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Dynamic characteristics of the multilobe journal bearings with the lobes of different geometry

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Abstract Multilobe journal bearings are applied in different types of rotating machines such as, e.g. turbounits. The design of such bearings assures proper load capacity, thermal conditions of oil film and stability of the operation of responsible turbounits. However, the design of these bearings can be changed by application of lobes with different geometries. It allows obtaining new family of the dynamic characteristics. The paper presents the multilobe journal bearings with 2- and 3-lobes characterized by different geometry of lobes. The dynamic characteristics in form of the stiffness and damping coefficients of oil film were obtained by perturbation method. Stability ranges of simple symmetric rotor were determined for considered bearings. The iterative solution of Reynolds, energy and viscosity equations allows the obtaining the oil film forces, which were the basis of the bearing dynamic characteristics. Adiabatic, laminar oil film and the static equilibrium position of journal were assumed.

Keywords: multilobe journal bearing, different geometry of lobes, dynamic characteristic.

1. Introduction

The demands of power industry are for very durable and reliable turbounits, which fulfil simultaneously the requirements of shortest time of overhaul and maintenance [1-5] to assure their maximum disposability. The development of turbounits and requirements in achieving higher efficiency and the reduction of costs yields to higher rotor weights and consequently to enlarged specific bearing loads. Finally, the bearings can become the limiting factor for the turbine design and for the efficiency of the turbo generator set. Current status and aim of development of turbounits journal bearings are presented in Fig. 1 [3].

In large multistage turbo generators the journal bearings have the basic effect on the reliability and durability of these machines [3, 4]. The rotors of these units operate mainly in 2- and 3-lobe journal bearings, which assure the operation of turbounits at the assumed temperature and minimum friction loss. The application of these bearings causes the operation at the correct vibration frequency of shafts line and the largest resistance against the accidental external loads that generate the unstable behaviour of the rotor.

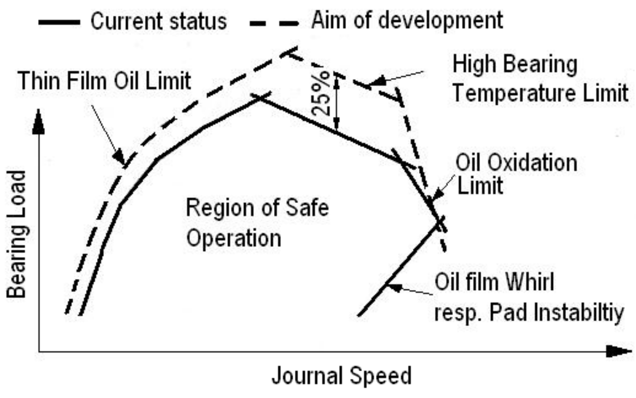

Figure 1. Current status and aim of development of turbounits journal bearings.

The investigation into new types of 2-lobe journal bearings of turbo generator [3] points out on the increase in the bearing load capacity. Such bearing should meet the rotor dynamic criteria, be interchangeable with existing bearings and the friction losses as well as the lube oil supply requirements should be maintained. The paper [3] provides information on theoretical and experimental studies aimed at obtaining optimal working conditions of 2-lobe turbo generator bearings by introducing design changes. There are no data on the design of the tested bearings, however it was found that the introduced changes reduced the resistance to motion by 25%.

For ensuring the reliable operation of the bearing and rotor-bearing system it is important to know its static and dynamic characteristics which are also affected by the bore profile of bearing sleeve. An example of the effect of different bore profile of 2- and 3 lobe bearings on the power loss is shown in Fig. 2 [4, 6].

