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STEAM TURBINE, GAS TURBINE, COMBINED-CYCLE PLANTS, AND THEIR AUXILIARY EQUIPMENT

Calculation and Interpolation of the Characteristics of the Hydrodynamic Journal Bearings in the Domain of Possible Movements of the Rotor Journals

A. I. Kumenko*, V. N. Kostyukov, and N. Yu. Kuz'minykh

DYNAMICS Scientific-and-Production Center, Omsk, 644007 Russia

**e-mail: 1949kai@mail.ru*

Abstract—To visualize the physical processes that occur in the journal bearings of the shafting of power generating turbosets, a technique for preliminary calculation of a set of characteristics of the journal bearings in the domain of possible movements (DPM) of the rotor journals is proposed. The technique is based on interpolation of the oil film characteristics and is designed for use in real-time diagnostic system COMPACS®. According to this technique, for each journal bearing, the domain of possible movement of the shaft journal is computed, then triangulation of the area is performed, and the corresponding mesh is constructed. At each node of the mesh, all characteristics of the journal bearing required by the diagnostic system are calculated. Via shaft-position sensors, the system measures—in the online mode—the instantaneous location of the shaft journal in the bearing and determines the averaged static position of the journals (the pivoting vector). Afterwards, continuous interpolation in the triangulation domain is performed, which allows the real-time calculation of the static and dynamic forces that act on the rotor journal, the flow rate and the temperature of the lubricant, and power friction losses. Use of the proposed method on a running turboset enables diagnosing the technical condition of the shafting support system and promptly identifying the defects that determine the vibrational state and the overall reliability of the turboset. The authors report a number of examples of constructing the DPM and computing the basic static characteristics for elliptical journal bearings typical of large-scale power turbosets. To illustrate the interpolation method, the traditional approach to calculation of bearing properties is applied. This approach is based on a Reynolds two-dimensional isothermal equation that accounts for the mobility of the boundary of the oil film continuity.

Keywords: journal bearing, power-generating turboset, Reynolds equation, triangulation, interpolation, technical diagnostics, real-time mode

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Design, mounting, and repair of large-scale power-generating turbosets impose strict requirements to assembling of the rotors and selection of clearances in the gaskets and the bearings. A correct selection of the clearances and maintenance of their sizes ensure high reliability of the turboset during its operation.

The basic mechanical and hydrodynamic processes that occur in the bearings and determine the vibration safety of the shafting during the lifetime of large-scale turbosets have been checked in an indirect way, until recently, on the basis of the absolute vibration of the supports. Introduction of sensors that record the relative static and dynamic positions of the rotor journals according to [1] will allow monitoring the physical processes considerably better in the future. However, the principles of the method for taking advantage of all capabilities of such sensors have not been developed yet, in particular, for diagnostic purposes.

To effectively resolve this topical and complicated problem, new methods should be developed that combine a detailed mathematical modeling and modern means of information acquisition and analysis of the operating characteristics. This article presents one of the methods developed for the COMPACS® diagnostic system [2].

Application of modern technical diagnostics means enables detection at early stages of major defects related to wear, irreversible strains, and misalignments of the supports as well as detection of assembling faults that occur in the course of repairs. Early diagnostics prevents accidents, allows reduction in the downtime, and, consequently, a considerable decrease in the operating costs.

Key components of the turboset shafting are bearings. In power engineering, in large-scale turbosets, hydrodynamic bearings are traditionally used, both journal and thrust bearings. For a number of reasons, electromagnetic bearings have not found wide appli-

cation in large turbosets yet. Most frequently, hydrodynamic journal bearings of two types are used, viz., bearings with fixed pads and segment with tilting pads. There is a great variety of the designs of the journal bearings of both types [3–6]; each of them has its advantages and disadvantages and its own field of application.

In this work, we propose a method for preliminary calculation of the static characteristics of hydrodynamic journal bearings to be applied to a real-time monitoring system.

Various requirements are imposed on the turboset journal bearings. They have to possess

(i) Required load-bearing capacity, i.e., a capacity of maintaining the shaft in the preset position;

(ii) Required dynamic stiffness and damping, i.e., an ability not to allow too great vibration amplitudes upon disturbances of different nature in a wide frequency range;

(iii) Significantly expressed stiffness anisotropy; and

(iv) Minimum power losses due to friction.

