L) = p_2 \]  

(2)

condition along the coordinate \( x = \alpha R \) shall be written based on the Sommerfeld hypothesis, according to which the bearing is covered by the lubricating layer. Then we can write the conjugation conditions:
\[ p(0, z) = p(2\pi R, z); \quad \frac{\partial p}{\partial x}(0, z) = \frac{\partial p}{\partial x}(2\pi R, z) \]  

(3)

The radial clearance between the journal and the bearing sleeve exerts a significant influence on the operation of the fluid-film bearing. When considering fluid-film bearings with rough surfaces, the function of the radial gap becomes more complicated. In the general case, all errors can be divided into dimensional errors, deviations in the shape of the surface, and roughness. These errors appear in the manufacturing process, their magnitude depends directly on the parameters of the process. To model the gap function, taking into account the real topology of the operating surfaces of the bearing assemblies, measurements were made and the surface of the experimental samples was studied.

An important issue in modeling the radial clearance function is the consideration of the effect of roughness, when its magnitude is commensurable with the thickness of the film. So, in [4-6] it was revealed that the maximum pressure and side leakage are reduced, and the coefficient of friction is increased. Further study of microroughness showed that for the longitudinal and transverse direction of the relief, the integral characteristics take different values. This observation became the basis for the development of the direction of creating bearings with textured surfaces [7]. The presence of a special texture allows to achieve optimum performance of the bearing unit.

To establish the mathematical relationships between the parameters of the roughness and its description, a measurement was made of the altitude parameter Ra and the pitch of microroughness, and the study of the portion of the supporting surface on a metallographic microscope. As a result, it was found that the basis of microroughness is the risks left as a result of the passage of the cutting tool over the surface. To describe microroughnesses, one can use the following function:

\[ h_{\text{micro}}(x, z) = A_1 \sin\left(\frac{2\pi}{k_1}z - sx\right) + A_2 \sin(k_2x) \]  

(4)

where

- \( A_1 \) - amplitude, which is equal to half the height of microroughness and is measured from a certain midline;
- \( k_1 \) - the coefficient that determines the step of the microroughnesses;
- \( s \) - the coefficient taking into account the displacement of the direction of microroughness, which corresponds to the feed of the tool during the processing of the supporting surface.

The shape of the microroughnesses described with the help of this function has the form shown in the Figure 2.

\[ Figure 2. \] Microroughness of a bearing’s surface.
The obtained mathematical description corresponds to a sufficient degree to the shape of the roughness of the bearing’s surfaces of the fluid-film bearing obtained as a result of machining and implements a combined approach to the description of surface’s microroughness.

To develop a mathematical model of the real surface, measurements were also made of the deviation of the profile of the support surface from roundness in different sections of the experimental samples, and also the circular log data were analyzed. The results of the analysis showed that the support surface in general is a superimposed macro- and micro-deviations. Moreover, the real surface profile can be obtained as a consequence of imposing several types of deviations. Conditionally this overlay can be represented as a superposition of surface irregularities, since unevenness reduces the radial clearance by an amount equal to the sum of the heights of the irregularities [8].

$$h(\varphi, z) = h_{\text{ideal}}(\varphi, z) + h_{\text{skewness}}(\varphi, z) + h_{\text{macro}}(\varphi, z) + h_{\text{micro}}(\varphi, z)$$  \hspace{1cm} (5)

From this equation, the component of the misalignment should be eliminated, since earlier in the system of assumptions it was assumed that the axes of the shaft and the bearing are parallel. For the case of an arbitrary eccentric shaft position in the bearing, during operation, the gap function is most conveniently represented as:

$$h(\varphi, z) = R(\varphi, z) - r(\varphi, z)$$  \hspace{1cm} (6)

where

- $R(\varphi, z)$ - bearing’s radius function;

- $r(\varphi, z)$ - rotor’s radius function.

To describe the macro deviations of the support surface of a radial hybrid bearing with slotted throttling, it is advisable to use the second-order surface equation, which is affected by some affine transformation operator, which establishes the corresponding rotation and displacement relative to the canonical coordinate system. Microroughness is imposed with respect to some given macroroughness [9]. Then the cross-section of the support surface will have the form shown in the Figure 3.

