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JOURNAL BEARINGS OF MULTILOBE PROFILES

ŁOŻYSKA ŚLIZGOWE WIELOPOWIERZCHNIOWE

***Summary**: This article considers the static characteristics of multilobe, classic journal bearings as well as the multilobe bearings with the lobes of different geometry. For obtaining of these characteristics the equations of Reynolds, energy and viscosity as well as the geometry of oil film have to be solved numerically by means of finite differences method. The assumptions that are usually applied concern the laminar, adiabatic oil film, and parallel orientation of the axes of journal and sleeve and the conditions of the static equilibrium positions of journal.*

***Keywords**: sliding bearings, static characteristics of multilobe bearings, classic journal bearings, multilobe bearings with lobes of different geometry, Reynolds equations, oil film geometry, computer simulation, bearing design*

***Streszczenie**: W artykule przedstawiono charakterystyki statyczne klasycznych łożysk wielopowierzchniowych, jak również łożysk wielopowierzchniowych z segmentami o różnej geometrii. Charakterystyki otrzymano z równań Reynoldsa, energii i lepkości, oraz geometrii filmu smarowego – zastosowano rozwiązanie numeryczne metodą różnic skończonych. Założono laminarny, adiabatyczny film smarowy, równoległe położenie osi czopa i panewki oraz warunki statycznego położenia równowagi czopa.*

***Słowa kluczowe**: łożyska ślizgowe, charakterystyki statyczne łożysk wielopowierzchniowych, klasyczne łożyska wielopowierzchniowe, łożyska wielopowierzchniowe z segmentami o różnej geometrii, równanie Reynoldsa, energii, geometria filmu smarowego, symulacja komputerowa, projektowanie łożysk*

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. There are the needs 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 shows Fig. 1 [2].

In large multistage turbounits the journal bearings have the basic effect on their performances [2–6]. The main task of these, mainly 2- and 3-lobe journal bearings [1–3] is to assure the operation of turbounits at the assumed temperature and minimum power losses, at correct vibration frequency of shafts line and the largest resistance against the accidental external loads that cause the unstable behaviour of the rotor. Fulfilling of these tasks is very important for both the designer and the operator of the rotating machinery.

The investigation into new types of 2-lobe journal bearings of turbo generator points out on the increase in the bearing load capacity. The newly developed bearing should meet the rotor dynamic criteria, be interchangeable with existing bearings and the power losses as well as the lubricating oil supply

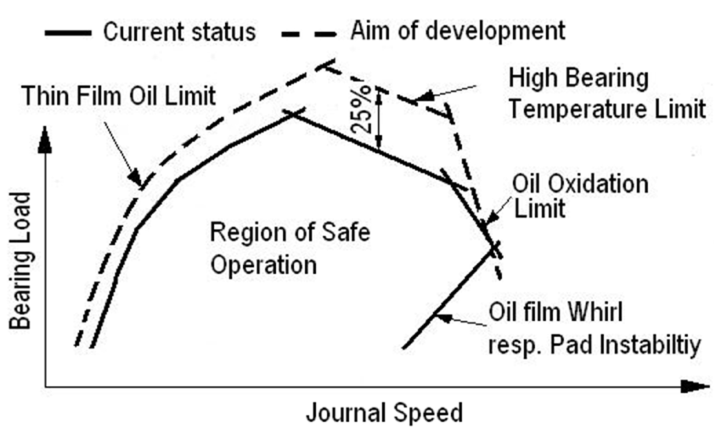
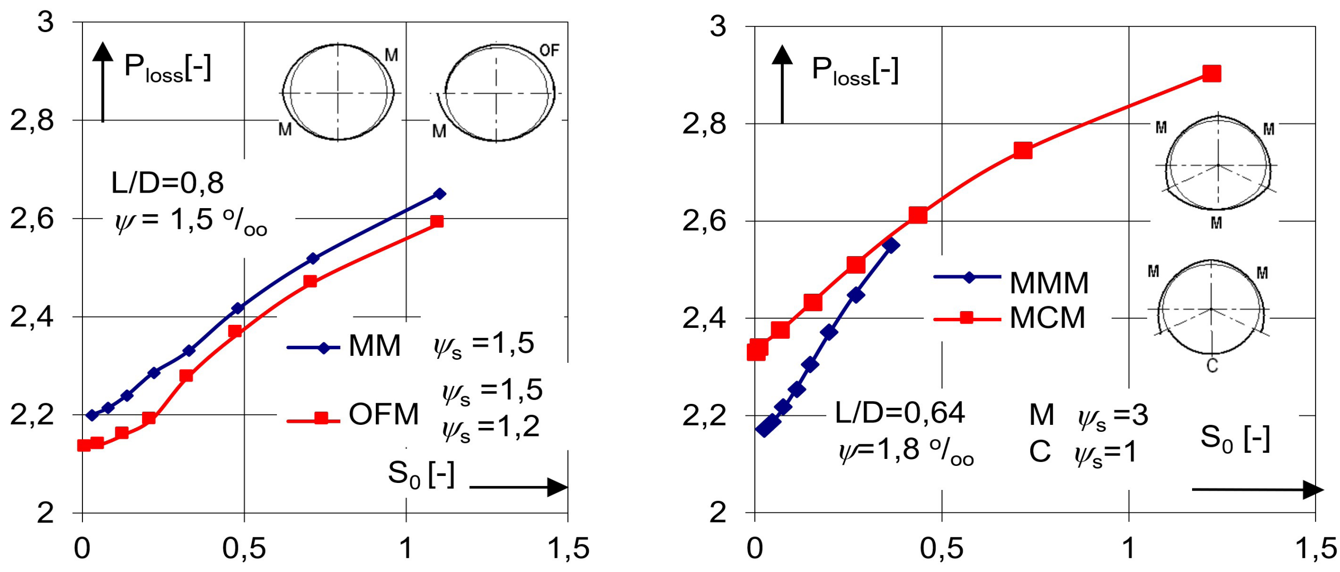


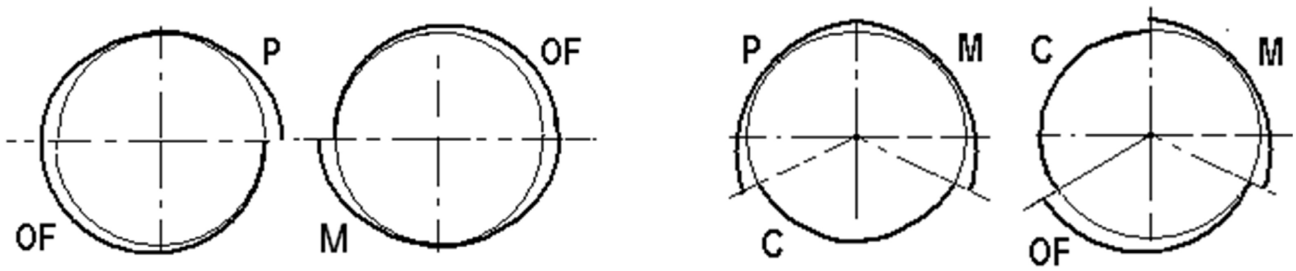
Fig. 1. Current status and aim of development of turbounits journal bearings [2]



a)

b)