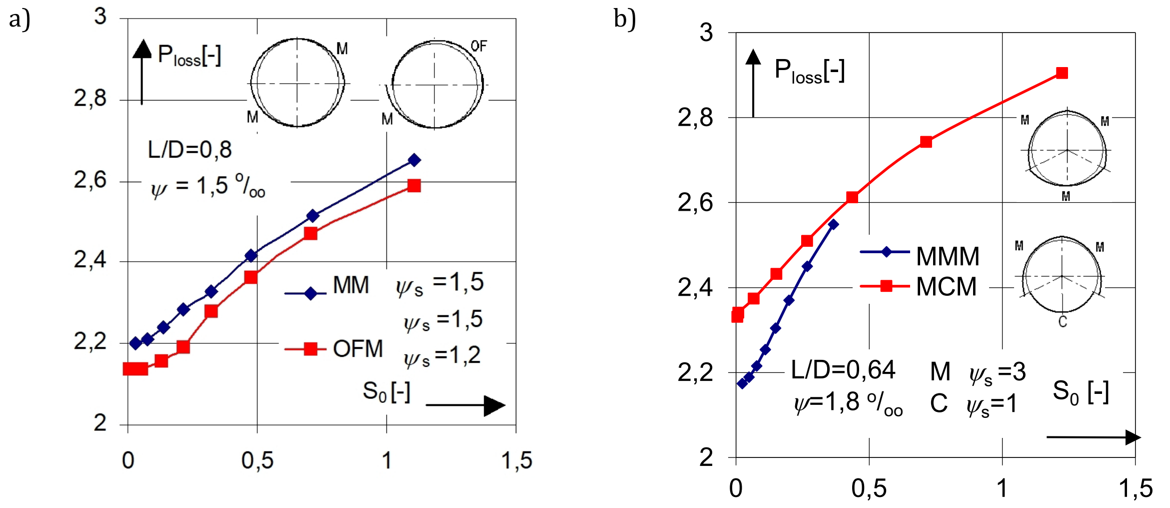

Figure 2. Power loss of two types of two types of 2- and 3-lobe journal bearings versus Sommerfeld number.

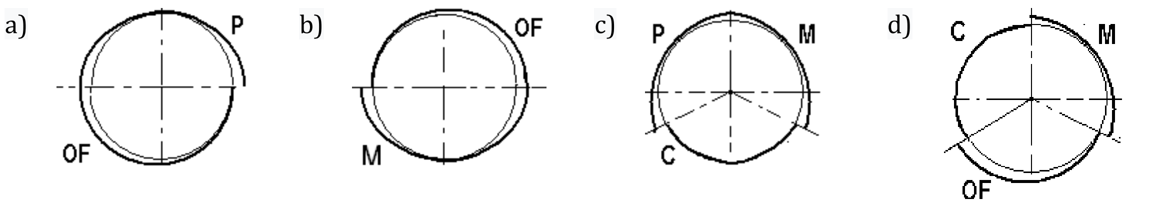

Figure 3. Multilobe journal bearings with the lobes of different geometries: a) 2-lobe (P OF), b) 2-lobe (OF M), c) 3-lobe (M P C), d) 3-lobe (M C OF); Symbols: C – cylindrical, M – classic (discontinuous), P – pericycloid (continuous), OF – offset.

New dynamic properties of multilobe journal bearings can be obtained by application in one bearing [4, 7, 8] the lobes with a circular, classic multilobe, pericycloid profile. The design of such a bearing with the lobes of different geometry (Fig. 3) may provide e.g. different temperature conditions of the lubricating film, a change (reduction) of the resistance to motion (friction losses) and new dynamic characteristics.

In journal bearings the knowledge of the static characteristics allows determining the dynamic characteristics, which are expressed by the stiffness, and damping coefficients of the lubricating film and stability ranges of assumed rotor [1, 9, 10].

This paper presents the dynamic characteristics of multilobe journal bearings with the lobes of different geometry. The equations of Reynolds, energy and viscosity were solved numerically using finite differences method. It was assumed laminar, adiabatic oil film, and parallel orientation of the axes of journal and sleeve and the conditions of the static equilibrium positions of journal. The dynamic characteristics in form of stiffness and damping coefficients of oil film were obtained by perturbation method. Stability ranges of simple symmetric rotor operating in considered journal bearings were determined, too.

2. Geometry of the oil film of comparable, multilobe journal bearings with the lobes of different geometry

Typical multilobe (classic) journal bearing is composed of single circular sections whose centres of curvature are not in the geometric centre of the bearing (Fig. 4). The geometric configuration of the bearing as a whole is discontinuous and not circular. The multilobe pericycloid journal bearings ("wave bearings" [4] is characterised by continuous profile and multihydrodynamic oil films.

