MAXIMUM OIL FILM TEMPERATURE OF 8-LOBE JOURNAL BEARINGS

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Multilobe journal bearings of steam turbines have well-known thermal problems. The paper presents the results of the calculations of maximum oil film temperature of 8-lobe journal bearings. Generally, there is small effect of the bearing relative clearance and lobe relative clearance on the displacement of journal and static equilibrium position angles.

The orientation of load with regard to the bottom lobe (lobes) of multilobe bearings affects the maximum oil film temperatures of oil film. Also, the bearing geometry parameters such as the bearing relative clearance, lobe relative clearance cause the variation in the maximum oil film temperature.

It was stated in this investigation that an increase in the lobe relative clearance and journal rotational speed causes the decrease in the maximum oil film temperature.

Keywords: oil film temperature, journal bearing.

1. Introduction

High-speed rotating machinery, such as steam turbines, turbo-compressors, pumps, turbo pumps, accelerating gearboxes contain multilobe journal bearings. They include the bearings with two or more lobes on the bearing peripheral. The reason for the application of such bearings is that each of these types has specific static and dynamic characteristics. The number of lobes is usually, depending on the industrial application from 2 through 4. The range of peripheral velocities for multilobe bearings is from 30 to 100 m/sec [1–2].

The bearings with 4, 6 or 8-lobes (Fig. 1) are applied in the turbo-compressors and the spindle bearings systems of grinding machines [3–7]. In case of 6 or 8-lobe bearings there is a lack of information on their static and dynamic characteristics. The acquaintance of both types of characteristics for these bearings, operating at the aligned conditions of journal and bush axis, allows choosing the correct design and operating parameters of the bearing.

Solution of the basic equations of thermo-hydrodynamic theory of lubrication allows receiving the necessary data on the pressure, temperature distributions, the maximum value of pressure and temperature of oil film [1, 8], minimum oil film thickness, oil flow and friction forces, that means the static characteristics determining the input variables for the design of tribosystem. The knowledge of the static characteristics of journal bearing allows determining the dynamic characteristics.

The tribosystem of journal bearing consists of journal and bearing sleeve that are separated by the layer of proper viscosity lubricant. The heat generated in the layer of lubricant determines the thermal behaviour of such tribosystem and has a decisive effect on its reliable operation. In case of improper choice of bearing geometrical and operational parameters the thermal problems can arise.

Thermal problems in journal bearings are the result of excessive oil film temperature; thermal gradients generating the bearing distortion, heat flow to and from bearing [9]. The clarification of such problems in high speed multilobe journal bearings tribosystem and their effect on the bearing operation is very important from the point of reliable and durable operation of bearing and bearing systems. An excessive oil film temperature leads to the degradation of lubricant and bearing material. The temperature gradients may cause cracking, while unequal thermal growth of bearing components can cause seizure [1, 9]. In multilobe journal bearings the temperature rise is comparatively high and it causes the thermal problems that in some cases result in the damage of bearing.

The ground for the safe operation of multilobe journal bearings at proper oil film temperature is the full knowledge of bearing thermal performances particularly the oil film temperature distribution and maximum oil film temperature [8, 9]. Such temperature is of great deal in the bearings of the spindle of grinding machines because of very high grinding requirements. The journal bearings of grinding machines are characterised by very small bearing relative clearance. The bearing relative clearances smaller than 1 can generate the thermal problems. In such design case the choice of bearing geometry and operational parameters should base on the analysis of computed static characteristics.

The paper presents the results of the calculations of maximum oil film temperature of 8-lobe journal bearings. Different orientation of lobes with regard to the applied vertical load and different bearing and lobe relative clearances were assumed. The results of calculations are given in form of the diagrams of bearing performances with the emphasis on the maximum oil film temperature. Numerical method by means of finite differences was applied for the simultaneous solution of geometry, Reynolds and energy, viscosity equations on the assumption of diathermal oil film, i.e. the film with the exchange of heat among the bearing system elements.