Fig. 2. Power loss of two types of two types of 2- and 3-lobe (a and b) journal bearings versus Sommerfeld number (L/D- bearing length to diameter ratio, P_{loss} –dimensionless power loss, ψ – bearing relative clearance, ψ_s – lobe relative clearance (for the cylindrical lobe ψ_s =1), M, OF – multilobe and Offset profile, respectively [3]



a)

b)

Fig. 3. Multilobe journal bearings with the lobes of different geometries; a) 2-lobe, b) 3-lobe; profiles: C – cylindrical, M –classic (discontinuous), P – pericycloid, OF- Offset

requirements should be maintained. The paper [2] 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 and caused an increase in the load capacity by 25%.

Static characteristics of the journal bearings [1–3] consists the oil film pressure and temperature fields, load capacity (expressed by the Sommerfeld number), minimum oil film thickness, maximum value of oil film pressure and temperature, friction losses, oil flow. These all parameters are very important in the process of reliable bearing design. The profile of bearing bore has an effect on the static characteristics of the bearing and they knowledge allows for correct design of single bearing and bearing system.

For ensuring the reliable operation of the bearing and rotor-bearing system it is important to know its the 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 [3, 6].

The world bibliography on theoretical and experimental studies into the multilobe journal bearings to date, gives no information on the static and dynamic characteristics of bearings with the lobes of different geometry.

New properties of journal bearings can be obtained by application in one multilobe bearing [3] the lobes with a different geometry, e.g. circular (C), classic multilobe (M), pericycloid (P) or Offset (OF) profiles. The design of such bearings (Fig. 3) may provide e.g. larger hydrodynamic load capacity, different temperature conditions of the lubricating film or a change (reduction) in the resistance to motion (power losses).

This article considers the static characteristics of multilobe, classic journal bearings as well as the multilobe bearings with the lobes of different geometry. For obtaining of these characteristics the equations of Reynolds, energy and viscosity as well as the geometry of oil film have to be solved numerically by means of finite differences method. The assumptions that are usually applied concern the laminar, adiabatic oil film, and parallel orientation of the axes of journal and sleeve and the conditions of the static equilibrium positions of journal.

Classification, selection and application of multilobe journal bearings

The classification of multilobe journal bearings with fixed lobes and tilting-pad show Fig. 4. Bearings with offset lobes are referred to as "Offset" bearings [1, 3] and are applied in accelerating turbo-gear-trains; they can operate in one direction, only. Figure 4 does not include the foil bearings that are characterized by a flexible support of a sliding foil [3, 7].

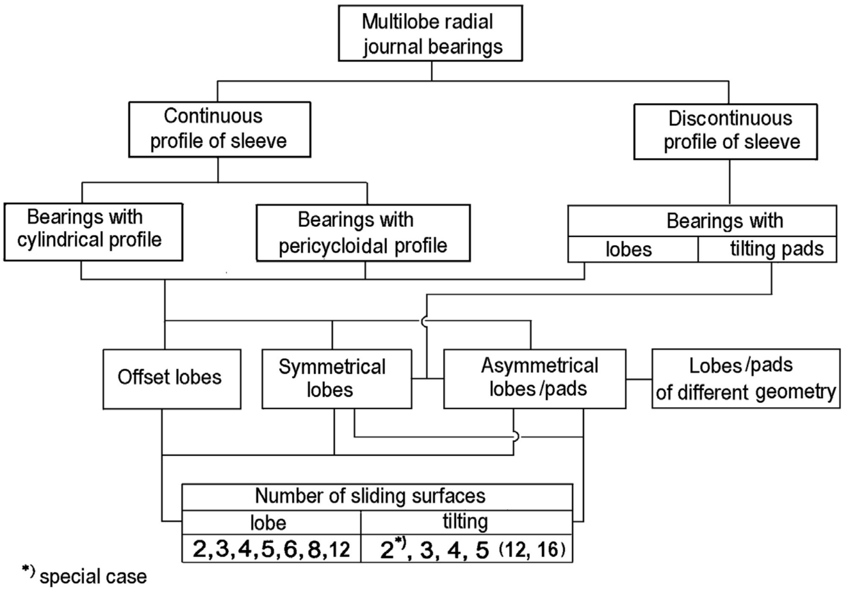
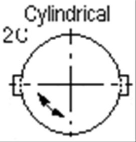
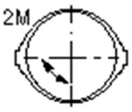
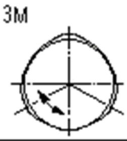
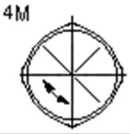


Fig. 4. Classification of multilobe journal bearings; 6, 8, 12 lobes – grinding machines (12 or 16 tilting-pads used as the bearings positioning the vertical rotors of water turbines [3, 7])

Table 1. Characteristic of the chosen types of multilobe journal bearings

Type of bearing	Peripheral speed	Pressure 1MPa=1N/mm^2	Sommerfeld number	Stiffness Damping	Cost	Application
Cylindrical 2C 	0....30 (35) m/s	0,2...4,5 (5) MPa	0,5...10	o ooooo	o	Gearboxes Steam tubines Electric motors Generators
Lemon 2M 	25....65 (70) m/s	0,2...3,5 (4) MPa	0...1,5	oo oooo	oo	Gearboxes Steam tubines Electric motors Generators
3M 	30....90 (95) m/s	0,1...3,0 (3,5) MPa	0...1,0	ooo ooo	oo	Generally: shaft of low and high rotational speeds Turbocompressors
Offset Halves 2OF 	25....70 (80) m/s	0,2...3,5 (4,0) MPa	0...2,0	oo oooo	oo	Turbines Accelerating gear trains Steam turbines
4M 	30...100 m/s	0,1...2,0 (2,5) MPa	0...1,0	ooo o	oo	Gearboxes Turbopumps Steam turbines Spindles of working machines
4M - 4- lobe, ooooo - high o - low Maximum values given in parenthesis						