On the assumption of the parallel axis of journal and bearing sleeve, the geometry of oil film gap of multilobe journal bearing (Fig. 1) is described by

$$\bar{H}(\varphi) = \bar{H}_{Li}(\varphi) - \varepsilon\cos(\varphi - \alpha), \tag{1}$$

where: $\bar{H} = h/(R - r)$ – dimensionless oil film thickness, h – oil film thickness (m), R, r – sleeve and journal radius (m), α – attitude angle (°), ε – relative eccentricity, φ – peripheral coordinate (°). The first term of this equation gives the geometry of multilobe bearing [4, 9].

Multilobe and pericycloid geometries are described by the equations:

$$\bar{H}_{Li}(\varphi) = \psi_{si} + (\psi_{si} - 1)\cos(\varphi - \gamma_i), \tag{2}$$

$$\bar{H}_P(\varphi) = \lambda^*[1 + \cos(n_p\varphi)], \tag{3}$$

where: γ_i – angle of lobe centre point (°), ψ_{si} – lobe relative clearance, λ^* – pericycloid relative eccentricity, n_p – multiply of pericycloid.

Cylindrical and multilobe profiles in journal bearings can be combined to give new designs that allow the increase in e.g. the rotational speeds of rotor [6, 11, 12]. Considering the basic four types of bearing profiles, i.e. cylindrical C, multilobe M, pericycloid P and offset OF it is possible to obtain 16 and 64 variations of bearings for 2- or 3-lobe, respectively [4]. Figure 5 shows the oil film thickness of bearing with the lobes of different geometry: 1 – multilobe, 2 – cylindrical, 3 – offset.

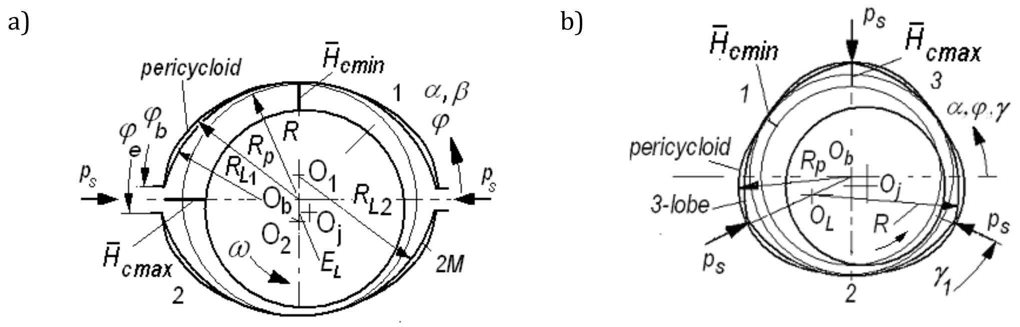

Figure 4. Geometry of 2- and 3-lobe journal bearings: a) 2P and 2M, R_{L1}, R_{L2} – radius of lobe 1 and 2, respectively, b) 3P and 3M; O_b, $O_{1,2}$, O_j – centre of sleeve, lobe and journal, R_p – radius of pericycloid; 1, 2, 3 – number of lobe. ; $\bar{H}_{c\,\min}$, $\bar{H}_{c\,\max}$ – minimum and maximum oil film thickness in central position of journal.

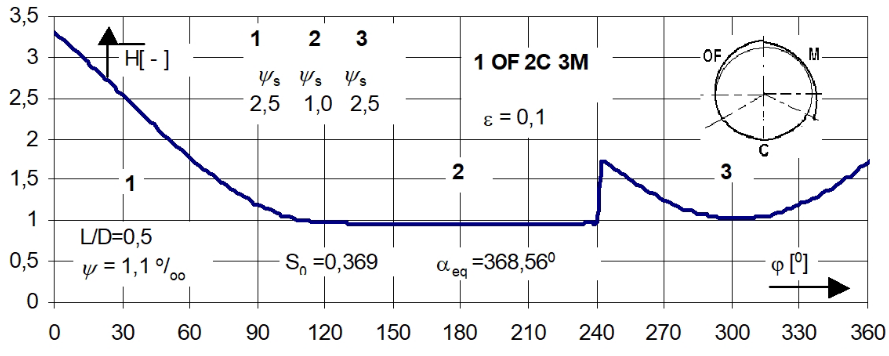

Figure 5. Oil film thickness of journal bearing with the lobes of different geometry: 1 – offset, 2 – cylindrical, 3 – multilobe.