![Figure 3](image)

**Figure 3.** Determination of the gap function with assumptions of the microroughness of the surface: (a) the cross section in the coordinate system associated with the carrier surface; (b) studied cross-section in the coordinate system of the scan.

Let us perform numerical simulation of the processes taking place in the lubricating layer with a fluid-film bearing with slotted throttling on a finite-difference model on an uneven-dimensional orthogonal computational grid with adaptive condensation according to the procedure described in [10, 11]. The boundary conditions for finite-difference approximation take the following form:

$$p_{i,j} = p_{N,j}; \quad p_{i,j} = p_{i,N} = p_2$$  \hspace{1cm} (7)

A typical solution of the Reynolds equation for a slotted throttling bearings is shown in the Figure 4.
Figure 4. Pressure distribution of a bearing with slotted throttling.

For the tasks set, we consider the basic integral static characteristics of a fluid-film bearing: the bearing capacity, the flow rate a lubricant, the power loss due to friction. These characteristics are calculated through the following relationship [8]:

$$W = \sqrt{R_X^2 + R_Y^2}$$  \hspace{1cm} (8)

where $R_X, R_Y$ - hydrodynamic force in the projection on the $X$ and $Y$ axes, which are determined by numerical integration of the pressure distribution using the formulas [2]:

$$R_X = \int_0^{L_n D} p \cdot \sin(2x/D) \cdot dx \cdot dz; \quad R_Y = \int_0^{L_n D} p \cdot \cos(2x/D) \cdot dx \cdot dz$$  \hspace{1cm} (9)

$$Q_V = \int_0^{L_n D} - \frac{h^2}{12\mu K_x} \cdot \frac{\partial p}{\partial x} \cdot dx$$  \hspace{1cm} (10)

$$N_{fr} = \omega_o \frac{D}{2} \int_0^{L_n D} \left[ \frac{h}{2} \cdot \frac{\partial p}{\partial x} + \frac{U \cdot \mu \cdot K_x}{h} \right] \cdot dx \cdot dz$$  \hspace{1cm} (11)

A computer experiment was performed to investigate the characteristics of a fluid-film bearing with slotted throttling, taking into account the geometric parameters of the unevenness of the real surface. As geometric parameters of the experiment, the geometric parameters of the irregularities are taken: the height and step of the microroughness, the magnitude of the displacement relative to the longitudinal axis of the bearing, and the magnitude of macroroughness. For carrying out the computational experiment, the value of the macro-unevenness is taken as a percentage of the tolerance value for the bearing size. The main results of the computational experiment given in the Figure 5.
According to the findings of the research, inequalities of bearing area may not be considered if the ratio of the height of surface roughness to radial clearance does not exceed 0.015. For small and average radial clearance, a change in roughness by 1 μm leads to the change in bearing capacity 19-32%, flow rate - 7-13%, power loss on friction - 8-15%. An increase in roughness leads to an increase in lubricant consumption, lower bearing capacity and power loss on friction within 9%. An increase in the rate of micro roughness displacement in the longitudinal direction of the plain bearing leads to an increase bearing capacity within 5%, increase in flow rate by 7-8%. When taking into account the deviation of the shape when describing the surface profile, the bearing capacity decreases by up to 50% compared to perfect bearing. The affect of by the lateral shape and corsetry when using the real surface profile differs by 1.5 ... 2%, which is insignificant. If you look at the slider bearing with the real surface profile, you can see from the graph on Figure 5 that with an increase in the relative eccentricity, a degree of roughness appears, which is superimposed on macro-unevenness. It was found that the influence of roughness as part of a real profile at large relative eccentricities leads to the increase in bearing capacity and frictional power loss by 23-48%, depending on the parameters of surface roughness.

3. **Confirmation of the adequacy of the developed model**

To confirm the adequacy of the previously presented mathematical model for the operation of a bearing with slotted throttling and nonsmooth support surfaces, experimental studies were carried out. For this, a set of bearings of different lengths with different parameters of micro and macroroughness was developed. Each experimental sample was subjected to measurement of the parameters of

**Figure 5.** Integral characteristics of a hybrid fluid-film bearing with slotted throttling versus unevenness’s height.
irregularities according to the requirements of state standards with the use of a profilometer and a circular meter.