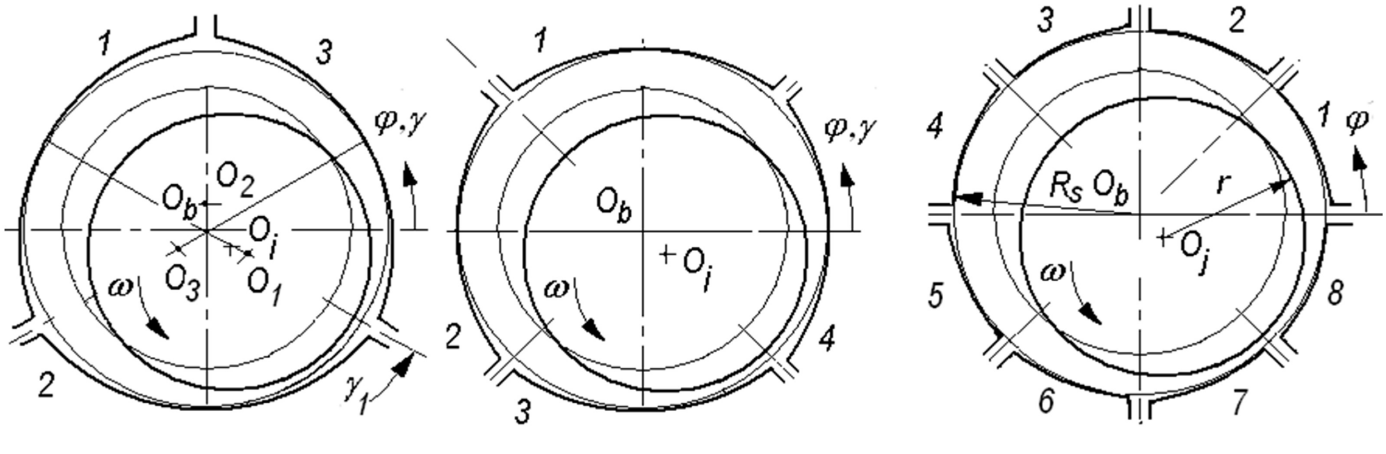
The characteristics of selected multilobe bearings with a discontinuous profile of the sleeve bore in terms of peripheral speeds, pressures, dimensionless load capacity S_0 (Sommerfeld number), stiffness and damping characteristics, manufacturing costs and applications are shown in Table 1; the information provided can be used to choose a bearing for a specific application.

The circular (cylindrical) profile of the bearing bore is the most simplest and creates a single sliding surface, but does not ensure proper operating conditions in high-speed rotating machinery.

By dividing the bearing circumference into lobes of the same or different length and separated by lubricating grooves, a multilobe bearing with several sliding surfaces is obtained (Fig. 5 and 6). If the bearing lobes are arcs with a radius equal to the radius of the circle inscribed in the contour of sleeve, then it is a multi-lobe bearing with cylindrical lobes [1–9]. A classic multilobe bearing is characterized by the fact that the centre of the lobe radius is shifted by a certain distance in relation to the centre of the circle inscribed in the outline of the sleeve (Fig. 5, O_bO_2 – shift of the centre of the radius).

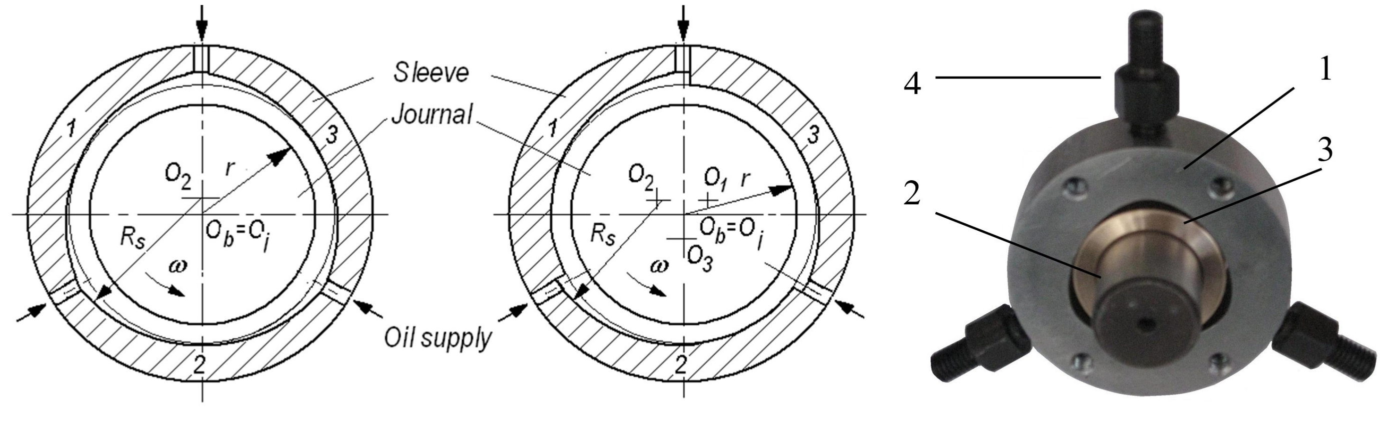
The continuous profile of the bore opening provides a multilobe pericycloid bearing [3, 4]. The bearings described above can operate in two directions of rotation.

Figure 6c shows the multilobe floating ring bearing with the pericycloid 3-lobe sleeve. General view of 4 – and 8-lobe journal bearings applied in grinding machines presents Fig. 7 (Fig. 7a – radial-axial thrust bearing).



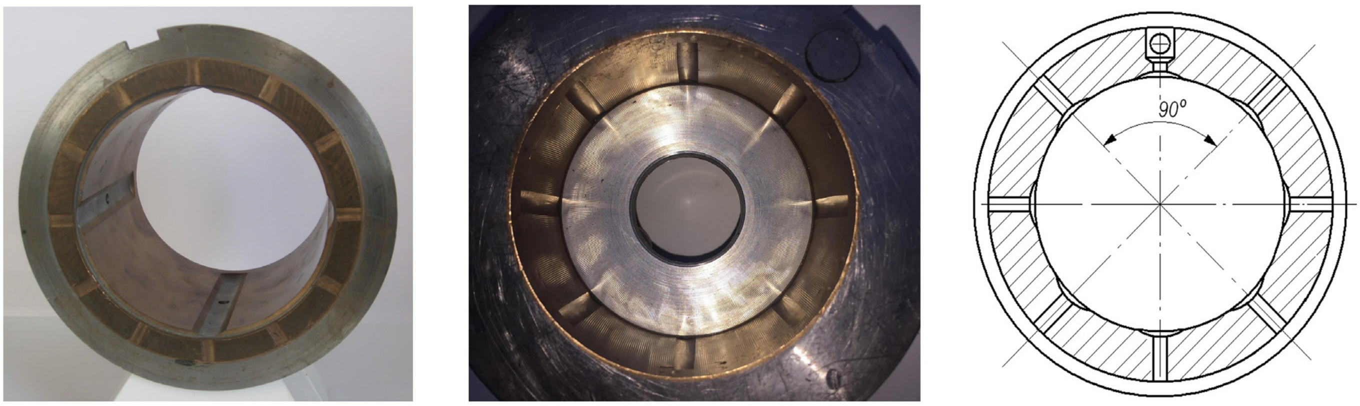
a) b)