3. Stiffness and damping coefficients of oil film

The equations of motion for the journal and the centre of elastic shaft [1, 10, 12] are given in the matrix form by

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{B}\dot{\mathbf{x}} + \mathbf{C}\mathbf{x} = \hat{\mathbf{a}}\cos(\omega t) + \hat{\mathbf{b}}\sin(\omega t), \tag{4}$$

where: **M**, **B**, **C** – matrices of mass, damping and stiffness, $\hat{\mathbf{a}}$, $\hat{\mathbf{b}}$ – coefficients of dynamic constraints.

After transformations of Eq. (4) the real and imaginary part was obtained [10]. The stability of elastic rotor-bearing system is analysed based on the following characteristic frequency equation of 6th order with regard to (λ/ω) [12, 13]:

$$c_6\lambda^6 + c_5\lambda^5 + c_4\lambda^4 + c_3\lambda^3 + c_2\lambda^2 + c_1\lambda + c_0 = 0. \tag{5}$$

The assumption of the solution of Eq. (5) is $\lambda_j = -u_j + \mathrm{i}v_j$ $(1 \le j \le 6)$, with u as damping and v representing the self-vibrations. Stability of the linear vibrations of the system occurs only when all real parts of eigenvalues λ_j are negative. The coefficients c_0 through c_6 in Eq. (5) are the functions of a_0, b_0, g_{ik}, b_{ik}:

$$c_0, c_1, \ldots, c_6 = f(a_0, b_0, g_{ik}, b_{ik}), \tag{6}$$

where:

a_0 – ratio of angular velocity ω to the angular self-frequency of stiff shaft, $a_0 = (\omega/\omega_c)^2$,
ω_c – angular self-frequency of stiff rotor, $\omega_c = \sqrt{c^*/m}$,
b_0 – ratio of Sommerfeld number to the relative elasticity of shaft, $b_0 = S_0/c_s$,
c^* – shaft stiffness (Nm^{-1}),
c_s – relative elasticity of shaft, $c_s = f/\Delta R = g/(\omega_c^2 \Delta R)$,
f – static deflection of shaft (m),
F – resultant force of oil film (N),
F_{stat} – static load of bearing (N),
g – acceleration of gravity (m·s^{-2}),
g_{ik} – dimensionless stiffness coefficients, $g_{ik} = S_0(\Delta R/F_{\mathrm{stat}})$,
g'_{ik} – stiffness coefficients (N/m),
b_{ik} – dimensionless damping coefficients, $b_{ik} = S_0(\Delta R/F_{\mathrm{stat}})\,\omega\, b'_{ik}$,
b'_{ik} – damping coefficients (Ns/m),
m – mass of the rotor (kg),
S_0 – Sommerfeld number, $S_0 = F\cdot\psi^2/(L\cdot D\cdot\eta\cdot\omega)$.

The coefficients of the characteristic frequency equation of 6th order [4] depend on the stiffness g_{ik} and damping b_{ik} coefficients, Sommerfeld number S_0, relative elasticity of shaft c_s and the ratio of angular velocity to the critical angular velocity of stiff rotor. As the result of transformations, the expression determining the ratio of boundary angular speed Ω_b to the critical one, ω_c, and the stability of rotor, has the form [1, 4, 12]:

$$\frac{\Omega_b}{\omega_c} = \frac{1}{1 + b_0\dfrac{A_3}{A_1}} \cdot \frac{A_2 A_3^2}{A_1^2 + A_1 A_3 A_4 + A_0 A_3^2}, \tag{7}$$

where: A_0, A_1, …, A_4 are the combination of four stiffness g_{ik} and four damping b_{ik} coefficients.