In addition, the journal bearings have to ensure:

(i) Hydrodynamic lubrication mode and the minimum heating of the babbit metal and

(ii) Sufficient stability threshold margin of the rotor system in terms of the rotation frequency.

No rubbing[1] is tolerated in the bearings under vibration of the rotors at critical frequencies and inevitable misalignments, or mutual displacements, of the supports.

The diagnostic system has to control the clearances in the bearings and promptly signal imminent emergency situations. The following parameters are to be controlled:

(i) Minimum clearances, shaft misalignments in the bearings, and the probability of rubbing the babbit metal;

(ii) Oil film temperature at the bearing inlet and the oil discharge;

(iii) Babbit temperature; and

(iv) Deviations of the current rotor journal positions from the calculated and experimental parameters obtained under nominal loads on the supports.

The aim of this work is to develop an efficient method for application of the pre-calculated bearing characteristics in an operating real-time diagnostic system. The application of the method is illustrated by examples of the calculated basic static characteristics of the elliptic journal bearing, a typical bearing used in domestic turbines. The developed method can be applied, however, to calculation of dynamic properties of the bearings, such as the coefficients of rigidity and damping. These coefficients are used in diagnostics of the technical condition of the bearing system and to solve other problems that are beyond the scope of this work.

Storage and the further use of the calculated data are necessary for the following reason. It is sufficiently difficult to solve the complete equation system for the journal bearing; this requires considerable computational resources. Such time-consuming operations are unacceptable when the static and dynamic properties of the bearing are calculated in a real-time technical diagnostic system, e.g., the COMPACS® system.

The most effective solution to this problem is the online use of already precalculated and verified results. For this purpose, it is intended to interpolate the characteristics of the bearings for the domains of possible movements of the shaft journal center.

When developing the interpolation method, the traditional method for calculation of the bearing properties was applied. A similar method for calculation of the oil film is set forth in xUSSR handbooks on bearing design [3, 4] as well as in foreign handbooks [7]. In the domestic practice, the bearings are still calculated according to the tables presented in this literature. These tables, however, provide the characteristics of the oil film only for a strictly vertical load on the bearing.

Deviations from the calculated static load on the bearing determined by the diagnostic system can be caused by mutual displacements of the supports and the influence of the geometry and the operating variables of the neighboring bearings. These deviations result in distorted design line of shafting (see the dashed line in Fig. 1a). Such changes in the loads are an important diagnostic sign for evaluation of the technical condition of the shafting. One should bear in mind that the line of the shafting may deflect from the ideal shaft bending line in both horizontal and vertical directions.

The mutual displacements of the supports and disturbances in the support reactions are accounted for by the following factors:

(i) Spread in the clearance sizes within the tolerance ranges upon assembling the bearings;

(ii) Wear and deformation of the babbit metal;

(iii) Overheating and transition of the stator components from the cold state into the hot one;

(iv) Uneven foundation settlement; and

(v) Seasonal temperature fluctuations of the ground, the supporting plate, and the foundation stator.

One of two adjacent bearings may appear to be unloaded and the other overloaded, which is often accompanied by horizontal loadings in the bearings.

Figure 1 schematically shows a shafting with several turboset stages and one of the variants of mounting a pair of the shaft-position sensors. The sensors are

[1] Rubbing in a bearing is a direct contact between the shaft journal and the babbit metal. This is an extremely undesirable phenomenon that results under certain circumstances in considerable vibrations, overheating, and rapid wear. Under normal operating conditions of the bearing, the journal and babbit are separated by a lubricant layer in which a lifting force is induced during rotation of the shaft.