Fig. 5.: 3-, 4- lobe classic (a) and 8-lobe Offset (b) journal bearings (lobes of the equal geometry); O_b, $O_{1,2,3}$, O_j – center of sleeve, lobes and journal, R_s, r – lobe and journal centers, respectively, φ – peripheral coordinate, γ – co-ordinate of the segment center



a) b) c)

Fig. 6. Geometry and design solution of 3-lobe journal bearings: 3M and 3OF; O_1, O_2, O_3 – position points of the sliding surface radius, 1, 2, 3 – sliding surfaces; c- assembly of floating ring bearing with the 3-lobe sleeve of pericycloid profile (1 –sleeve, 2 – journal, 3 – floating ring, 4 – oil supply) [3, 7]



a) b) c)

Fig. 7. General view of 4- and 8-lobe journal bearings; a) 4-lobe, b) 8-lobe, c) cross-section of 8-lobe bearing [3]

The basic types of sleeves geometry (i.e. C, M, P, OF) allow the design of special multi-lobe bearings with the lobes of various profiles, e.g. circular, classic multilobe, Offset and pericycloid (Fig. 8) [3]. Such solutions of the bearing geometry ensure the static and dynamic characteristics that are different from those of the classic multilobe bearings used so far.

An example of the application of multilobe bearings is the accelerating planetary gear of the drive system of UNITURBO SULZER high-speed compressor operating at output rotational speed 18000 rpm [3]. The bearings of central shaft with blades wheel (Fig. 9, item 1) are 4-lobe with fixed lobes (Fig. 9, items 2 and 4). Satellites gears (Fig. 8, item 5) rotate on a fixed bronze sleeve with four sliding surfaces on the external peripheral (Fig. 9 b, c).

Figure 10 shows the design solutions of 4-lobe journal bearings with fixed (Fig. 10a), rotating (Fig. 10 b, c) sliding surfaces.

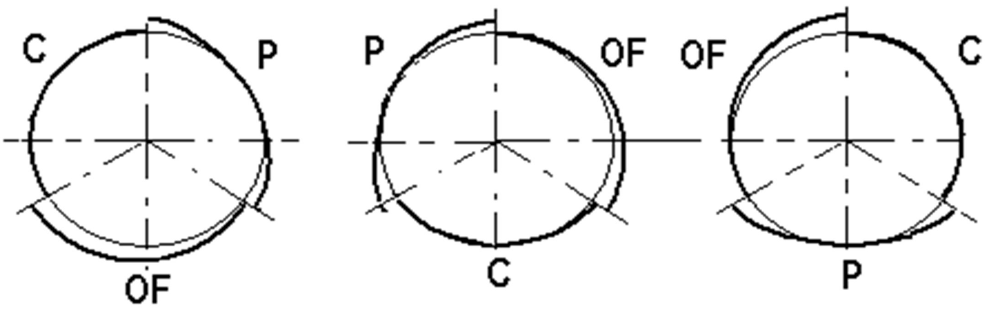


Fig. 8.: 3-lobe journal bearings with the lobes of different geometry

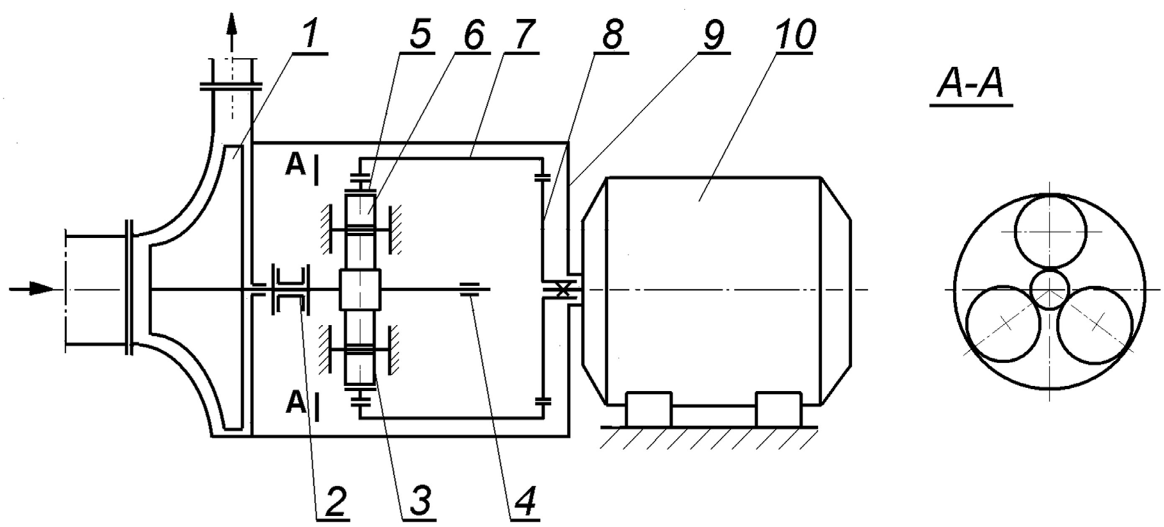
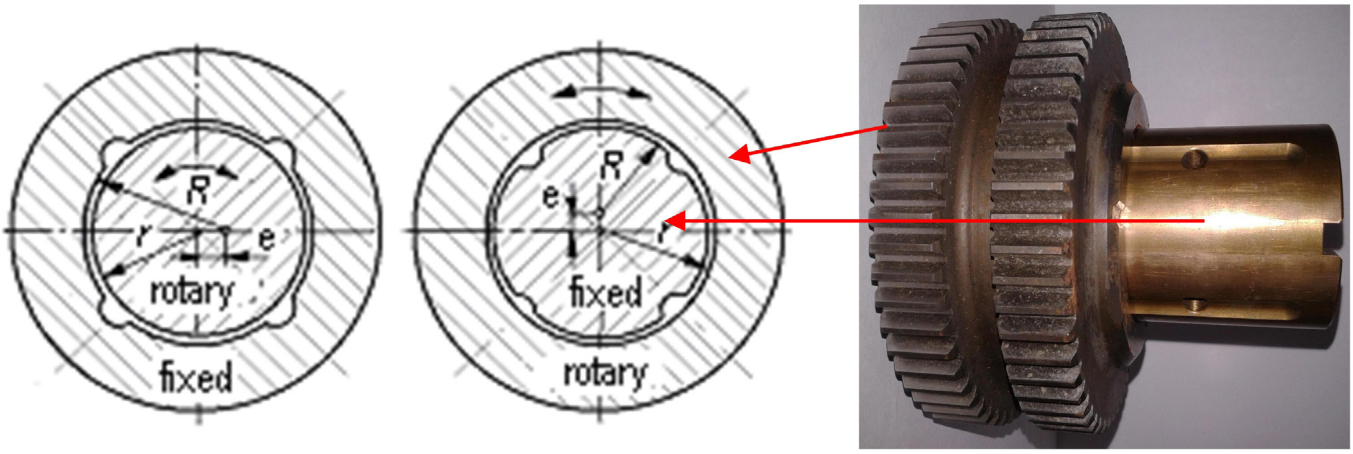