2. Bearing geometry

The geometry of the oil film gap of multilobe journal bearings (Fig.1) describes Eqn. (1) [1–6]

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

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 following form:

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

where ε – relative eccentricity; φ – peripheral co-ordinate; α – attitude angle of centres line.

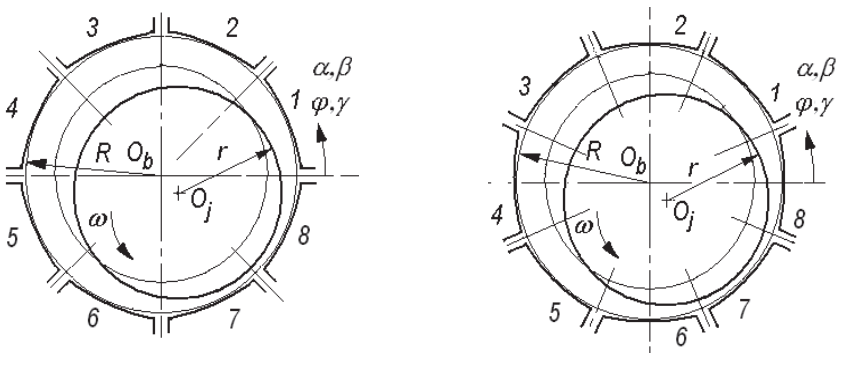

Fig. 1. Geometry of the multilobe journal bearings; a, b – 8-lobe journal bearings with load between lobes (LBL) and load on the lobe (LOL), respectively; O_b, O_j, r, R – centres and radii of the journal and the circle inscribed into the bearing profile lobe, p_i (i=1 through 8) – oil supply pressure

However, the second term determining the geometry of the oil film thickness $\overline{H}_L(\varphi)$ of multilobe profile of bearing at concentric orientation of journal and bearing axis is described by Eqn. (3).

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

where ψ_s – lobe relative clearance; γ – co-ordinate of the segment centre.

3. Oil film pressure and temperature distribution

The journal bearing static characteristics for adiabatic or diathermal model of oil film can be determined by the numerical solution of the oil film geometry, Reynolds, energy and viscosity. The oil film pressure distribution was defined from Eqn. (4) [1–6].

$$\frac{\partial}{\partial \varphi}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial \overline{p}}{\partial \varphi}\right) + \left(\frac{D}{L}\right)^2 \frac{\partial}{\partial \overline{z}}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial \overline{p}}{\partial \overline{z}}\right) = 6\frac{\partial \overline{H}}{\partial \varphi} + 12\frac{\partial \overline{H}}{\partial \phi} \tag{4}$$

where $\overline{H} = h/(R - r)$ – 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 – bush diameter (m), R – bush radius (m), t – time, φ, z – peripheral and axial co-ordinates, $\phi = \omega t$ – dimensionless time, ω – angular velocity, $\overline{\eta}$ – dimensionless viscosity, ψ_s – lobe relative clearance (‰).

Oil film temperature distribution of the diathermal model of oil film was found from the solution of Eqn. (5) and additional equations of heat conduction.

$$\frac{1}{2\bar{\eta}}\frac{\partial \bar{p}}{\partial \varphi}\left(\bar{y}^2-\bar{H}\,\bar{y}\right)+\left(1-\frac{\bar{y}}{\bar{H}}\right)\frac{\partial \bar{T}}{\partial \varphi}=\bar{\eta}K_T\left[\frac{1}{2\bar{\eta}}\frac{\partial \bar{p}}{\partial \varphi}\left(2\bar{y}-\bar{H}\right)-\frac{1}{\bar{H}}\right]+\frac{1}{Pe}\frac{\partial^2 \bar{T}}{\partial \bar{y}^2} \qquad (5)$$

where K_T – thermal coefficient, $K_T=\omega\cdot\eta_0/(c_t\cdot\rho\cdot g\cdot T_0\cdot\psi^2)$, $\bar{T}$ – dimensionless oil film temperature, $\bar{T}=T/T_0$, T – temperature of oil film, (°C), T_0 – temperature of supplied oil, (°C), $\bar{y}$ – dimensionless radial coordinate, Pe – Peclet number, $Pe=\rho c\omega r^2/h$, ρ – oil density, (kg/dm^3), c – specific heat capacity, (J/(K kg).