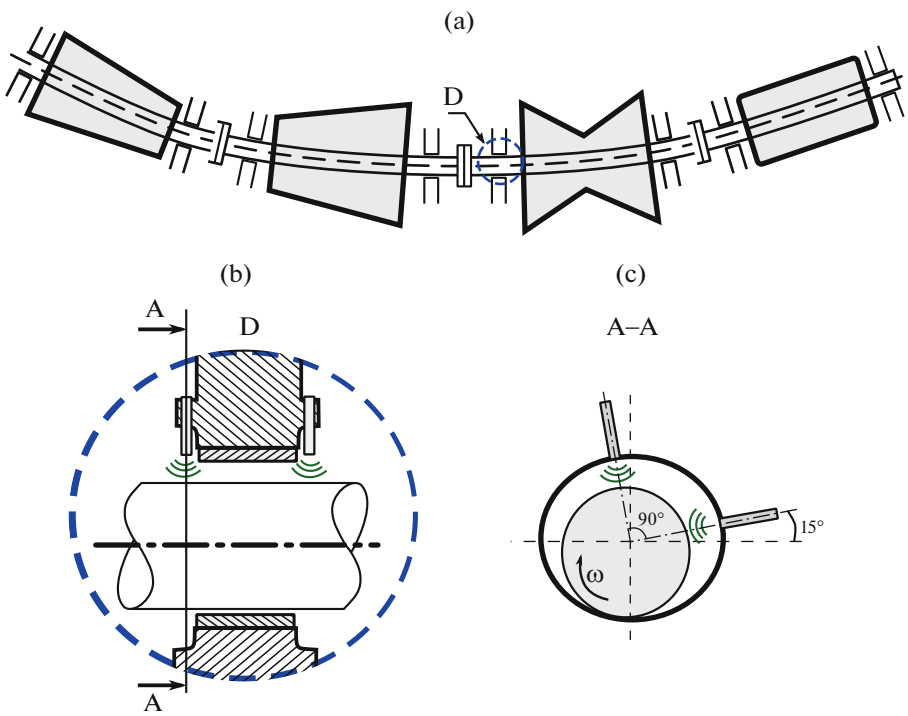

Fig. 1. Schematics of (a) the shafting, (b) an elliptical bearing, and (c) the shaft- position sensors.

positioned at an angle of 90° to each other and measure simultaneously horizontal and vertical movements of the shaft journal.

According to [1], the pair of the shaft-position sensors is installed at one end of the bearing; however, to perform comprehensive diagnostics of the technical condition of the shafting and control the dynamic and static misalignments in the rotor journals, installation of a pair of sensors at both sides of each bearing is recommended for turbosets with a power of 500 MW and above.

Localization of the shaft journal in the clearance is necessary to calculate the static and dynamic parameters of the oil film in the bearing. Using the values of the above parameters, the diagnostic system calculates the loads on the shaft, the minimum clearances, the power losses, the oil flow rate, and the maximum oil temperature, which is always noticeably higher than the temperature at the oil discharge controlled separately.

Comparison of these parameters for all bearings enables online evaluation of the condition of the turboset and identification of the sources of increased vibrations. Analysis of the obtained data allows calculation of the optimal correction movements of the supports and selection of the optimal clearances in the bearings and the required oil flow rate for each of the bearings.

The proposed method enables preliminary calculation of the characteristics under any possible loading directions. It should be noted that the interpolation method does not depend on a particular algorithm for calculating the properties of the journal bearings. In the real-time diagnostic system, more complicated and more accurate physical models of the journal bearings can be applied if necessary.

Below, the method for interpolation in the domains of possible movements is set forth. The application of the method is illustrated by the results of calculation of the static characteristics for three bearings of a K-1000-5.9/1500 turbine different in their parameters, viz., cylindrical bearing no. 12, elliptical bearing no. 10, and elliptical bearing no. 11 with an circumferential groove in the upper pad. The results are presented in the form of the level lines in the entire domain of possible movements constructed in dimensionless coordinates.

METHOD FOR INTERPOLATION OF THE BEARING CHARACTERISTICS IN THE DOMAINS OF POSSIBLE MOVEMENTS

The task is to determine all necessary static and dynamic characteristics for each particular bearing by the position of the shaft journal in the clearance. To simplify the calculation, the shaft axis is assumed to be parallel to the bearing axis, i.e., there is no misalignment in the bearing. It is convenient to use in calculations and the representation dimensionless variables defined below.

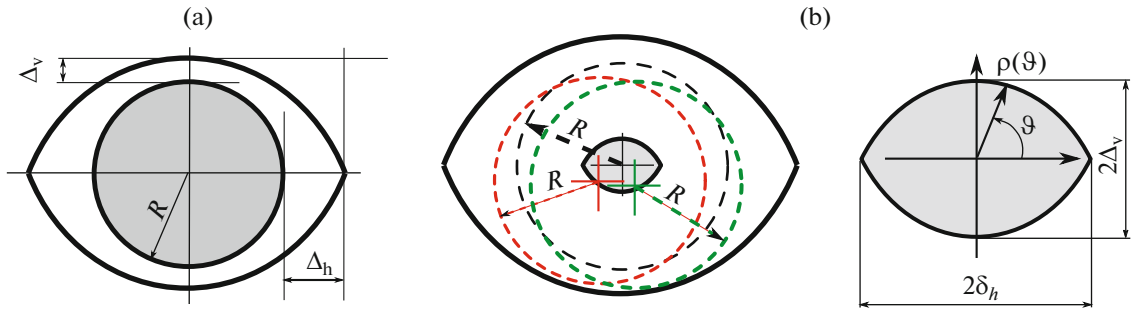

Fig. 2. (a) Horizontal and vertical clearances of the elliptical bearing and (b) construction of the DPM of the shaft journal center.