Fig. 9. Lay-out of high speed compressor unit driven by electric motor: 1 – rotor with blades disk (18000 rpm), 2 – axial-radial bearing, 3 – satellite bearing (4-lobe fixed journal), 4 – radial bearing (4-lobe sleeve), 5 – toothed ring, 6 – satellite, 7 – toothed coupling, 8 – toothed disk, 9 – housing, 10 – electric motor



a) b) c)

Fig. 10.: 4-lobe journal bearings; a) rotating journal and fixed sleeve, b) fixed journal (4-lobe) and rotating sleeve, c) design of fixed journal and rotating sleeve; e – eccentricity [3]

Multilobe bearings geometry

Classic multilobe journal bearings are composed of single circular sections whose centres of curvature are not in the geometric centre of the bearing. The geometric configuration of the bearing as a whole is discontinuous and not circular. The multilobe pericycloid journal bearings ("wave bearing" [3, 4]) are characterised by continuous profile and multi hydrodynamic oil films on the journal perimeter [6]. Pericycloid is a continuous curve that is a trajectory of plane point of circle undergoing pure rolling with internal curvature on a fixed circle. Continuous curvature of the operating surface is an important feature of the pericycloid bearing.

The geometry of the oil film gap of multilobe journal bearings (Fig. 11) describes Eqn. (1) [1, 3].

$$\overline{H}(\varphi) = \overline{H}_c + \overline{H}_L, p(\varphi) \tag{1}$$

The first term of right side of Eqn. (1) giving the oil gap thickness for eccentric orientation of journal in the bearing bush has the form:

$$\overline{H}_c = 1 - \varepsilon \cdot \cos(\varphi - \alpha) \tag{2}$$

where: ε – relative eccentricity, α – attitude angle of centres line.

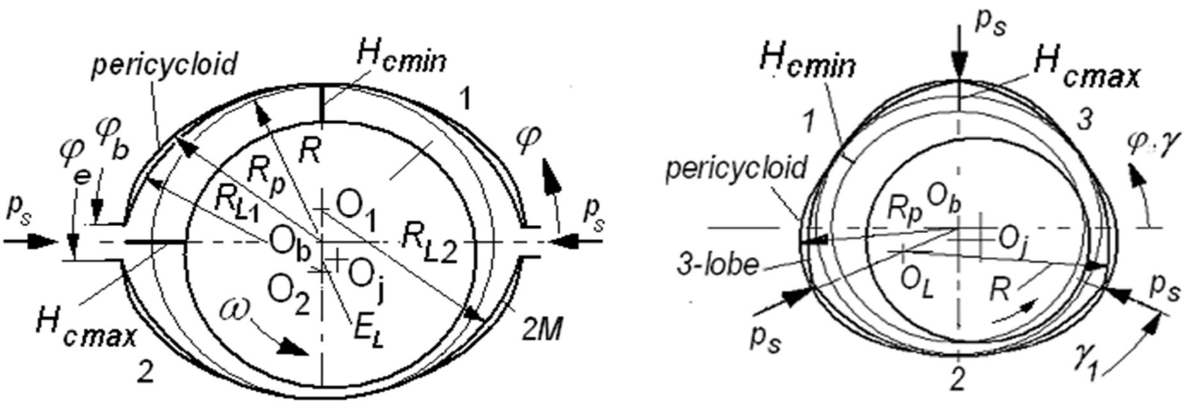
The geometry of the oil film thickness of multilobe and pericycloid [3] bearings at concentric orientation of journal and bearing axis describe Eqn. (3) and Eqn. (4), respectively.

$$\overline{H}_L(\varphi) = \psi_s + (\psi_s - 1) \cdot \cos(\varphi - \gamma) \tag{3}$$

where: ψ_s – lobe relative clearance.

$$\overline{H}_P(\varphi) = \lambda^* (1 + \cos n_p \varphi) \tag{4}$$

where: λ^* – pericycloid relative eccentricity, η_p – multiply of pericycloid.



a) b)

Fig. 11. Geometry of 2- and 3-lobe journal bearings; a) 2P and 2M, $O_{1,2}$, RL_1, RL_2 – center and radius of lobe 1 and 2, respectively, b) 3P and 3M; O_b, O_L, O_j – centre of sleeve, lobe and journal, R_p – radius of pericycloid, 1,2,3 – number of lobe, p_s – oil supply pressure, H_{cmin}, H_{cmax} – minimum and maximum height of lubricating gap at concentric position of journal and sleeve; numbers 1 through 3 denote sliding surfaces; p_s – oil supply pressure