It has been assumed for the pressure region that on the bearing edges and in the regions of negative pressures the oil film pressure $\bar{p}(\varphi,\bar{z})=0$. The oil film pressure distribution computed from Eqn. (4) was put in the transformed energy equation (5) [8, 9]. Temperature values on the bearing boundaries have been determined by means of parabolic approximation [3, 6]. Exponential equation of viscosity was applied [3].

The developed program of numerical calculation [5–7] solves all above-mentioned equations.

4. Results of calculations

The static characteristics of 8-lobe bearings including the displacement of journal ε, static equilibrium position angle α_{eq}, the maximum oil film temperature $T_{\max}$ were determined for two profiles of 8-lobe journal bearings (Fig. 1). It was assumed that the bearing length to diameter ratio is $L/D=1$, bearing relative clearances $\psi=0.6$ ‰, $\psi=1$ ‰, $\psi=1.2$ ‰. The lobe relative clearances were: $\psi_s=1$ (the case of cylindrical profile), $\psi_s=2$ and $\psi_s=4$. The temperature of supplied oil was 40 °C. The rotational speeds of journal were: 1500 rpm, 2000 rpm, 2500 rpm and 3000 rpm.

The results of calculations are presented in Fig. 2 through Fig. 15.

Oil film temperature distributions on the bearing peripheral that were obtained at assumed relative eccentricity of journal ($\varepsilon=0.4$) for diathermal model of oil film and cylindrical ($\psi_s=1$) as well as multilobe profile ($\psi_s=2$) and the load between lobes, are shown in Fig. 2 and Fig. 3; at considered values of lobe relative clearance there is small difference in obtained temperature distributions. The highest temperatures are generated on the lobe No. 8 (e.g. Fig. 3) and the lowest on the lobe No. 4; lobe No. 1 shows comparatively high temperatures because the oil film transfers the heat from the lobe No. 8, i.e. the lobe showing the highest temperature. The rotational speed of journal was 1500 rpm.

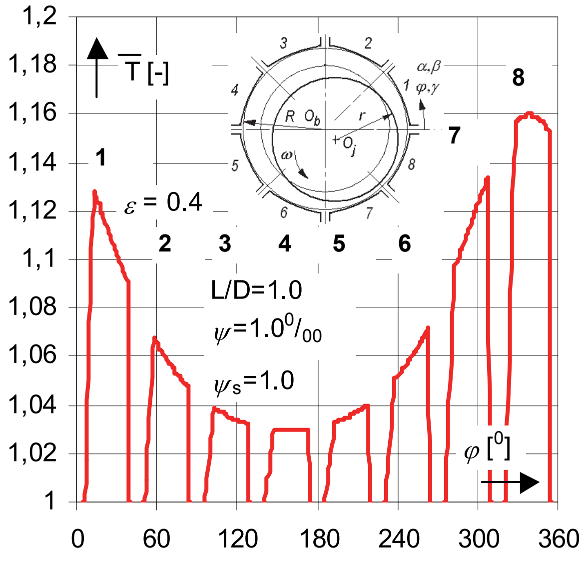

Fig. 2. Oil film temperature distribution in 8-lobe cylindrical journal bearing

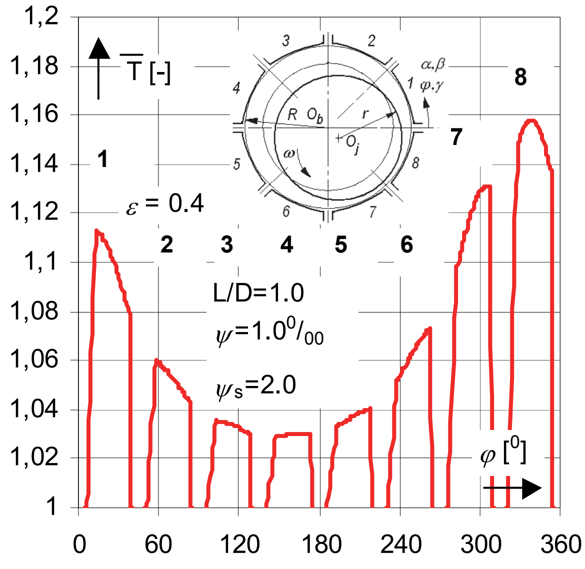

Fig. 3. Oil film temperature distribution in 8-lobe multilobe classic journal bearing