4. Results of calculations

The dynamic characteristics comprising the stiffness and damping coefficients as well as the stability regions of simple symmetric rotor are presented in Figs. 6 – 13. It was assumed that the bearing length to diameter ratios are L/D = 0.5 and L/D = 0.64 as well L/D = 0.8; for the first ratio the bearing relative clearance was ψ = 1.5‰ with different lobe relative clearances, for the second ratio ψ = 1.8‰ with the lobe relative clearance of considered profiles ψ_s = 3 (M) and ψ_s = 1 (C), but in the case of third ratio it was ψ = 1.5‰ and ψ_s = 1.2 (for both profiles M and OF). The rotational speed of journal was 3000 rpm and the feeding oil temperature T_0 = 40°C.

Figures 6 – 9 present the stiffness and damping coefficients of 2- and 3-lobe lobe journal bearings with the lobes of equal or different geometry. For the following stiffness coefficients (Fig. 6, 2-lobe) and the ranges of the Sommerfeld number values, i.e. g_{11}, g_{12}, g_{21}, g_{22} and S_0 (0 – 0.009), S_0 (0 – 0.32), S_0 (0 – 0.18), S_0 (0 – 0.21), the coefficients of the bearing OFM are larger than the ones of bearing 2M. However, for the values larger than given above, the coefficients of both types of bearing are almost equal (e.g. Fig. 6, coefficients g_{21}).

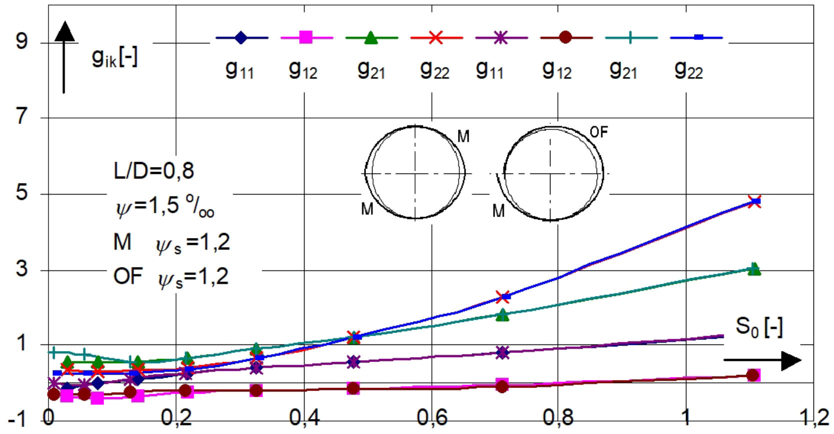

Figure 6. Stiffness coefficients of two types of 2-lobe journal bearings versus the Sommerfeld number.

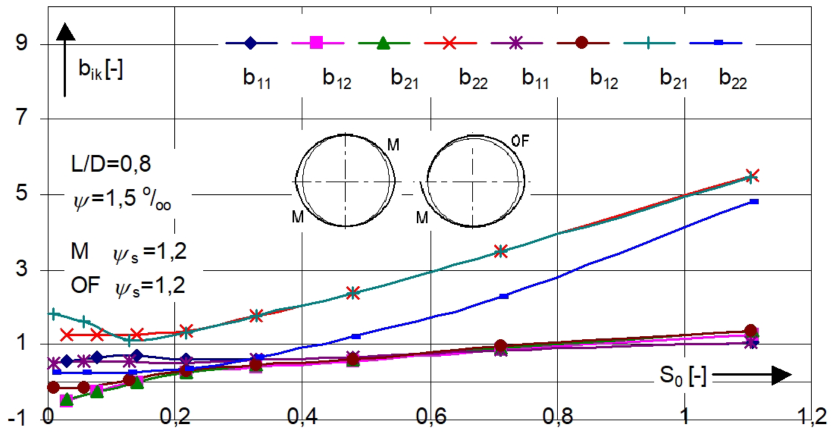

Figure 7. Damping coefficients of two types of 2-lobe journal bearings versus the Sommerfeld number.

