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STATIC AND DYNAMIC CHARACTERISTICS OF HIGH SPEED MULTILOBE JOURNAL BEARINGS

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A B S T R A C T

Multilobe journal bearings operate at high speeds, low or medium external loads and at small eccentricities. These high performance bearings find the application in the responsible rotating machinery as the steam and gas turbines, turbochargers or turbo compressors. Excellent operating properties and reliable design of multilobe journal bearings can be achieved by calculation of static and dynamic characteristics and the correct choice of bearing geometry, bearing material, and lubricant.

Static characteristics of bearing include the oil film pressure and temperature distributions, minimum oil film thickness, power loss, oil flow, maximum oil film temperature. Dynamic characteristics are determined by four stiffness and damping coefficients of oil film.

The paper introduces the calculation of static and dynamic characteristics of multilobe journal bearings which are applied in the high speed rotating machinery. Reynolds, energy, viscosity and geometry equations were solved on the assumptions of thermo-hydrodynamic theory of lubrication. Aligned orientation of bearing and journal axis without deflections was assumed.

INTRODUCTION

Multilobe journal bearings applied in turbines and turbogenerators should assure long and reliable operation of these responsible rotating machinery. They are characterised by very good stability in the range of high rotational speeds assuring simultaneously very good cooling conditions for the oil film. Any failure occurring during operation of these bearings can cause very high power loses.

Multilobe journal bearings are mainly represented by 3-,4-, 5-lobe bearings (Flack 1984, Han 1979, Pinkus 1959, Someya 1953, Strzelecki 1993, 1996, 1999a, 2001a) with lubricating pockets placed between each lobe. These bearings are designed as the arc of the circle with the centre points placed on the symmetry line of the single lobe. The centre of each lobe do not coincide with the centre of circle inscribed in the bearing profile. In the symmetric multilobe bearing the circle inscribed in the bearing profile is tangent to the lobe exactly at the middle point of each lobe.

Static characteristics of bearing include the oil film pressure and temperature distributions, resulting force, attitude angle, minimum oil film thickness, power loss, oil flow, maximum oil film temperature (Strzelecki 2001b) Dynamic characteristics are determined by four stiffness and damping coefficients of oil film. They allow calculation of the stability of rotor- bearing system. Both type of characteristics can be obtained on the assumption of static equilibrium position of the journal.

Basic equations of thermo-hydrodynamic theory of lubrication allow to receive the oil film pressure, temperature and viscosity distributions. These distributions are the ground for the static characteristics of bearing, which are the input variables for the design of bearing. The above mentioned equations are derived on the assumptions of the adiabatic laminar or turbulent flow (Strzelecki 2002b, 2000a, b c, 2001b, 2002c) of incompressible Newtonian fluid of constant thermal conductivity and specific heat. For mathematical simplicity the pressure and viscosity are taken as constant through the thickness of the oil film.

Extreme temperatures (Strzelecki 2000b) of bearing for different geometric and operation parameters of multilobe journal bearings, allow to choose the best design and operating parameters of the bearing fulfilling at the same the conditions of the durable and reliable operation at aligned or misaligned axis of journal and bush (Strzelecki 2002d,e).

The static equilibrium positions of the journal for an assumed eccentricities were found by means of the iterative procedure. For faster convergence of the calculation of static equilibrium position angles, the Newton procedure has been applied.

Static and dynamic characteristics of bearings film have been computed for different sets of bearing parameters. Calculation have been performed in the conditions of the static equilibrium position of journal which allows the application of the results in the determining of stiffness and damping coefficients and stability of bearing and bearing system.

GEOMETRY OF OIL FILM

Geometry of multilobe journal bearings (Fig. 1 to Fig. 3) with parallel axis of journal and bush describe the equations (1) through (4).

$$H(\varphi, z) = H_c + \overline{H}_i(\varphi) \tag{1}$$

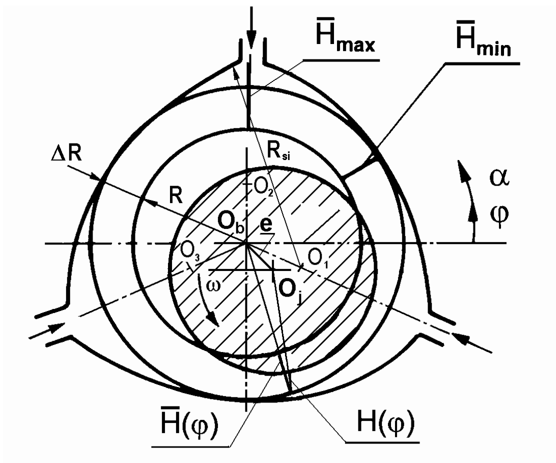

Fig. 1 Geometry of 3– lobe journal bearings; $\overline{H}(\varphi)$ - oil film thickness at concentric position of journal, $H(\varphi)$ - oil film thickness at an arbitrary position of journal, $\overline{H}_{max}$, $\overline{H}_{min}$ - maximum and minimum oil film thickness, R_{si} –lobe radius, R- journal radius, ΔR – minimum radial clearance, O_b, O_J – centre of bush and journal, O_1, O_2, O_3,–centres of lobes No. 1,2,3.

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The respective members of equation (1) have the following meaning:

$$H_c = 1 - \varepsilon \cdot \cos(\varphi - \alpha) \qquad (2)$$

where: ε- relative eccentricity, φ - peripheral co-ordinate, α - attitude angle,

determines the oil film thickness for the cylindrical bearing.
Dimensional equation of oil film thickness at concentric orientation of journal and bush can be read from Eqn. (2).

$$\overline{h}_i(\varphi) = R_{vi} - R + (R_{si} - R - \Delta R)\cos(\varphi - \gamma_i) \qquad (3)$$

Equation (3) determines the oil film thickness in concentric position of the journal and bearing bush axis [1].

$$\overline{H}_i(\varphi) = \psi_{si} + (\psi_{s_i} - 1)\cdot\cos(\varphi - \gamma_i) \qquad (4)$$

with: ψ_{si}- lobe clearance, γ_i - angle of the lobe centre point.