Under the laminar isothermal flow of the lubricant, the problem of calculation of the lubricant layer characteristics is considerably simplified. The solution can be written as a function of several dimensionless parameters, since physical similarity is observed. The dimensionless form of representation of the characteristics is widely used in engineering calculations [3, 4, 7]. There are three dimensionless parameters for an even, without grooves and hydrolift pockets, elliptical bearing, namely:

(i) Bearing length to diameter ratio L/D;

(ii) Pad wrap angle α; and

(iii) Ellipticity of the bearing m.

The domain of possible movements (DPM) can also be represented as a function of dimensionless parameters. The construction of the DPM for an elliptical bearing is set forth below.

The dimensionless form of representation of the characteristics allows for the once calculated properties to be applied to objects that have a similar set of dimensionless parameters. For example, bearings 10 and 11 of the K-1000-5.9/1500 turbine have different diameters but identical values of m and α and the ratios L/D are very close for them, 0.69 and 0.73, respectively. For most diagnostic problems, this difference is not significant and the state diagram can be calculated only one time for these bearings. In the calculations presented as the examples, the ratio $L/D = 0.7$ is used.

Unfortunately, consideration of the real temperature distribution and the lubricant flow turbulence breaks the similarity. However, under moderate loads on the shaft, the effect of these factors is considered in a relatively simple way by the correction coefficients and the iterative search for the lubricant temperature [3, 4]. In the examples given below, these factors were not taken into consideration.

In Fig. 1c, the cross section of the shafting is shown at the location of the sensors. With this view direction, from the regulator to the generator (the left-sight view), the shaft rotates clockwise. Such a projection is traditionally used in the domestic literature on the bearings; in the English literature, the opposite direction is accepted (the right-sight view) [7, 8]. The shaft rotates at a cyclic frequency (in terms of rad/s) $\omega = 2\pi n/60$ where n is the rotation frequency in terms of min^{-1}.

The stress-loading vector $\vec{\zeta}$ is defined in the following way [3, 4]:

$$\vec{\zeta} = \frac{\vec{q}\Psi^2}{\mu\omega},$$

where $\vec{q} = \dfrac{\vec{Q}}{LD}$ is the specific load vector (Pa); $\vec{Q}$ is the load vector on the support (N); μ is the averaged lubricant density (Pa s); $\Psi = \Delta_h/R$ is the relative oil film thickness (a small dimensionless parameter); R is the shaft journal radius; and Δ_h is the horizontal clearance.

The components of the dimensionless stress-loading vector $\vec{\zeta} = (\zeta_x, \zeta_y)$ are defined as projections on the coordinate axes. The stress loading ζ is directly proportional to the load and inversely proportional to the rotation frequency. In the English literature, dimensionless quantities are also used; however, instead of the stress loading ζ the Sommerfeld number S is used that is inversely proportional to ζ [7].

DOMAIN OF POSSIBLE MOVEMENTS OF THE SHAFT JOURNAL IN THE BEARING BORES

Below, we consider the construction of the DPM of the journal by an example of an elliptical bearing. The image of such a domain for bearings of two types is given in [3]; the algorithm for construction of the DPM is not presented there, however. The working surface of the pads of an elliptical bearing is formed by two cylindrical segments (see Fig. 2a). The radius of the pad slightly exceeds the journal radius. The bearing is assembled with the preset clearance within the tolerance range, and the clearance is filled with a fluid lubricant. For an elliptical bearing, the determining parameters are the values of the horizontal (Δ_h) and vertical (Δ_v) clearances (see Fig. 2a). These clearances constrain the movements of the shaft journal. The boundary of the domain of all possible movements of the journal center at which it touches the bore surface

is the boundary of the DPM. In Fig. 2b, the DPM is marked in gray; it is confined by two elliptical arcs. The half-height of the DPM from the bore center equals Δ_v and the half-width of the domain equals $\delta_h = \Delta_h\sqrt{1-m^2}$, where $m = 1 - \Delta_v/\Delta_h$, is the ellipticity.