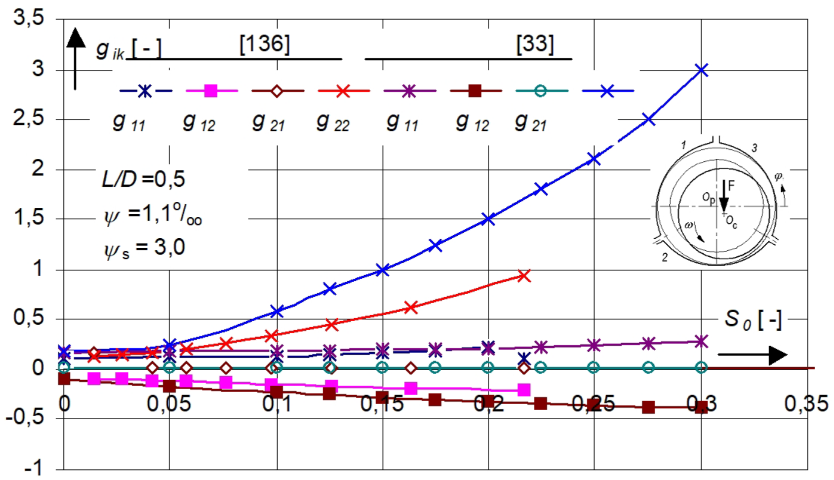

Figure 8. Comparison of experimentally [2] and theoretically [4] obtained stiffness coefficients of 3-lobe journal bearing versus the Sommerfeld number.

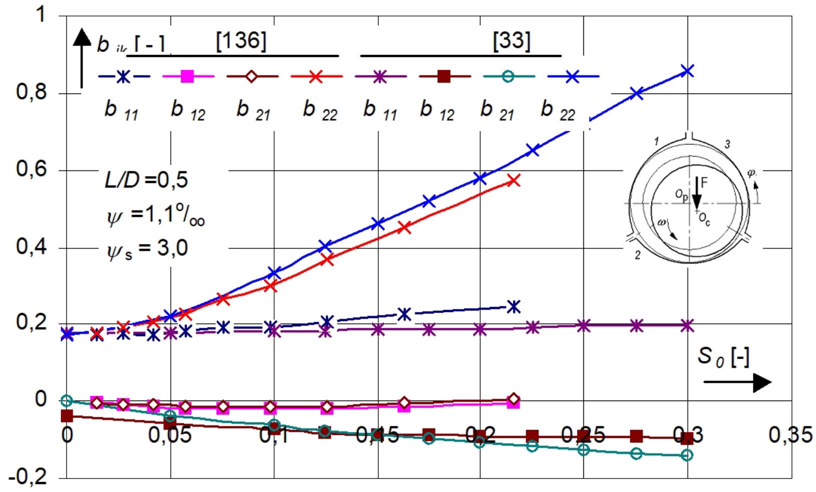

Figure 9. Comparison of experimentally [2] and theoretically [4] obtained damping coefficients of 3-lobe journal bearing versus the Sommerfeld number.

Figure 7 presents the damping coefficients of two types of 2-lobe journal bearings. The damping coefficients change in the following way: b_{11} – increases up to S_0 = 0.14, next shows the decrease up to S_0 = 0.35, and from this value there is an increase for both types of bearings; b_{12} – increases up to S_0 = 0.21 and from this value an increase is observed for both types of bearings, but for the OFM bearing b_{12} is larger than for the 2M bearing; b_{21} of bearing OFM is larger than for the bearing 2M, but the coefficient of the OFM bearing shows the decrease in the range $S_0 \in (0, 0.15)$, and next it increases for both types of bearings; b_{22} of bearing 2M is larger than for the OFM bearing.

The comparison of experimentally [2] and theoretically [4] obtained stiffness coefficients of 3-lobe journal bearings is presented in Fig. 8 and Fig. 9; there is similar run of all coefficients. It certifies the right assumptions and method of computation, that were applied in theoretical work [4].