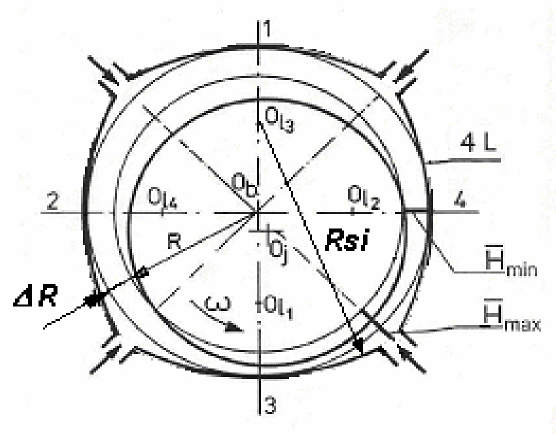

Fig. 2 Geometry of 4-lobe multilobe journal bearings (4L – 4lobe bearing, 4P - 4 lobe pericycloid on e(Strzelecki 2002a), 1, 2, 3, 4 – number of lobes,

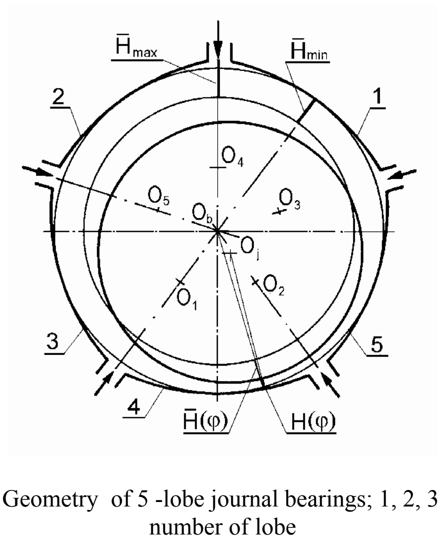

Fig. 3 Geometry of 5 -lobe journal bearings; 1, 2, 3, 4, 5 – number of lobe

OIL FILM PRESSURE AND TEMPERATURE

The oil film pressure, temperature and viscosity distribution have been determined from Reynolds, energy and viscosity equations (Han 1979, Strzelecki 2000b). These equations were derived on the assumption that in the bearing gap exists the laminar, adiabatic flow of non compressible Newtonian fluid characterised by the constant heat conductivity and specific heat. For the simplicity of considerations the pressure and viscosity were assumed as constant on the thickness of oil film layer. These assumptions allow the receiving of the equation of oil film pressure distribution

$$\frac{\partial}{\partial\varphi}\left(\frac{H^3}{\eta K_x}\frac{\partial\overline{p}}{\partial\varphi}\right) + \frac{\partial}{\partial z}\left(\frac{H^3}{\eta K_z}\frac{\partial\overline{p}}{\partial z}\right) = 6\frac{\partial H}{\partial\varphi} + 12\frac{\partial H}{\partial\phi} \qquad (5)$$

where: . D – bearing diameter (m), L – bearing length (m), H - dimensionless oil film thickness, $\overline{p}$ - dimensionless oil film pressure, $\overline{p} = p\psi^2/(\eta\omega)$, p – oil film pressure (MPa), η - dimensionless viscosity of lubricant, φ, z - peripheral and axial co-ordinates, O_x , K_x, K_z - turbulence coefficients, $\phi = \omega\ t$ - dimensionless time, t - time (s), ω- angular velocity,.

It was assumed that on the bearing edges exists the pressure $p(\varphi, \overline{z}) = 0$, and in the regions of negative pressures $p(\varphi, \overline{z}) = 0$. The oil film pressure received from equation (5) was put into energy equation allowing the receiving of the oil film temperature and viscosity distributions (Strzelecki 1993). The temperature $T(\varphi, \overline{z})$ on the bearing edges ($\overline{z} = \pm 1$) was computed by the method of parabolic approximation (Strzelecki 1993). Oil film viscosity was determined by the exponential equation.

RESULTS OF CALCULATIONS

Static characteristics were obtained for the bearings with length to diameter ratio L/D=0.5 and L/D=0.8, the lobe relative clearance $\psi_s = 2{,}0$ $\psi_s = 3$ and $\psi_s = 4$ as well as for the range of relative eccentricities ε = 0.1 to ε = 0.8. For the considered bearings the vertical direction of load $\beta = 270^o$ was assumed. The value of thermal coefficient was $K_T = 0{,}1$, $K_T=0{,}014$ and $K_T=0{,}5$ ($K_T = \omega \cdot \eta_0 / (c_t \cdot \rho \cdot g \cdot T_0 \cdot \psi^2)$ where: c_t – specific heat of oil, ρ - oil density, g – acceleration of gravity, T_0, η_0 – temperature and dynamic viscosity of supplied oil).
Oil film temperature distributions of 3-lobe and 4-lobe journal bearings are shown in Fig. 4 through Fig. 6. Higher temperatures can be observed in case of bearing with two lobes in the bottom part of bearing (LBL – load between lobes) than in case of bearing with the bottom lobe loaded (LOL – load on lobe). Oil film temperature distribution in 3-lobe journal bearing at parallel (q=0) and misaligned (q=0,2) axis of journal and bush. Misaligned orientation of axis affects the oil film temperature distribution (Strzelecki 1996).
When the bearing lobes are arranged in such way, that two lobes are at the bottom of bearing and the load acts between them, then the maximum oil film temperature is on the last lobe (lobe No. 3 - Fig. 5). In the case of lobe on load direction, the maximum value of oil film temperature occurs on the lobe which is on the load direction (case LOL; Fig. 5 and Fig. 6).
Sommerfeld numbers and the static equilibrium position angles obtained for 3-, 4-, and 5-lobe journal bearings are given in Fig.7 through Fig. 10. In the range of relative eccentricities of journal ε=0,1 to ε=0,7 the largest Sommerfeld numbers are for the 3-lobe journal bearing and the smallest ones shows the 5-lobe journal bearing (Fig. 7 and Fig.8). Static equilibrium position angles have the largest values for 3-lobe journal bearing in the range of Sommerfeld numbers larger than So= 0,05 (Fig. 9 and Fig. 10). Comparison of static equilibrium position angles obtained by different authors for 4-lobe journal baring, is given in Fig. 11).
Sommerfeld numbers and static equilibrium position angles obtained on the assumption of turbulent oil film at different