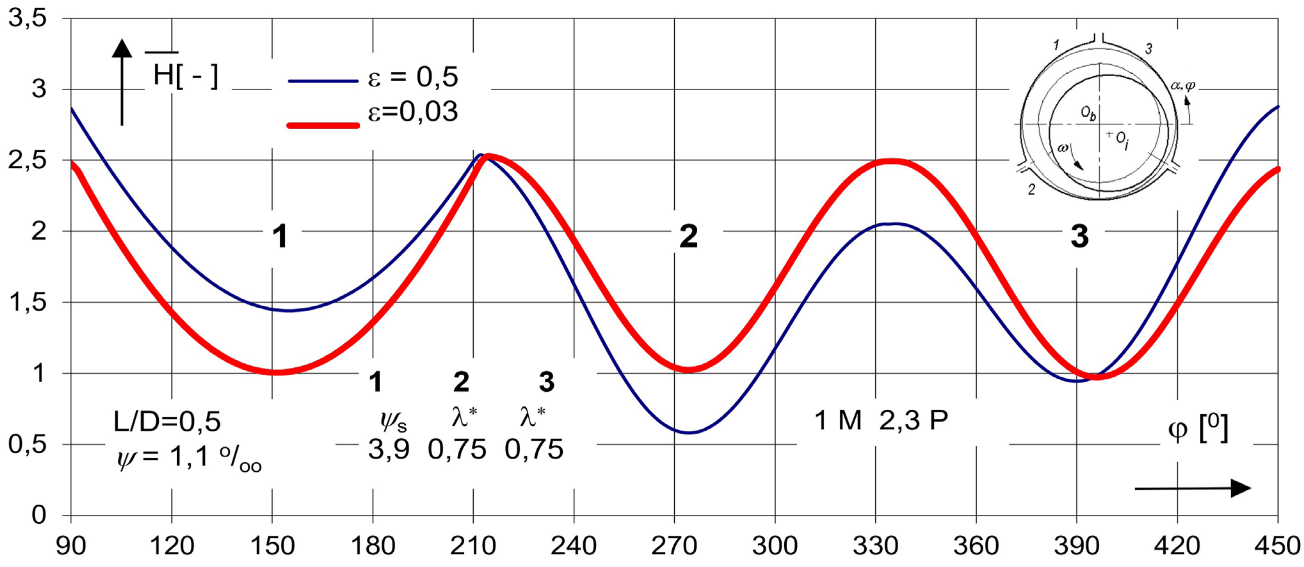


Fig. 12. Dimensionless oil film thickness of the 3-lobe journal bearing with the lobes of different geometry; 1 – multilobe M, 2, 3 - P – pericycloid [3]

For the purpose of comparison of the characteristics of multilobe classic and pericycloid journal bearings their geometry should be identical in the characteristic points of considered profiles. In these characteristic points of profile and at the concentric position of journal in the sleeve, the dimensions of oil gap are equal. Exemplary, for the 3-lobe bearing the dimension of oil gap for the angles φ = 30⁰, 90⁰, 120⁰, 210⁰, 270⁰ and 330⁰ has the same value (Fig. 11b).

Minimum height H_{cmin} of lubricating gap at concentric position of journal in the sleeve occurs at the determined angles of peripheral co-ordinate φ (exemplary, for 3-lobe bearing Fig. 11b it is each 120⁰).

Dimensionless oil film thickness of the 3-lobe journal bearings with the lobes of different geometry present Fig. 12 (relative eccentricity ε =0,03 means that the bore and journal centres are very close) [3].

Pressure and temperature distributions in oil film

The flow of mass and energy in a lubricating film govern three basic laws: conservation of mass, maintaining the quantity of motion and the energy conservation. Applying these laws, the Navier-Stokes and geometry of oil film equation allows obtaining the Reynolds, energy, viscosity equations [3, 8] which numerical solutions gives the static and dynamic characteristics of journal bearing for adiabatic or diathermal model of oil film.

The oil film pressure distribution was defined from Eqn. (5).

$$\frac{\partial}{\partial\varphi}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\,\overline{p}}{\partial\,\varphi}\right)+\left(\frac{D}{L}\right)^2\frac{\partial}{\partial\,\overline{z}}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\,\overline{p}}{\partial\,\overline{z}}\right)=6\frac{\partial\,\overline{H}}{\partial\,\varphi}+\frac{12}{\omega}\frac{\partial\,\overline{H}}{\partial\,\phi} \qquad (5)$$

where: $H = h/(R\text{-}r)$ – dimensionless oil film thickness, h – oil film thickness (μm), $\overline{p}$ - dimensionless oil film pressure, $\overline{p}=p\psi^2/(\eta\omega)$, p – oil film pressure (MPa), r – journal radius (m), L – bearing length (m), D, L – sleeve diameter and length (m), R, r – sleeve and journal radius (m), $\overline{z}$ – dimensionless axial co-ordinate, $\phi=\omega\, t$ – dimensionless time $\overline{\eta}$ – dimensionless viscosity.

It has been assumed for the pressure region that the oil is supplied under pressure into the groove, on the bearing edges the oil film pressure p(φ, z) = 0 and in the regions of negative pressure, p(φ, z) = 0.

The oil film pressure distribution computed from Eqn. (4) has been introduced in the transformed energy Eqn. (6) for obtaining the temperature field [3, 9, 10].

$$\frac{\overline{H}}{Pe}\left[\frac{\partial^2\overline{T}}{\partial\varphi^2}+\left(\frac{D}{L}\right)^2\frac{\partial^2\overline{T}}{\partial\overline{z}^2}\right]+\left[\frac{\overline{H}^3}{12\overline{\eta}}\frac{\partial\,\overline{p}}{\partial\,\varphi}-\frac{\overline{H}}{2}\right]\frac{\partial\,\overline{T}}{\partial\,\varphi}+\left(\frac{D}{L}\right)^2\frac{\overline{H}^3}{12\overline{\eta}}\frac{\partial\,\overline{p}}{\partial\,\overline{z}}\frac{\partial\,\overline{T}}{\partial\,\overline{z}}=-\frac{\overline{H}^3}{12\overline{\eta}}\left[\left(\frac{\partial\,\overline{p}}{\partial\,\varphi}\right)^2+\left(\frac{D}{L}\right)^2\left(\frac{\partial\,\overline{p}}{\partial\,\overline{z}}\right)^2\right]-\frac{\overline{\eta}}{\overline{H}} \qquad (6)$$

where: $\overline{T}$ - dimensionless oil film temperature, P_e – Peclet number.

The boundary conditions for the oil film pressure and temperature take into account the inlet pressure and temperature. Temperature values $T(\varphi,\overline{z})$ on the boundaries ($\overline{z}$= ±1) have been determined by means of the parabolic approximation (9). Temperature and viscosity distribution were found by the iterative solution of equations (4), (5) and (6). The viscosity can be described by exponential equation [3,7].

Oil film pressure and temperature distributions in the 3-lobe journal bearing with different geometry of lobes show Fig. 13 and Fig. 14 respectively. The distributions of oil film pressure and temperature in 3-lobe bearing with the lobes of identical geometry shows Fig. 15 (computed on the assumption of diathermal oil film) [3].