The stability ranges of simple symmetric rotor operating in different type of 2- and 3-lobe lobe journal bearings versus the critical Sommerfeld number $S_{0k} = S_0 \cdot \omega / \omega_c$ are shown in Figs. 10 – 13 respectively. The coefficient tg τ (e.g. see Fig. 10) is the measure of stability properties of bearings [2, 4]. Larger values of angle τ mean the larger range of stability, i.e. at assumed load of bearing there is a higher boundary number of revolutions Ω_b / ω_c [13].

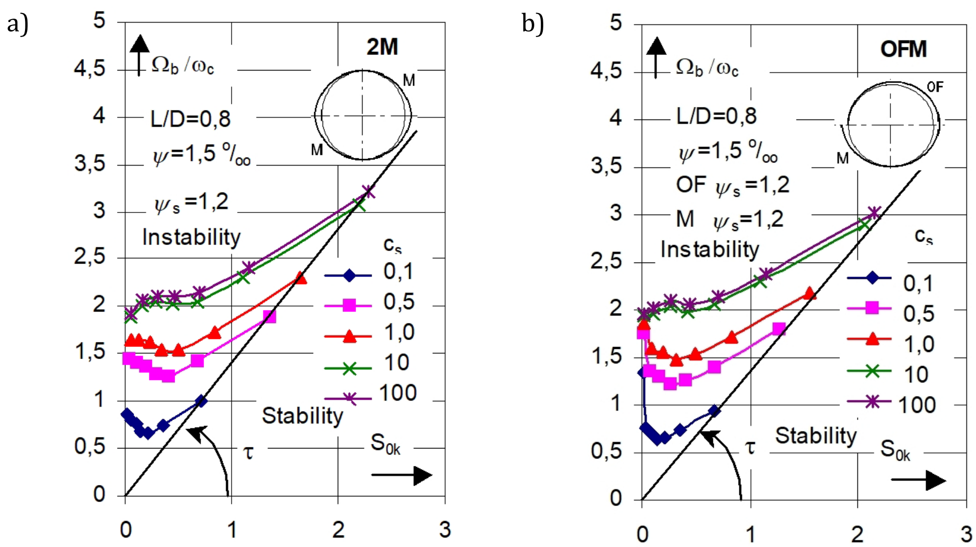

Figure 10. Stability ranges of different types of 2-lobe journal bearings versus the critical Sommerfeld number; the geometry of lobes: a) equal MM, b) different OFM.

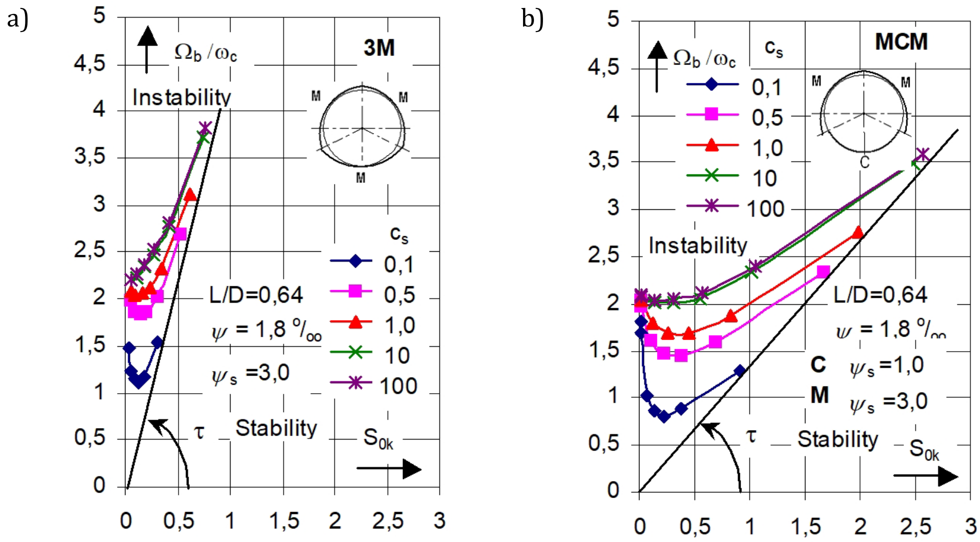

Figure 11. Stability ranges of different types of 3-lobe journal bearings versus the critical Sommerfeld number; geometry of lobes: a) equal MMM, b) different MCM.