Reynold's numbers are shown in Fig. 12 through Fig. 17. An increase in Reynold's numbers causes the increase in Sommerfeld number and static equilibrium position angles.
Power loss P_{loss} and oil flow q_{fl} obtained at different lobe numbers and for are shown in Fig. 18 and Fig. 19; the largest power losses occur in case of 5-lobe bearing in range of small values of Sommerfeld numbers (Fig. 18). The oil flow has the largest values for 5-lobe journal bearing too. (Fig. 19).
Maximum oil film temperature T_{max} obtained or turbulent oil film and at one value of heat number K_T=0,1 is shown in Fig. 22. Comparison of maximum temperatures obtained at two values of heat number K_T=0,1 and K_T= 0,5 is shown in Fig. 23.
For different arrangement of 4-lobe journal bearings, i.e. for the load directed between the lobes and on the lobe, the maximum oil film temperature is given in Fig. 24. Maximum temperatures of oil film are larger in the case of bearing with the load acting between the lobes.
Dynamic characteristics, expressed by four stiffness and four damping coefficients of oil film, as a function of Sommerfeld numbers, are shown in Fig. 25 through Fig. 32. Fig. 25 through Fig. 30 give the dynamic characteristics for the bearing loaded by force acting on the bottom lobe. There is an increase in the values of coefficients g_{11}, g_{22} at an increase in lobe number (Fig. 25, Fig. 27 and Fig. 29). the coefficients g_{12} show the decrease and g_{21} very small changes for the increase in lobe number (e.g. Fig. 27 and Fig. 29).
An increase in the lobe number causes the increase in damping coefficients b_{11} and b_{22} (Fig.26, Fig. 28 and Fig. 30). Damping coefficients b_{12} and b_{21} have the same values and they show the decrease at the increase in the lobe number (e.g. Fig.26 and Fig. 28).

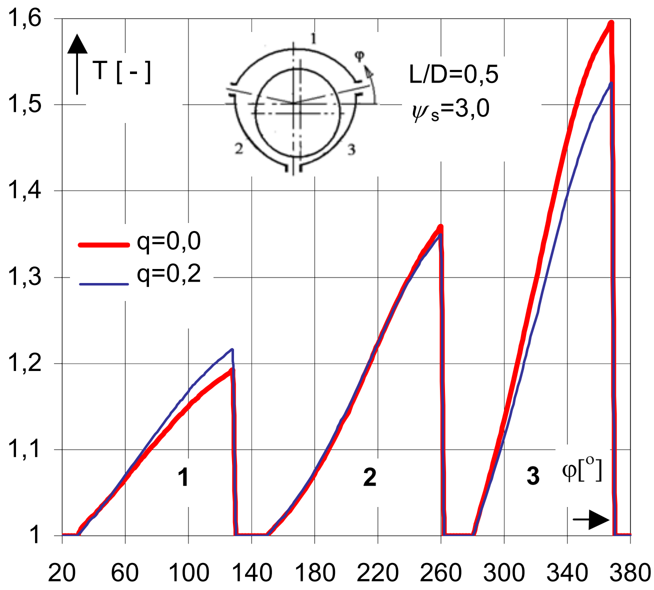

Fig. 4 Oil film temperature distribution in 3-lobe journal bearing at parallel (q=0) and nonparallel (q=0,2) axis of journal and bush [Strzelecki]; load between lobe 2 and 3.

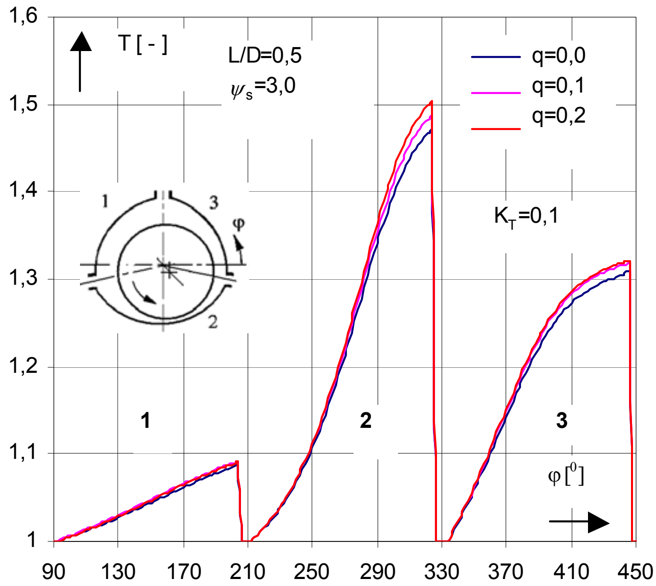

Fig. 5 Oil film temperature distribution in 3-lobe journal bearing at parallel (q=0) and nonparallel (q=0,2) axis of journal and bush [2]; load on lobe 2.

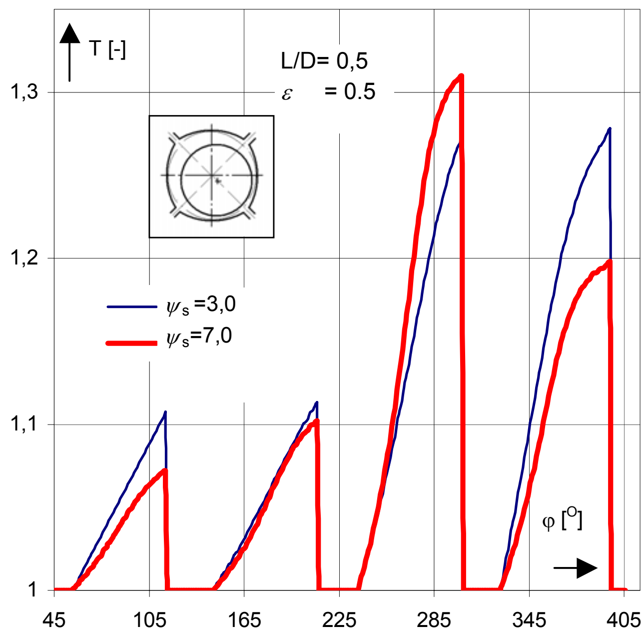

Fig. 6 Oil film temperature distribution in 4-lobe journal bearing (load on the bottom lobe)

The case of dynamic characteristics obtained for the load acting between the lobes is given in Fig. 31 and Fig. 32. It was stated ,that there is larger effect of bearing lobes arrangement on the coupled stiffness g_{12} , g_{21} (Fig. 25 and Fig. 31) and damping b_{12} ,b_{21} coefficients (Fig. 26 and Fig. 32) than on the other coefficients.