Figure. 16 shows the journal relative eccentricity (a) and static equilibrium position angles (b) of two types of 3-lobe journal bearings versus Sommerfeld number determined on the assumption of turbulent oil film (3M – multilobe, 3P – pericycloid

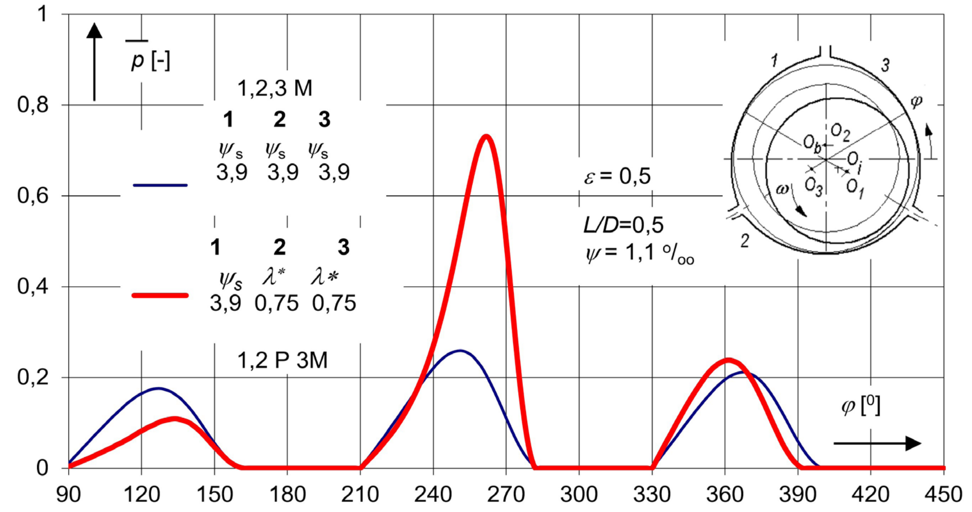


Fig. 13. Oil film pressure distributions in 3-lobe bearings with the lobes of identical and different geometry; the lines denote: thick line– classic multilobe, thin –1st classic profile and 2nd and 3rd lobe of pericycloid profile

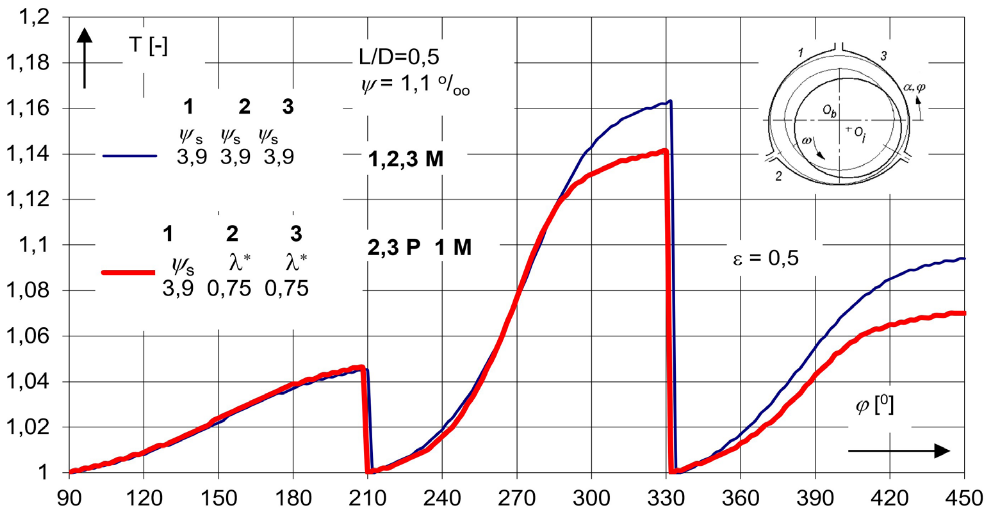


Fig. 14. Oil film temperature distributions in 3-lobe bearings with the lobes of identical and different geometry; the lines denote: thin line– classic multilobe, thick –1st classic profile and 2nd and 3rd lobe of pericycloid