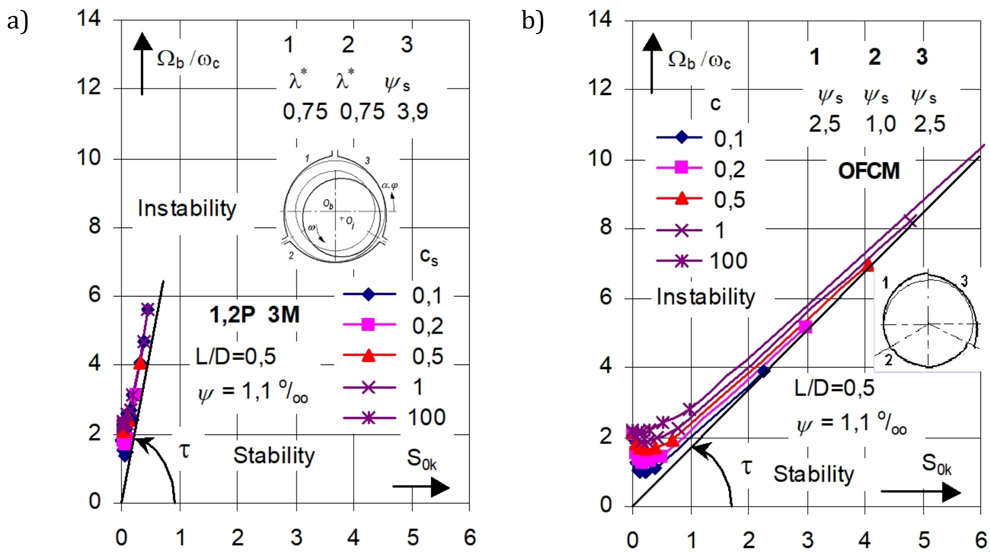

Figure 12. Stability ranges of different types of 3-lobe journal bearings versus the critical Sommerfeld number; geometry of lobes: a) equal MMM, b) different OFCM [5].

For the considered 2-lobe bearings, i.e. classic 2M and OFM, the stability of rotor and the run of curves are similar (Fig. 10). However, considering different variations fact needs more computations and analysis. Angle τ of 2M bearing (Fig. 10a) is larger than for OFM one (Fig. 10b).

It results from Fig. 11 that the stability of rotor operating in 3-lobe (3M) bearings is better than in case of bearing with the lobes of different geometry. The presence of cylindrical profile of bottom lobe causes the decrease in stability; the cylindrical journal bearings are characterized by lower stability than the multilobe bearings. Angle τ of 3M bearing (Fig. 12a) is larger than angle τ of OFM (Fig. 11b).

Figure 12 shows the results of the calculations of stability ranges of different types of 3-lobe journal bearings versus critical Sommerfeld number: the first and second lobe with the pericycloid profile (1,2P 3M, Fig. 12a) and the first offset, the second cylindrical and the third multilobe (OFCM, Fig. 13b).

In the case of 3-lobe bearings with different lobe geometries, the stability is better for the bearing that includes the lobes with pericycloid profile (1,2P 3M, Fig. 12a) and for this bearing the angle τ larger than in the case of bearing with the lobes OFCM.

4. Conclusions

The stiffness and damping coefficients of the oil film were determined for 2- and 3-lobe journal bearing of turbounits. The geometry of the lobes of considered multilobe journal bearings was assumed as equal or different. The characteristics obtained were used for the determination of the stability of simple elastic, symmetric rotor. Investigation into the stability of high-speed journal bearings was carried out on the assumption of equal or different geometry of bearings lobes.

Static and dynamic characteristics of the 2- and 3-lobe journal bearings of different lobes geometry can be obtained from a developed program for numerical calculations. The results from the developed program form the input data of the investigation and analysis of the stability of different types of multilobe journal bearings.

It was stated that the different geometry of the lobes of multilobe journal bearings has an effect on their static and dynamic characteristics and stability of rotor operating in such bearings. This fact gives new possibilities in the investigations into the problems of high-speed bearings.

Additional information

The author declares: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

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