Fig. 4 and Fig. 5 show the oil film pressure and temperature distributions on the lobe No. 7 (most thermally loaded lobe) of 8-lobe bearing at assumed journal relative eccentricity. In case of bearing with the load on the lobe (LOL), the load is directed on the lobe No. 6 (Fig. 1b) but for the bearings with the load between lobes (LBL) the load direction is between the lobes No. 6 and No. 7 (Fig. 1a). The maximum oil film temperature is located over the point of nil pressure value (e.g. Fig. 4, the peripheral coordinate φ is approximately 320°.

For the bearing with the load on the lobe and at assumed lobe relative clearance $\psi_s = 2$ an effect of bearing relative clearance ψ on the journal displacement ε and the static equilibrium position angle α_{eq} is presented in Fig. 6 and Fig. 7, respectively. The relative clearance of bearing has small effect on the journal displacement and static equilibrium position angle.

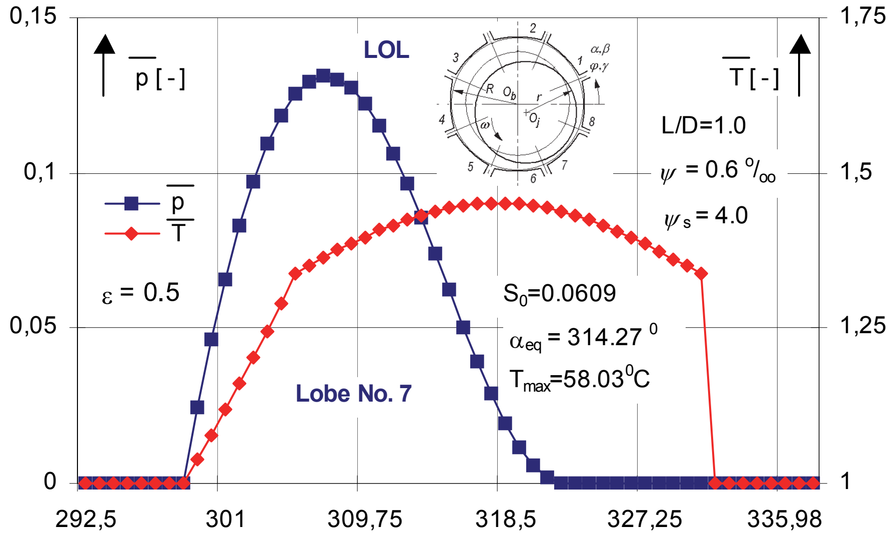

Fig. 4. Oil film pressure and temperature distributions on the lobe No. 7 and lobe No. 8 (most thermally loaded lobe) at assumed journal relative eccentricity; the load on the lobe No. 6