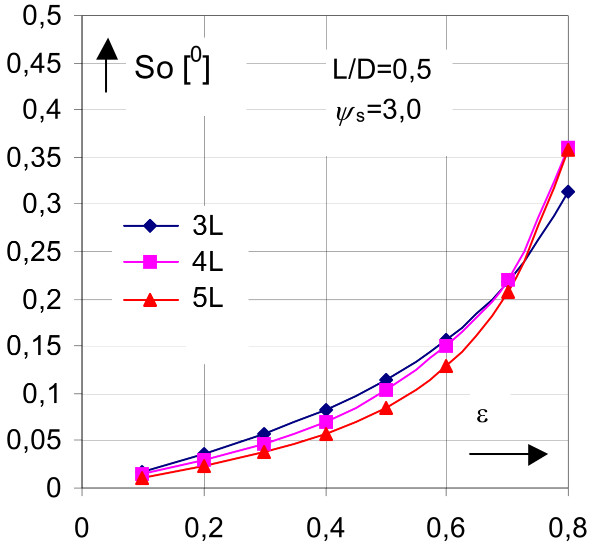

Fig. 7 Sommerfeld number for different types of bearings

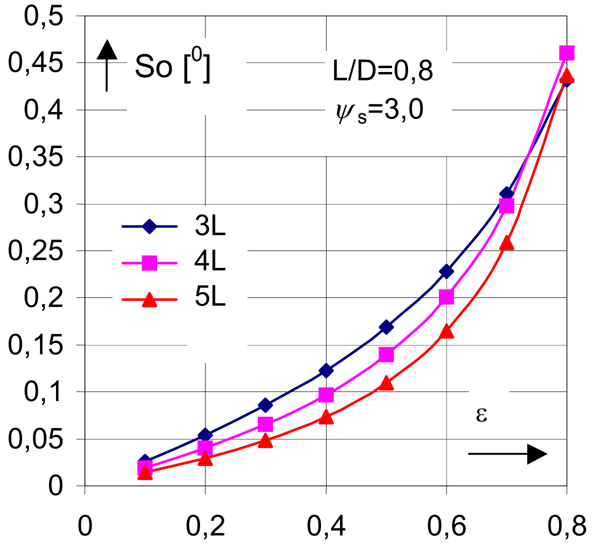

Fig. 8 Sommerfeld number for different types of bearings

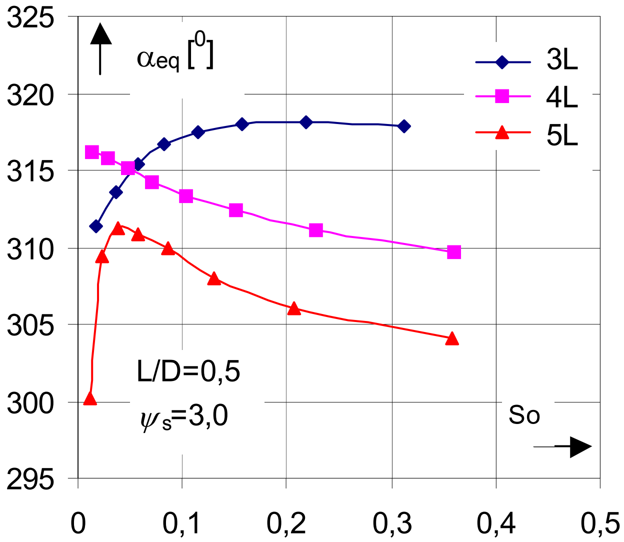

Fig. 9 Static equilibrium position angle for different bearings

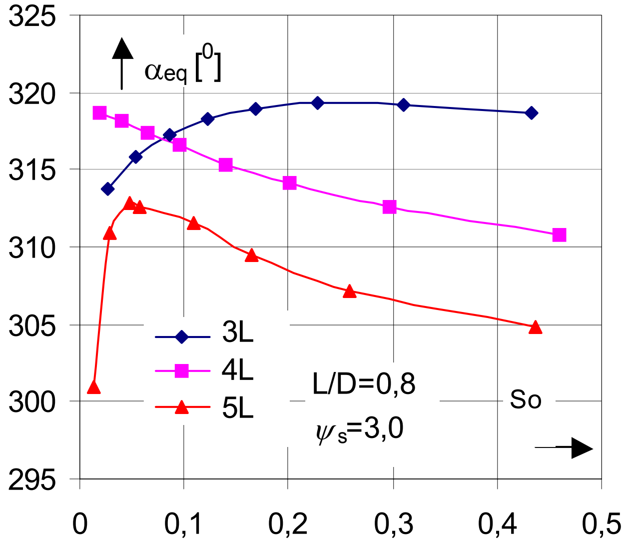

Fig. 10 Static equilibrium position angle for different bearings

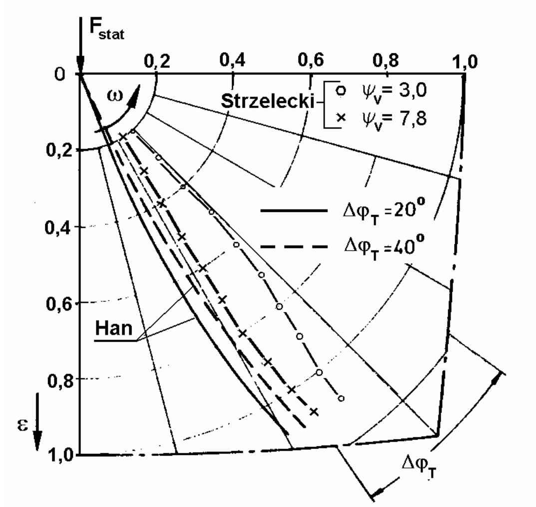

Fig. 11 Static equilibrium position lines of 4 lobe journal bearing with L/D=0,5; (Han (1979) ψ_v=7,8 3 and Strzelecki (1993) - ψ_v=3,0 and ψ_v=7,8