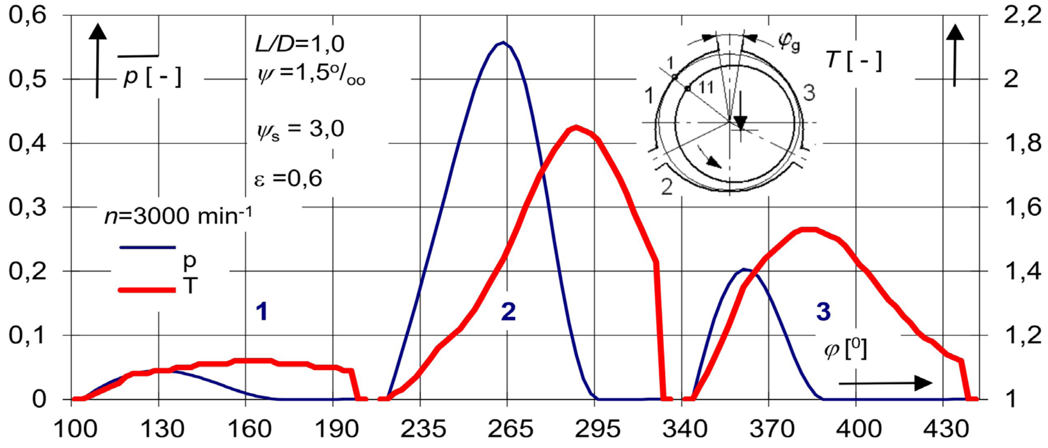
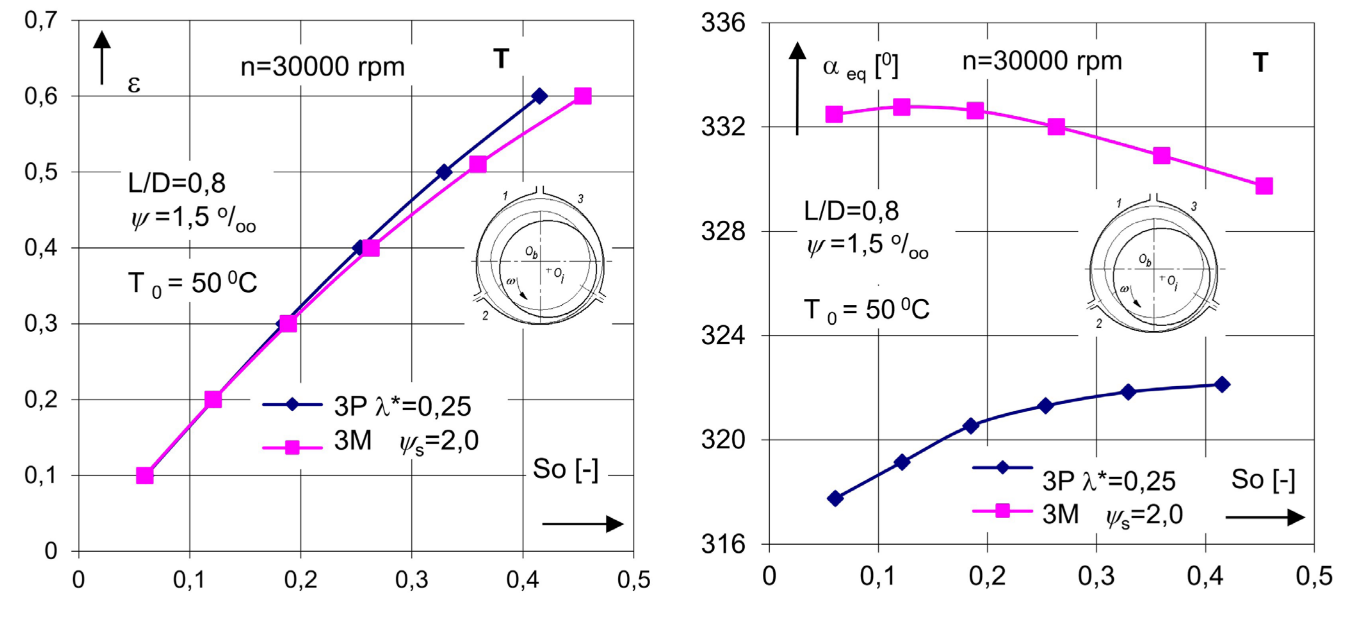
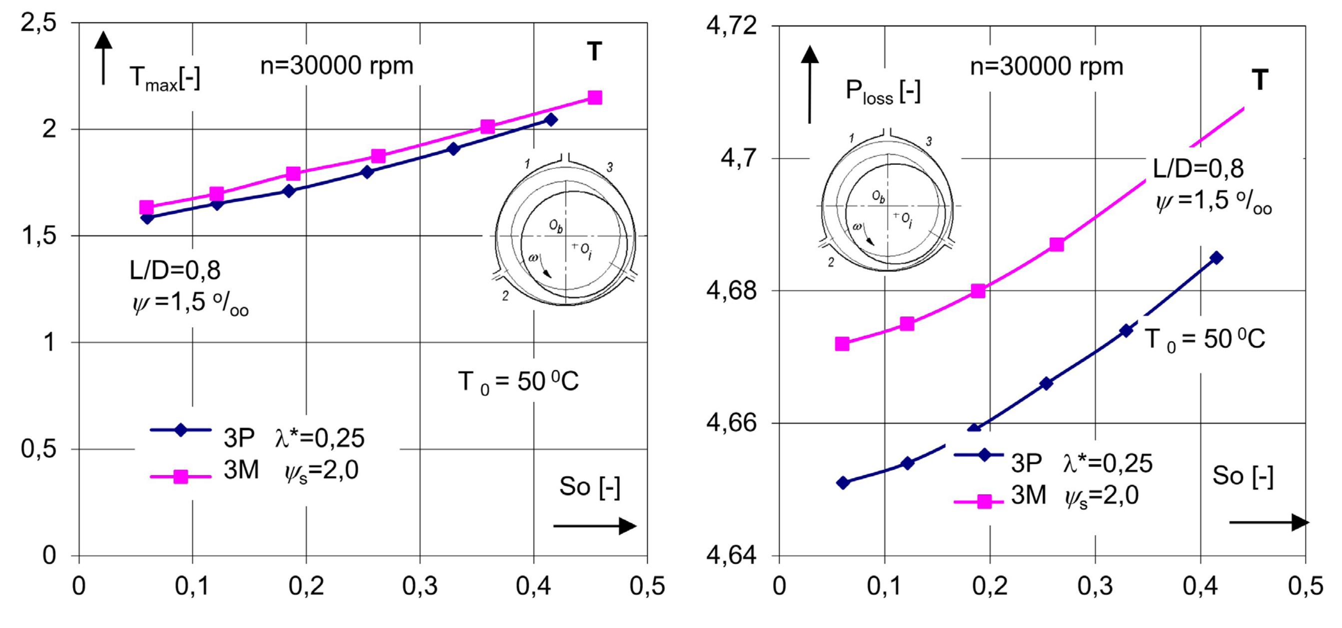


Fig. 15. Oil film pressure and temperature distributions in the 3-lobe bearings with the lobes of identical geometry determined for diathermal oil film [3]



a) b)

Figure. 16. Journal relative eccentricity (a) and static equilibrium position angles (b) of two type of 3-lobe journal bearings versus Sommerfeld number determined on the assumption of turbulent oil film (3M – multilobe, 3P – pericycloid profile)



a) b)

Fig. 17. Maximum oil film temperature (a) and power loss (b) of two type of 3-lobe journal bearings versus Sommerfeld number determined on the assumption of turbulent oil film (3M – multilobe, 3P – pericycloid profile).

profile). In case of journal displacements there is small difference in the bearing load capacity, which is determined by the Sommerfeld number (e.g. Fig. 16a, at ε = 0,3 the load capacity for both considered profiles of bearings is almost equal). Larger difference occurs in case of static equilibrium position angles and it is pointed in Fig. 16b; larger values of static equilibrium position angles are for the multilobe journal bearing (e.g. at Sommerfeld number equal to 0,3 α_{eq}= 322^0 for 3P bearing but for 3M bearing this angle is almost 332^0).

Maximum oil film temperature and power loss of two types of 3-lobe journal bearings (3M – multilobe, 3P – pericycloid profile) versus Sommerfeld number determined on the assumption of turbulent oil film can be observed in Fig. 17. Maximum oil film temperatures are slightly higher for classic 3M bearing (Fig. 17a).

Calculated power losses are higher in case of 3M bearing (Fig. 17b, e.g. at S_0=0,4 the dimensionless power loss is about 4,68 for 3P bearing but 4,71 for 3M bearing).

Final remarks

One of the most important issues in the design and operation of machines is the complex of activities aimed at improving the quality of machines or devices; there is the development of the methods that control their durability and reliability features. With it in mind and with regard to the journal bearings, the modern approach requires in various phases of bearing life, any measures to ensure the proper durability and operational reliability of the bearings system.

- Designs of multilobe bearings have been presented with lobes of different geometry.
- New design solutions of multilobe bearings, including multilobe floating ring bearings, should allow higher rotational speeds of rotating machines with more favourable temperature conditions of the lubricating film and better operating stability.

The computer simulation method and the numerical algorithm adopted ensure multilobe bearings calculations for different sets of input data. The results can be used in the process of the design of cylindrical and multilobe journal bearings with a discontinuous and continuous profile of the sleeve bore, or the bearings with the lobes of different geometry that are subjected to static and dynamic loads.

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