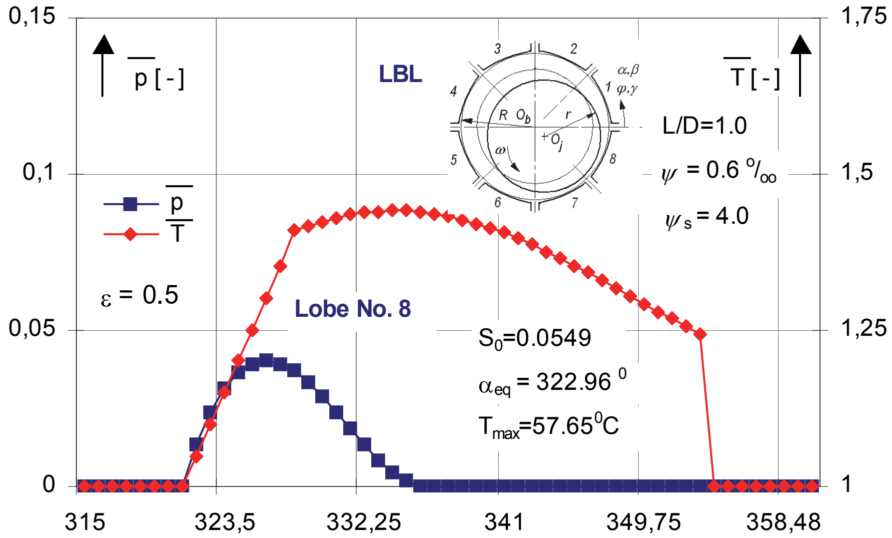

Fig. 5. Oil film pressure and temperature distributions on the lobe No. 8 (most thermally loaded lobe) at assumed journal relative eccentricity; load between lobes No.6 and 7

Fig. 8 and Fig. 9 present the maximum oil film temperature for different values of bearing relative clearance and for the bearing with load on the lobe. An increase in the bearing clearance causes the decrease in the maximum oil film temperature (e.g. Fig. 8 maximum oil film temperature T_{max} at assumed So = 0.1).

Fig. 10 and Fig. 11 show an effect of bearing relative clearance ψ on the journal displacement ε and the static equilibrium position angle α_{eq} at assumed lobe relative clearance $\psi_s = 4$ and for the bearing with the load between the bottom lobes. At assumed journal relative eccentricity the largest load capacity shows the bearing with the smallest relative clearance (e.g. Fig. 10 at $\varepsilon = 0.6$ the Sommerfeld number $S_0 = 0.1$ for $\psi = 0.6$ ‰ and $S_0 = 0.14$ for $\psi = 1.2$ ‰). The relative clearance of bearing has small effect on the journal displacement and static equilibrium position angle.

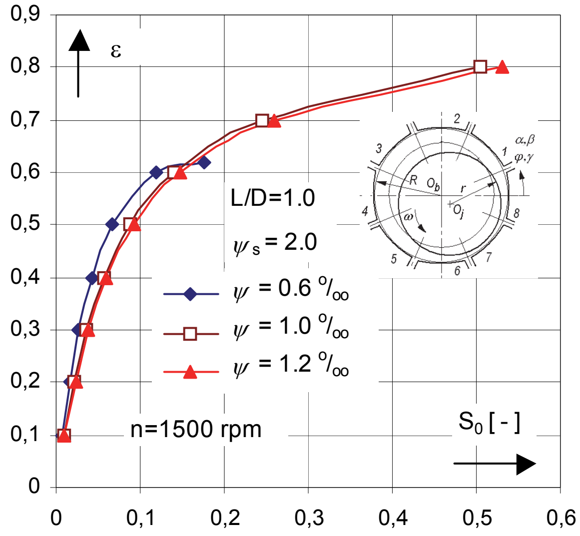

Fig. 6. Effect of bearing relative clearance on the journal displacement

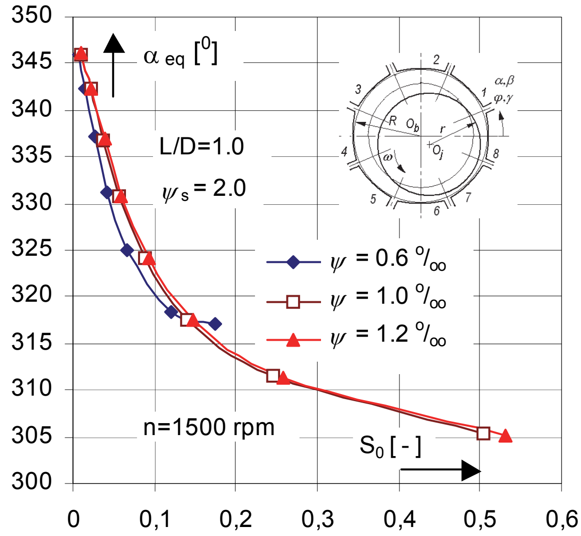

Fig. 7. Effect of bearing relative clearance on the static equilibrium position angle