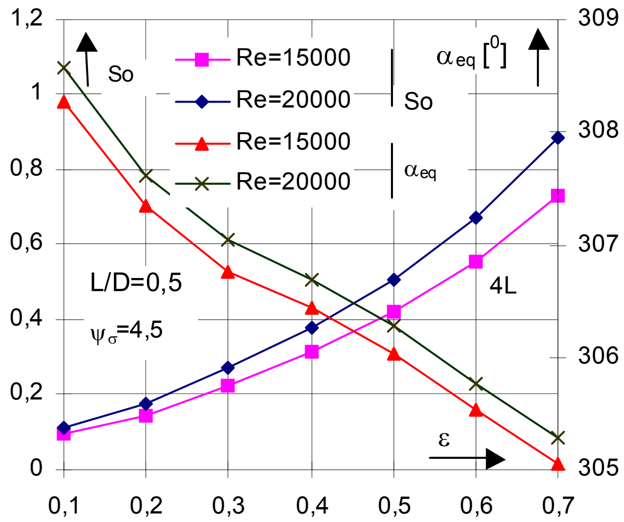

Fig. 12 Sommerfeld number and static equilibrium position angle for different Reynolds numbers

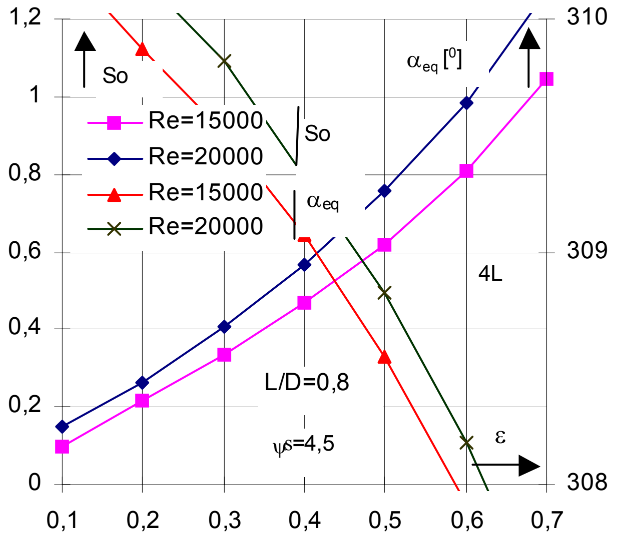

Fig. 13 Sommerfeld number and static equilibrium position angle for different Reynolds