In describing the real surface profile, it was assumed that the support surface is the result of superposition of macro and micro deviations. To confirm this assumption, a study was made of the deviation of the shape of the support from roundness on the basis of the obtained circular logs. The results of the measurement of the cross section were obtained taking into account the altitude parameter of the roughness. To confirm the hypothesis put forward, a comparison is made between the experimental and theoretical curves (Figure 6) describing the profile of the cross section.

![Figure 6. Comparison of real and simulated curves.](image)

As it can be seen from the comparison of the presented results of modeling the real surface profile and measuring the deviation of the profile from roundness, the theoretical and experimental data agree satisfactorily. When specifying the parameters of the unevenness obtained in the course of measurements of the profile of the experimental sample in the developed mathematical model, the discrepancy between the experimental and theoretical profiles did not exceed 5-7%.

The results of measurements of microroughness of prototypes at 12 points, located through equal segments, with the help of a profilomer (Figure 7) showed that in most cases the shape of the distribution of irregularities on the reference length is close to a harmonic function over which a certain random variable is superimposed. In addition, the values of the height parameters and the pitch step of the irregularities assumed during the simulation were confirmed.

Thus, it can be concluded that the proposed assumptions in the mathematical description of the surface roughness were fully confirmed during the experimental study of the profile of the supporting surface. When a roughness measurement parameter was substituted for a mathematical model, the discrepancy between the theoretical and real profile was not more than 4%. As a result, it can be concluded that the mathematical description of the microroughnesses obtained to a sufficient degree corresponds to the shape of the roughness of the reference surfaces of the fluid-film bearing obtained as a result of machining.

The parameters of unevenness of the experimental samples of sliding bearings obtained as a result of measurements are recorded in the developed mathematical model in order to obtain numerical
values of the basic integral characteristics for each of the samples. A physical experiment was carried out to confirm the adequacy of numerical modeling of processes in the lubricating layer of a plain bearing with a slit throttling. For this purpose, the installation Figure 8), the methods of conducting and processing the results, which are described in detail in [12], were developed. The experimental module includes a rotary installation (with a dynamic loading system) with fluid-film bearings, one of which is a radial hydrostatic bearing with a radial lubricant supply, and the second support is a cylindrical bearing with axial throttling of the lubricant, a device for supplying a lubricant, in addition, the module has places for placing the primary converters of the information-measuring system.

Figure 7. Representation of roughness measurement results using the profilometer model 296.

Bearing nodes are installed in the body of the experimental installation from two sides. To ensure the accuracy of the assembly of the housing with the bearing assemblies and to ensure their alignment, as well as to measure the real radial clearance between the bearing and the shaft, a laser centering device Kvant-LM was used. Water is used as the lubricant in the test rig.
Figure 8. Photo of the test rig.

The basis of the information-measuring system is the multifunction module NI6052E, which features multichannel digital and analog input-output and counters-timers. As a tool for managing the test rig, acquiring and processing data from primary converters, the software developed by the authors in the LabView was used (Figure 9).

Based on the readings of the sensor system of the rig, the motion of the rotor was measured in the bearing during steady-state regime, as well as the angular velocity with a free run-out of the rotor. We process the results of measurements on the basis of the technique given in [8,12]. Figure 10 shows the results of a comparison of the calculated bearing capacity and the friction coefficient of the fluid-film bearing and the results of the experiment.
Figure 10. Comparison of theoretical and experimental studies of the load capacity of the lubricant layer in channels of complex geometry.

As can be seen from the graphs, the form of the theoretical curves corresponds to a sufficient degree to the experimental form. The discrepancy between the results of theoretical calculations and the physical experiment is within 12%. As a result, it can be concluded that the hypotheses, assumptions and methods of numerical modeling of processes in the lubricating layer proposed in the modeling process can be adopted. The developed mathematical model of the flow of lubricant in circular channels of complex geometry is considered adequate.

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