In the polar coordinates, the location of the boundary of the domain of possible movements (see Fig. 2b, on the right) is defined as

$$\rho(\vartheta) = \Delta_h\left[\sqrt{1-(m\cos\vartheta)^2} - m|\sin\vartheta|\right].$$

For example, the journal diameter of bearing no. 10 of the K-1000-5.9/1500 turbine equals 800 mm, the nominal ("design") horizontal and vertical clearances equal 0.775 and 0.420 mm, respectively, and the ellipticity is 0.45. The sizes of the DPM are 0.690 mm in the horizontal direction (half-width) and 0.420 mm in the vertical direction (half-height).

In Fig. 3 and below, the DPM is constructed in the dimensionless coordinates χ_x and χ_y: $\chi_x = \frac{\varepsilon_x}{\Delta_h}$, $\chi_y = \frac{\varepsilon_y}{\Delta_h}$, where ε_x and ε_y are the displacements of the journal center with respect to the bore center in the horizontal and vertical directions. The boundary of the DPM for $m = 0.45$ is shown by the dotted line. For simplicity, a mesh with 377 nodes is shown in the figure; in practice, the computations are performed using a finer mesh with 1500–3000 nodes.

In a similar way, the DPM is constructed for bearings of other types, including the bearings with movable segments (this task is beyond the scope of this work).

CONSTRUCTION OF THE MESH AND TRIANGULATION OF THE DOMAIN OF POSSIBLE MOVEMENTS

The boundary of the DPM corresponds to such a movement of the shaft at which the latter's journal touches the babbit metal with the minimum clearance being equal to zero. The hydrodynamic calculation is meaningless for such a case; therefore, to perform the interpolation, the mesh is constructed near the boundary but does not touch it (see Fig. 3). In the examples given below, the distance from the edge mesh points to the boundary of the DPM equals 0.02 in terms of dimensionless units. In the examples, the Delaunay triangulation is applied to a two-dimensional domain and the linear interpolation on the triangles [9]. The qhull program library was used [10]. The interpolation function is continuous over the entire area inside the mesh.

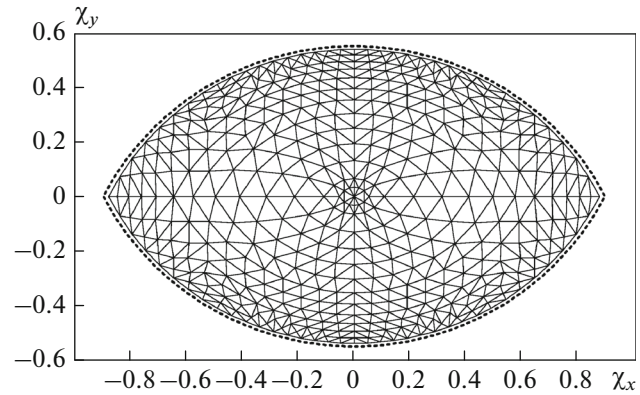

Fig. 3. Domain of possible movements of the shaft journal for the ellipticity $m = 0.45$ with the constructed triangular mesh with 377 nodes.

PRECALCULATION OF THE CHARACTERISTICS AND INTERPOLATION BY PRECALCULATED PARAMETERS

The problem is solved in the following sequence:

(1) For every bearing, the DPM and the mesh are constructed taking into account the really measured clearances and the structural features.

(2) For every mesh node, all static and dynamic parameters are computed that are required for functioning of the turboset diagnostic system. The computed results are stored in the database.

(3) In the real-time mode, the diagnostic system downloads the records from the database into the RAM.

(4) Via a set of shaft-position sensors, the real displacements of the shaft with respect to the position of rest are determined and the individual instantaneous positions of every bearing in the DPM are computed.

(5) The diagnostic system calls the subroutine for interpolation of the bearing properties; the value and the direction of the force acting on the shaft journal are computed; and the oil flow rate, the oil temperature, the power friction losses, and the rigidity and damping coefficient matrices are determined.

(6) The estimated values and directions of the loads in the bearings and the latter's dynamic parameters are used to control the condition of the shafting in a higher-level mathematical model (The description of such a model is beyond the scope of our work).