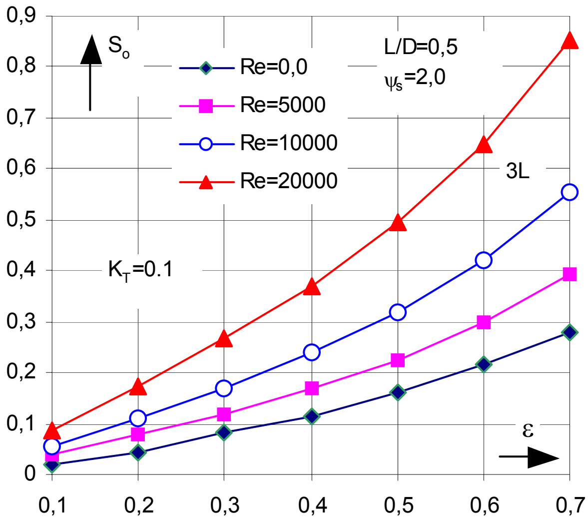

Fig. 14 Sommerfeld number for different Reynolds numbers

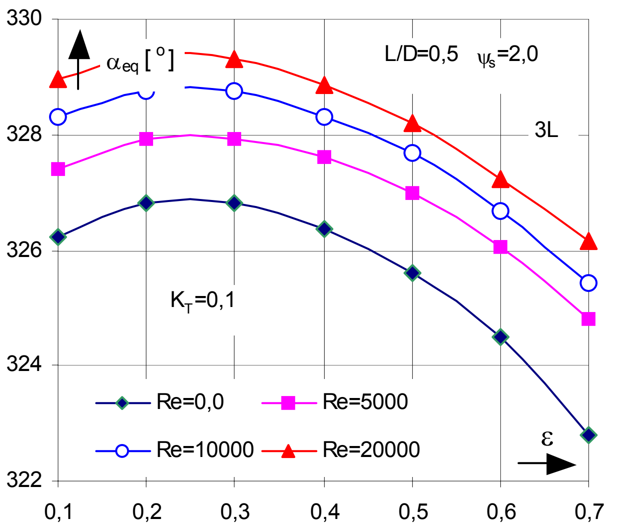

Fig. 15 Static equilibrium position angle for different bearings

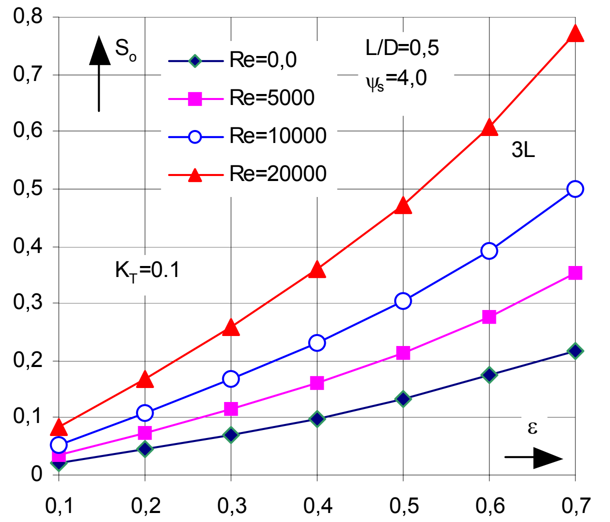

Fig. 16 Sommerfeld number for different types of bearings

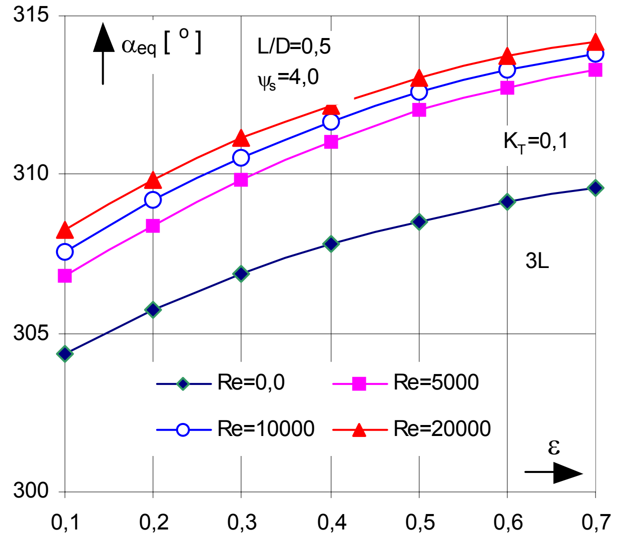

Fig. 17 Static equilibrium position angle for different Reynolds numbers

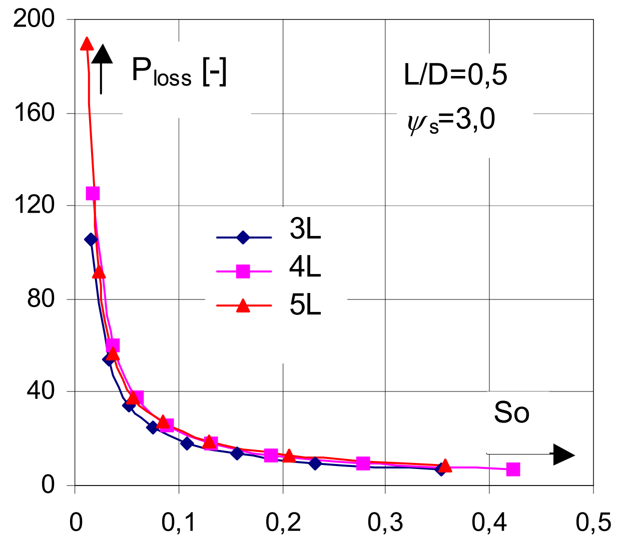

Fig. 18 Power loss for different types of bearings

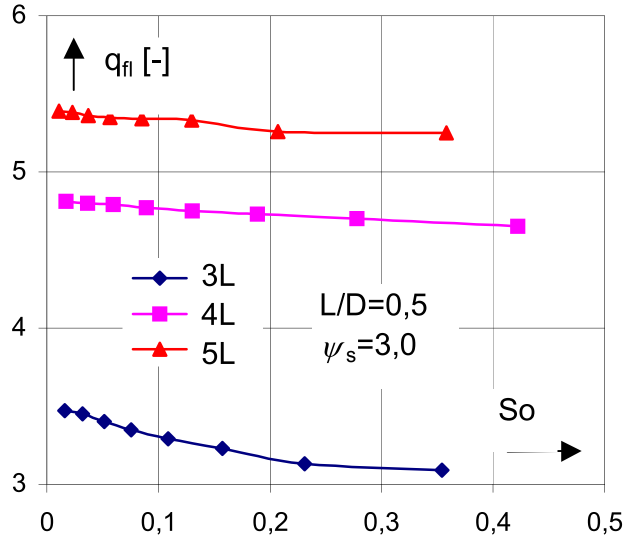

Fig. 19 Oil flow for different types of bearings

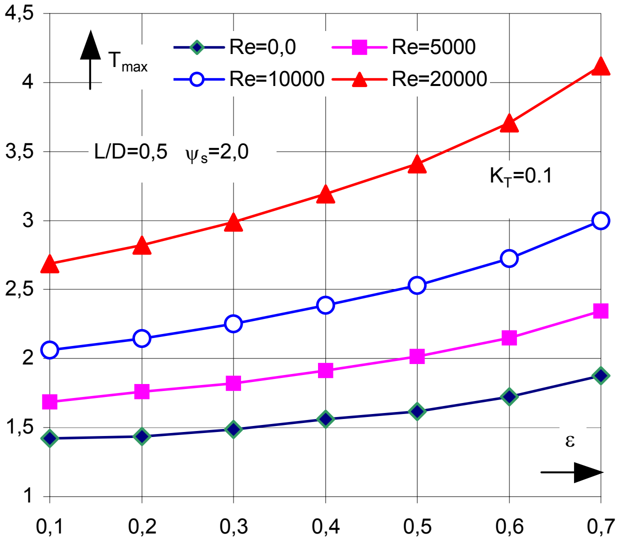

Fig. 22 Maximum oil film temperature for different Reynolds numbers

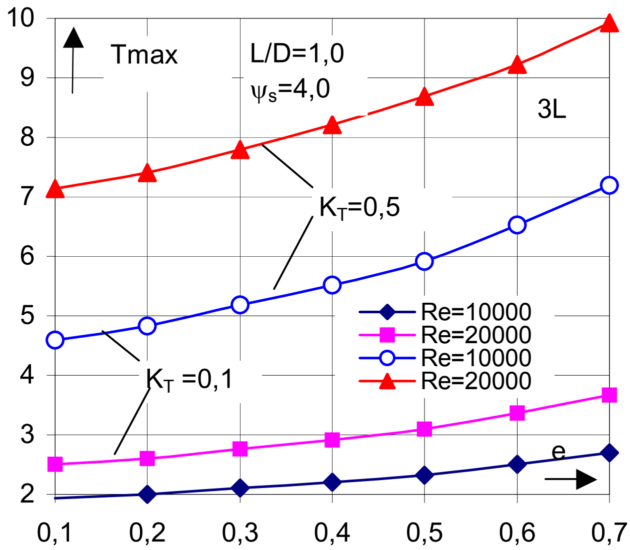

Fig. 23 Maximum oil film temperature for different different Reynolds numbers

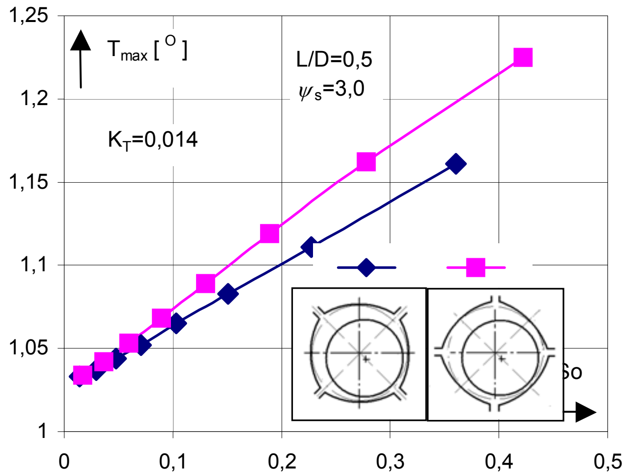

Fig. 24 Maximum oil film temperature for different arrangement of 4-lobe journal bearings