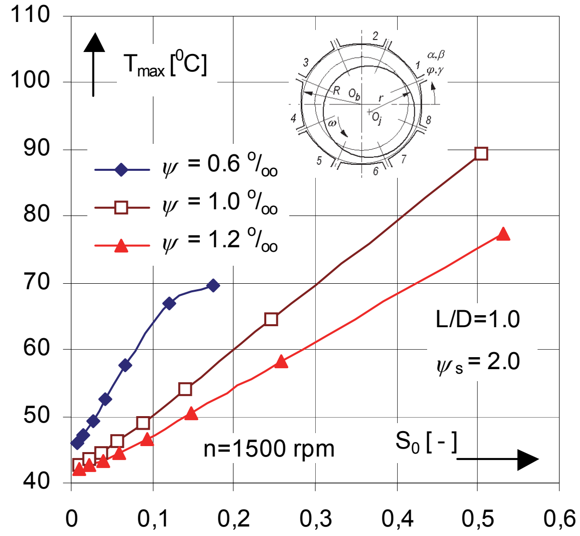

Fig. 8. Maximum oil film temperature versus Sommerfeld number

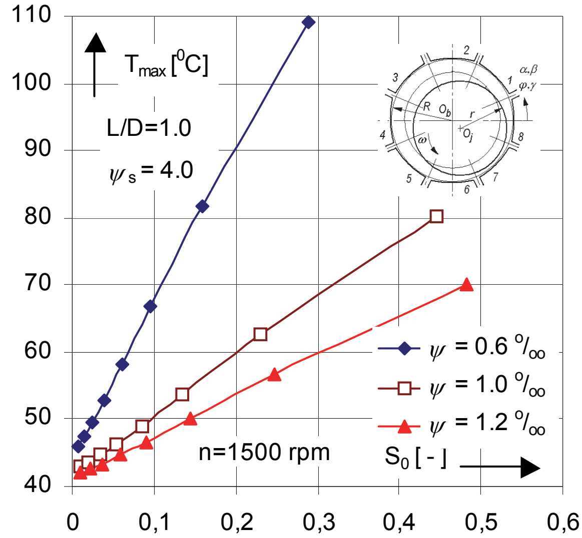

Fig. 9. Maximum oil film temperature versus Sommerfeld number

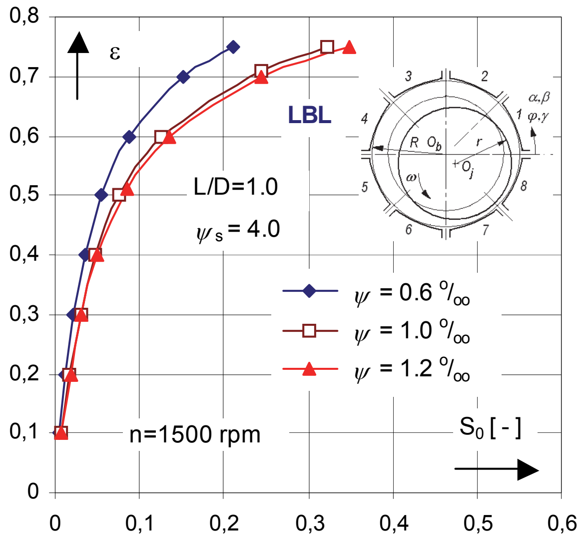

Fig. 10. Effect of bearing relative clearance on the journal displacement

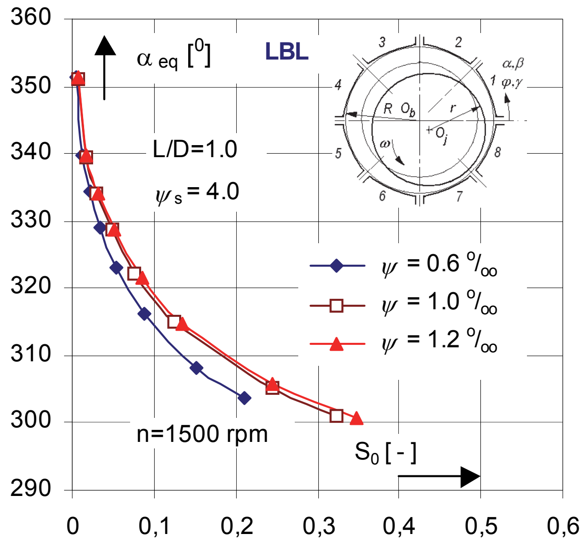

Fig. 11. Effect of bearing relative clearance on the static equilibrium position angle

The rotational speed of journal affects the bearing performances causing the increase in the operating temperatures of oil film. Exemplary results of the effect of rotational speed on