CALCULATED RESULTS

In this section, the calculated results are presented that illustrate the use of continuous interpolation of by the characteristics precalculated in the nodes of the DPM mesh. The calculations were performed for three elliptical bearings used in low-speed 1000-MW

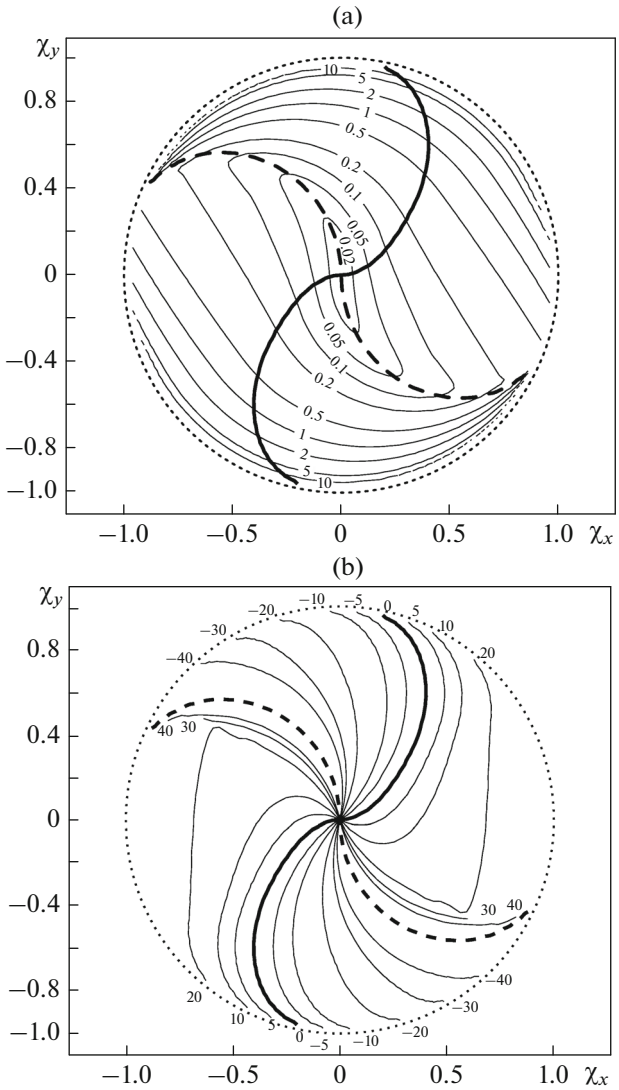

Fig. 4. (a) Stress loading ζ and (b) angles of deflection of the load vector from the vertical line (deg) calculated in the entire DPM for a cylindrical bearing: $m = 0$; $L/D = 0.7$; and $\alpha = 140°$.

turbosets for NPPs. The parameters of the bearings are provided in the table. The generator bearing has the zero ellipticity.

In Fig. 4, the calculated results are shown for a bearing with the cylindrical bore (variant 1 in the table). The fine lines correspond to the lines of equal stress loading in Fig. 4a and to the lines of equal angles of deflection of the load from the vertical line [$\theta = \arctan(\zeta_y/\zeta_x)$] in Fig. 4b.

Figure 5 shows similar static characteristics of an even elliptical bearing (variant 2), and Fig. 6 shows that of an elliptical bearing with a circumferential groove in the upper pad (variant 3).

The solid bold line in Figs. 4–6 corresponds to the strictly vertical load (the normal load line) and the dashed bold line corresponds to the strictly horizontal load. The points of intersection of the strictly vertical and the strictly horizontal loads correspond to the zero load on the shaft journal. For the symmetrical bearings (variants 1 and 2), this point is positioned in the center of symmetry. For an asymmetric bearing (variant 3) that has a groove in the circumferential direction in the upper pad center, the zero load point is displaced to the left towards the upper pad.

Below, an example of practical application of the interpolation method is provided.

When elliptical bearings are used, the degree of ellipticity is significant: the greater the ellipticity, the more rigid the bearing is in the vertical direction. The horizontal rigidity does not greatly depend on the ellipticity.

The effect of the ellipticity m on the power loss factor ξ is shown in Fig. 7. To construct this diagram, the DPM and the power loss factors were determined beforehand for 16 values of m.