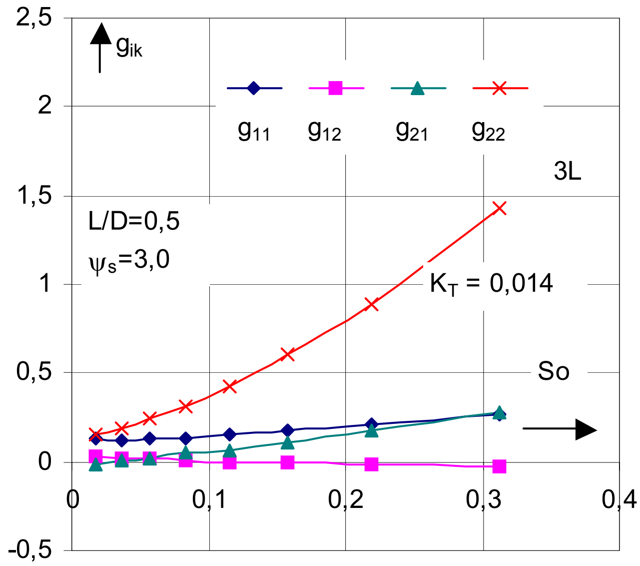

Fig. 25 Stiffness coefficients of 3-lobe journal bearing

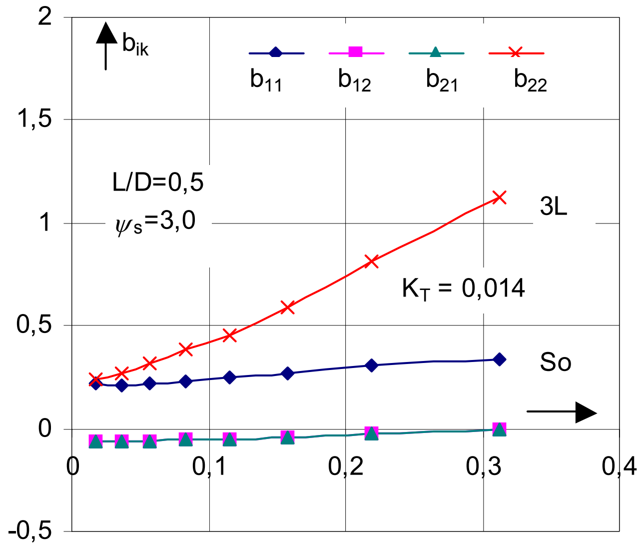

Fig. 26 Damping coefficients of 3-lobe journal bearing

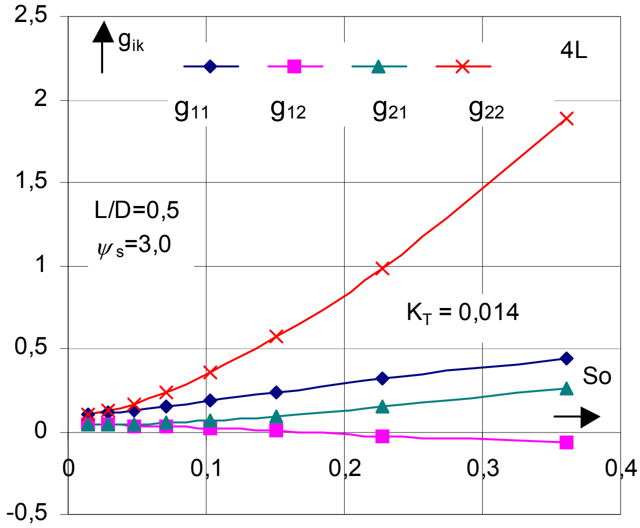

Fig. 27 Stiffness coefficients of 4-lobe journal bearing

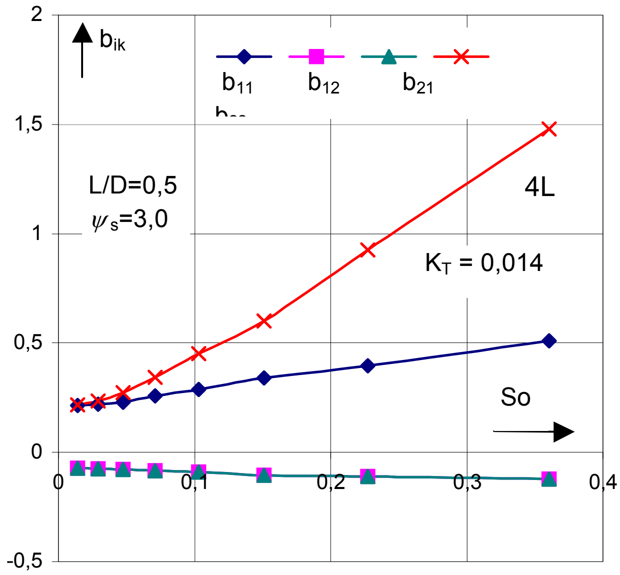

Fig. 28 Damping coefficients of 4-lobe journal bearing

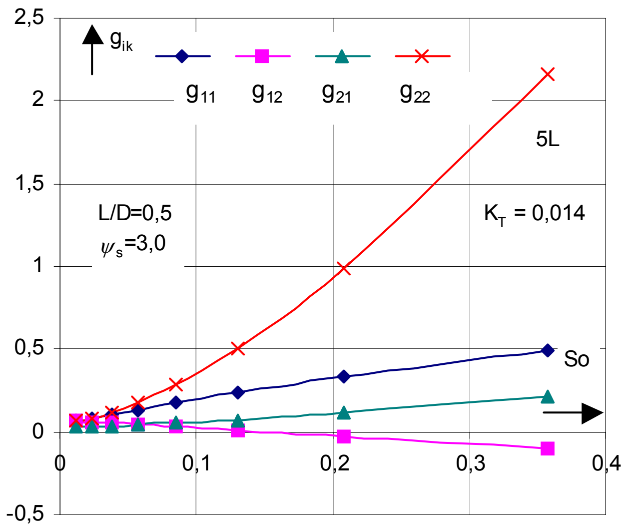

Fig. 29 Stiffness coefficients of 5-lobe journal bearing

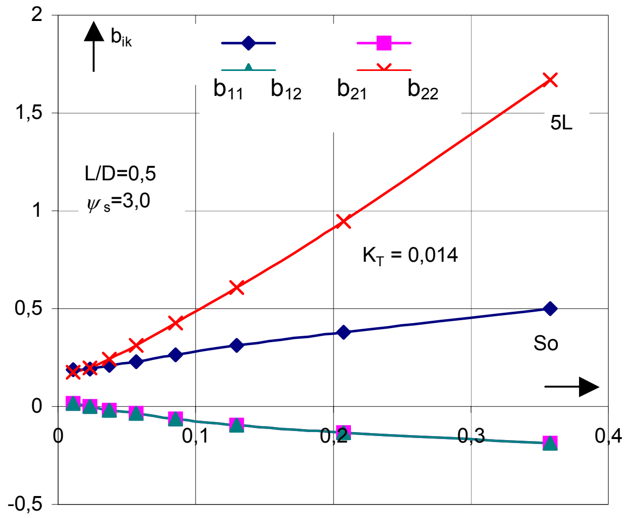

Fig. 30 Damping coefficients of 5-lobe journal bearing

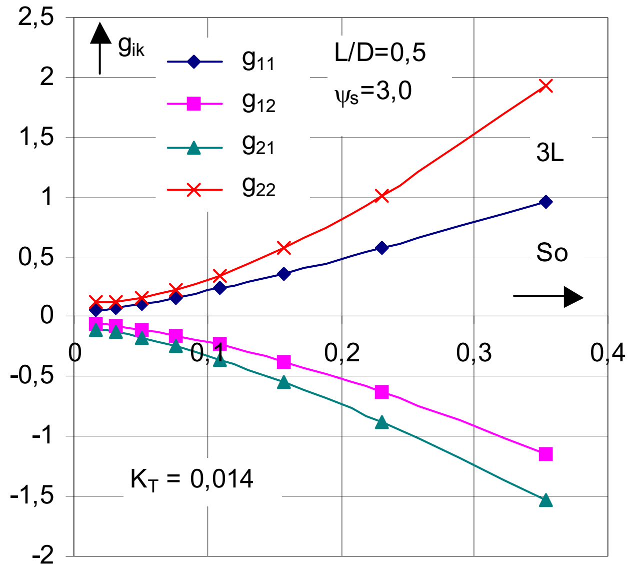

Fig. 31 Stiffness coefficients of 3-lobe journal bearing (load between lobes- LBL))

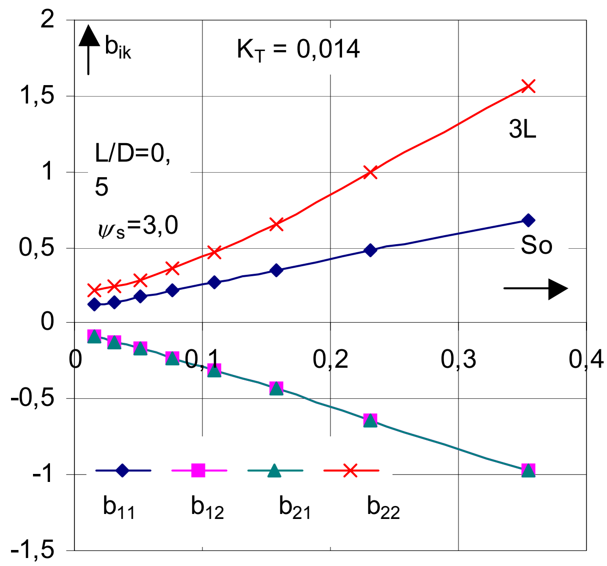

Fig. 32 Damping coefficients of 3-lobe journal bearing (load between lobes –LBL)

CONCLUSIONS

Calculation of static and dynamic characteristics of high speed journal bearings was carried-out. The results presented in this paper allow for the following conclusions:

1. The code of the computation of static and dynamic characteristics of high speed journal bearings was developed.
2. Oil film temperature distributions are affected by the arrangement of bearing lobes with regard to the load direction
3. Model of oil film, i.e. laminar adiabatic or or turbulent adiabatic has an effect on the static and dynamic characteristics of bearing.
4. Power loss shows small changes for considered types of bearings but they increase with the increase in lobe number.
5. There is an increase in oil flow at the increase in Sommefeld number.
6. Maximum oil-film temperature increases with the increase in Reynolds number.
7. An increase in the lobe number changes the dynamic characteristics affecting the stiffness and damping coefficients.

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