the journal displacement, static equilibrium position angle and maximum oil film temperatures are showed in Fig. 12 through Fig. 15.

It results from Fig.12 and Fig. 13 that at assumed bearing geometric parameters there is small effect of rotational speed on the journal displacement and static equilibrium position angle; these results were obtained for the bearing with the load between lobes.

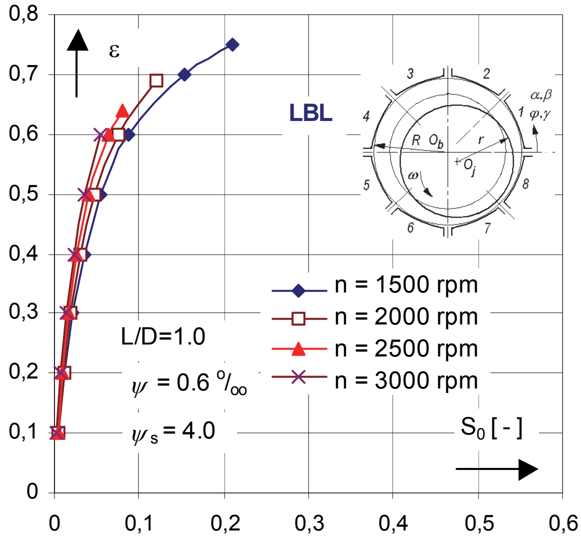

Fig. 12. Effect of bearing relative clearance on the journal displacement

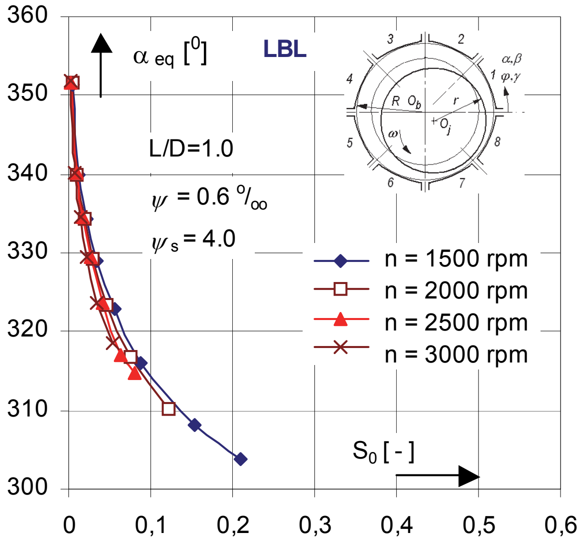

Fig. 13. Effect of bearing relative clearance on the static equilibrium position angle