In this case, the power losses in the bearing are found by the formula [3]

$$N_{\text{frict}} = \frac{LD^2\omega^2}{2\Psi}\xi.$$

The dimensionless power loss factor ξ depends on the dimensionless geometric parameters L/D, α, and m, the dimensionless stress loading vector $\vec{\zeta} = (\zeta_x, \zeta_y)$, and the sizes of the real clearance, as well as on the lubricant flow rate and the lubrication rate distribution.

The stress loading ζ_y is preset for the strictly vertical load ($\zeta_x = 0$). For example, for variant 2 (see the table), the stress loading $\zeta_y = 2.3$ at a load on the bearing of 860 kN and the mean viscosity $\mu = 0.02$ Pa s.

Parameters of the bearings used for calculations

Variant	Diameter D, mm	Length L, mm	L/D	Circumferential groove in the upper pad, mm		Pad wrap angle α	Ellipticity m	Horizontal clearance Δ_h, mm	Vertical clearance Δ_v, mm	Nominal load Q, kN
				depth h	width B					
1	750	550	0.733	0	0	140	0.00	0.750	0.750	600
2	800	550	0.688	0	0	140	0.45	0.775	0.423	860
3	750	550	0.733	12	150	140	0.45	0.775	0.425	600

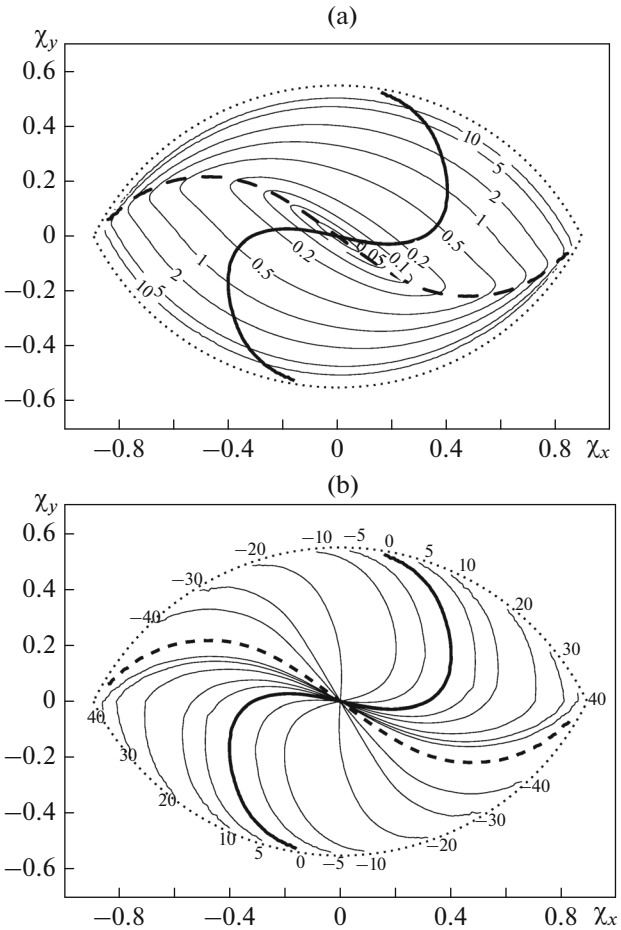

Fig. 5. (a) Stress loading ζ and (b) angles of deflection of the load vector from the vertical line (deg) calculated in the entire DPM for an elliptical bearing: $m = 0.45$; $L/D = 0.7$; and $\alpha = 140°$.

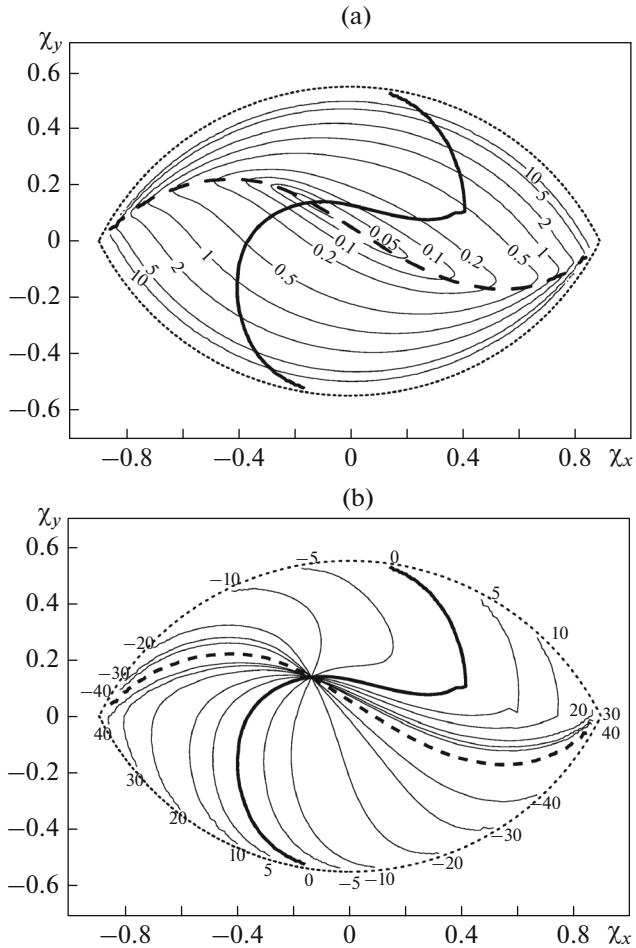

Fig. 6. (a) Stress loading ζ and (b) angles of deflection of the load vector from the vertical line (deg) calculated in the entire DPM for an elliptical bearing with a circumferential groove in the upper pad center: $m = 0.45$; $L/D = 0.7$; $\alpha = 140°$; and a groove width of 20% of the bearing length L.

With the fixed horizontal clearance and the varying vertical clearance, the ellipticity varies. At low values of m (see Fig. 7), the loss factor slowly decreases with the increasing m for different values of the stress loading ζ_y. With the further increase in the m, the loss factor begins to grow rapidly.

The ellipticity degrees in the range 0.4–0.6 are optimal for minimization of the losses. These values of m have been verified by practice and correspond to the maximum stability margin of the rotors [6].

In addition to calculating the stress loading and the power factor, static parameters, such as the minimum clearance, the oil flow rate, and the oil temperature as well as dynamic parameters, viz., the rigidity and damping coefficient matrices, have been calculated [3, 4, 7]. Owing to a limited length of the paper, the results of calculating these characteristics are not presented here.

The interpolation method is applicable to any bearing parameters precalculated at the mesh nodes and stored in the database.

CONCLUSIONS

(1) The proposed technique for preliminary calculation can be applied to determine any characteristics of the radial journal bearings in the domain of possible movements. The technique does not depend on the particular mathematical model of the bearing.

(2) Comparison of the set of measured and calculated parameters enables online evaluation of the condition of the turboset and identification of the sources of increased vibrations. Analysis of the data allows for the optimal correction displacements of the bearings to be calculated and the optimal clearances in the bearings and the required lubricant flow rates to be selected.

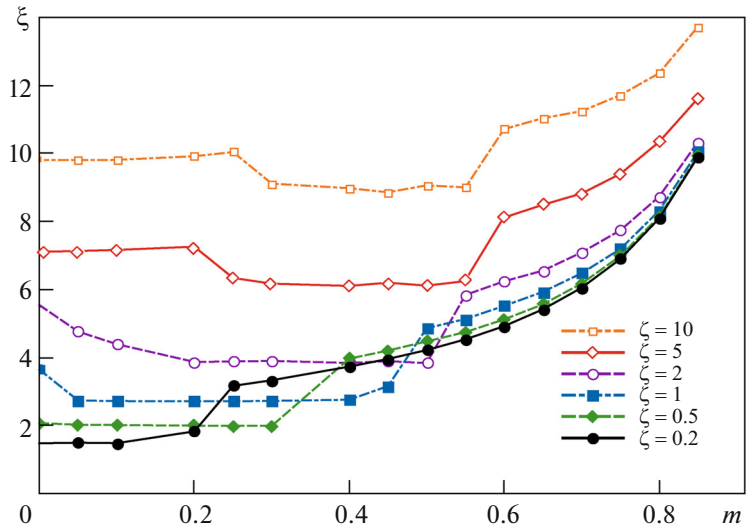

Fig. 7. Dependence of the loss factor ξ on the ellipticity *m* under the fixed stress loading ζ.

(3) The technique is intended for systems for technical diagnostics of physical processes that occur in the support systems of turbosets. It is also an integral part a mathematical model for calculation of the static and dynamic loads on the shafting components and their support system, including calculation of the natural frequency spectra, forced vibrations, and the stability thresholds of a shafting that rests on hydrodynamic journal bearings.

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