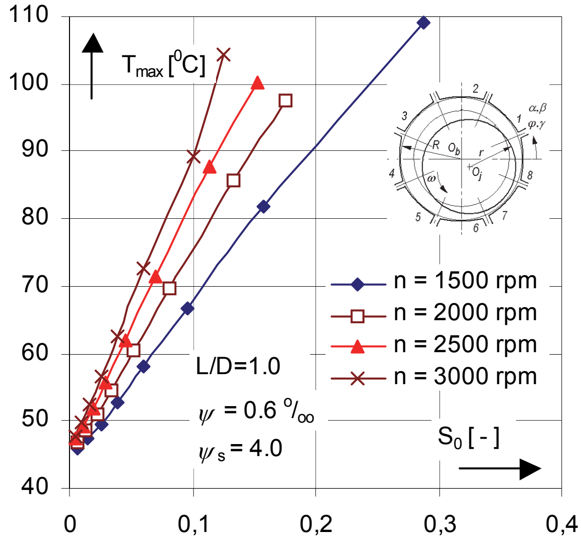

Fig. 14. Effect of rotational speed on the maximum oil film temperature

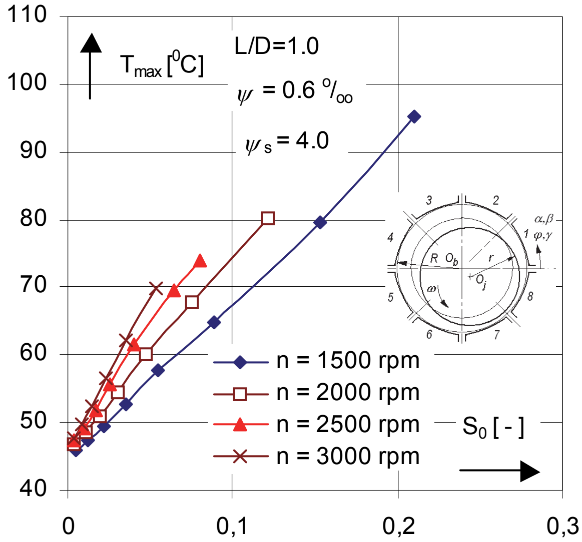

Fig. 15. Effect of rotational speed on the maximum oil film temperature

Effect of rotational speed on the maximum oil film temperature can be observed in Fig. 14 and Fig. 15. An increase in the journal rotational speed causes the increase in the maximum oil film temperature at assumed Sommerfeld number and in both cases of considered 8-lobe journal bearings, i.e. the bearing with the load on the lobe (Fig. 14) and the load between the lobes (Fig. 15). However, in case of bearing with the load between lobes the range of Sommerfeld numbers is smaller than in case of bearing with the load on the lobe.

5. Final remarks

Developed code of the computation and applied method of numerical calculation of static characteristics of 8-lobe journal bearings gives the grounds for the evaluation of the thermal state of multilobe journal bearings.

Generally, there is small effect of the bearing relative clearance and lobe relative clearance on the displacement of journal and static equilibrium position angles.

The orientation of load with regard to the bottom lobe (lobes) of multilobe bearings affects the maximum oil film temperatures of oil film. Also, the bearing geometry parameters such as the bearing relative clearance, lobe relative clearance cause the variation in the maximum oil film temperature.

It was stated in this investigation that an increase in the lobe relative clearance and journal rotational speed causes the decrease in the maximum oil film temperature.

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ОПРЕДЕЛЕНИЕ МАКСИМАЛЬНОЙ ТЕМПЕРАТУРЫ СМАЗОЧНОГО СЛОЯ В 8-ЛЕПЕСТКОВОМ ПОДШИПНИКЕ СКОЛЬЖЕНИЯ

С. Стрелецки, Г. Капуста

Многоклиновые подшипники скольжения паровых турбин имеют общеизвестные тепловые проблемы. В данной работе представлены результаты вычислений максимальной температуры масляной пленки подшипников скольжения с восемью выступами. В целом обнаружен небольшой эффект влияния люфтов на смещение вала и на статические углы.

Ориентация нагрузки относительно нижнего выступа (выступов) подшипников влияет на максимальные температуры масляной пленки. В то же время, геометрические параметры подшипника, такие как люфты, влияют на максимальную температуру масляной пленки.

В ходе исследования установлено, что увеличение люфта и скорости вращения вала вызывает уменьшение максимальной температуры в масляной пленке.

Ключевые слова: температура смазки, подшипник скольжения.

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