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NUMERICAL ANALYSIS OF INFLUENCE OF BEARING MATERIAL THERMAL CONDUCTIVITY COEFFICIENT ON HYDRODYNAMIC LUBRICATION OF A CONICAL SLIDE BEARING

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Abstract

One of the main parameters affecting the hydrodynamic lubrication of slide bearings is the viscosity of lubricating oil. Many studies show, that significant changes in the viscosity of oil occur along with changes in its temperature. The influence on the temperature distribution in the lubrication gap of the slide bearing have a variety of factors, and one of them is the amount of heat exchanged between the lubricant and the environment. The temperature of the lubricating oil of operating bearing is usually higher than the ambient temperature. In addition to the convection, which occurs during the flow (heat exchange related to the oil supply and discharge system) some amount of heat is transferred to the bearing sleeve material (and also to the bearing shaft), and then it is conducted to sleeve outer surface. The amount of heat transferred through the bearing sleeve is mainly dependent on the difference of temperatures between inner and outer sleeve surfaces and also depend on the heat conduction coefficient of sleeve material. This article presents the results of modelling of the influence of amount of heat conducted through the bearing material, on the hydrodynamic lubrication of a conical slide bearing. The study concerned various values of the heat conduction coefficient of the bearing material to investigate its influence on the temperature values of lubricating oil, and thus, on its viscosity, on the distribution of hydrodynamic pressure and on the calculated values of bearing load carrying capacities and friction forces.

Keywords: *slide bearing, hydrodynamic lubrication, conical bearing, heat conduction, pressure distribution*

1. Introduction

The viscosity of lubricating oil is a significant parameter affecting the hydrodynamic lubrication of the slide bearing and greatly depends on the temperature [5]. The temperature distribution in the lubricating wedge of a bearing depends on many factors, such as the temperature of the supplying oil, the ambient temperature, the amount of heat generated due to viscous heating [4]. A part of heat generated by internal friction in the oil is transferred to the bearing sleeve. The amount of heat transferred to the environment through the sleeve depends on its thermal conductivity. This work concerns the numerical analysis of the influence of the value of heat conduction coefficient of the bearing sleeve material on the hydrodynamic lubrication of conical slide bearing.

The appropriate material for the construction of the slide-bearing shaft is e.g. steel, due to its rigidity and strength. The bearing sleeve material must meet additional properties, e.g. low coefficient of friction, low thermal expansion or high value of the heat conduction coefficient to efficiently dissipate heat to the environment. In order to provide desired parameters, also the two- and multilayer bearings are designed. The most commonly used materials in slide bearings [3] are alloys of tin, lead, aluminium, bronzes and also polymers, characterized by relatively low thermal conductivity.

In the performed simulations, it was assumed that the sleeve is a homogeneous and isotropic shell with a constant thickness. The calculations were made for values of heat conduction coefficients in the range from $\kappa = 10$ [W/(m·K)] (similar values for stainless steel and some bronze alloys as well as lead alloys) $\kappa = 200$ [W/(m·K)] (similar to pure aluminum). Despite the adopted simplifying assumptions, it was shown, that in some cases, the thermal conductivity of the pan material, i.e. the amount of dissipated heat, has a significant impact on the operating parameters of the conical hydrodynamic slide bearing.

2. Bearing model

The bearing, which operates in a steady state, was considered in this research. It was assumed, that the bearing surfaces are smooth, rigid and without deformations. The bearing sleeve is stationary and the hydrodynamic pressure is generated due to the rotation of the bearing shaft. In Fig. 1 is shown the geometry of the investigated slide conical bearing (on the left-hand side – the radial cross-section in the location of the smallest radius of the shaft, on the right-hand side – the axial cross-section).

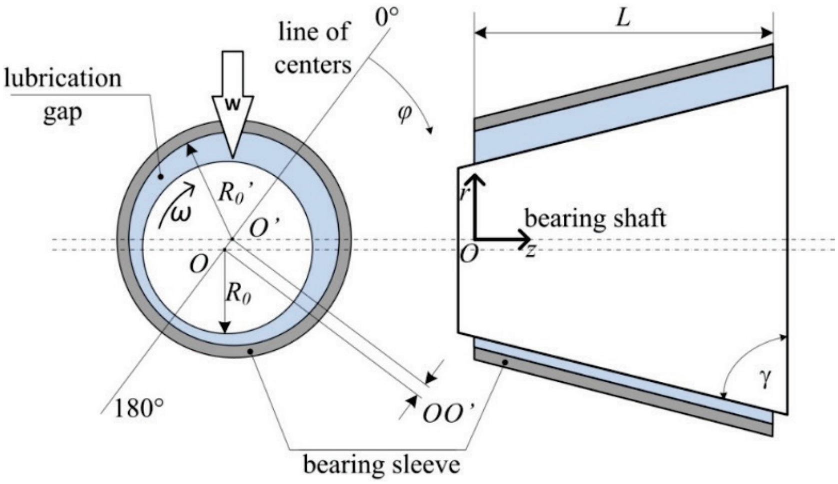

Fig. 1. The geometry of concerned slide conical bearing – radial and axial cross-section

The bearing has length of $L = 50$ [mm] (measured along the axis of the shaft). The smaller base of the shaft cone (frustum cone) has radius of $R_0 = 50$ [mm]. The radial clearance, defined as $\varepsilon = R_0' - R_0$, where R_0' is the radius of the sleeve (measured from the sleeve axis to its inner surface) is $\varepsilon = 0.025$ [mm]. The angle between the base of the cone and its generatix (both of the shaft and sleeve) is equal to $\gamma = 60°$. The shaft axis is parallel to the axis of the bearing sleeve. The relative eccentricity λ is defined as:

$$\lambda = \frac{OO'}{\varepsilon}, \tag{1}$$

where OO' is the absolute value of the eccentricity, which appears due to the load W.

The incompressible, laminar, viscous (non-Newtonian) and non-isothermal flow of lubricating oil was concerned in this study. The assumed physical properties of lubricating oil are:

– density: 850 [kg/m^3],
– specific heat: 1006 [J/(kg·K)],
– heat conduction coefficient 0.025: [W/(m·K)].

The changes in oil viscosity η [Pa·s], depending on the shear rate θ [1/s] and temperature T [K] values, have been included in the calculations by application of a function:

$$\eta(\theta, T) = \eta_s(\theta) \cdot H(T), \tag{2}$$

where η_s (θ) [Pa·s] is a part, describing changes of viscosity values due to spatial variations of shear rate, while $H(T)$ [–] introduces the dependence of viscosity on temperature of lubricating oil.

The Ostwald-de Waele [6-8] model was adopted in this investigation, thus:

$$\eta_s(\theta) = K \cdot \theta^{n-1}. \tag{3}$$

where K [Pa·s^n] is the flow consistency index and n [–] is the flow behaviour (power-law) index. The function $H(T)$, that determines the effect of temperature, is in the form:

$$H(T) = \exp\left[\alpha_T \cdot \left(\frac{1}{T} - \frac{1}{T_\alpha}\right)\right], \tag{4}$$

where $\alpha_T = E_a/R$ is the ratio of the activation energy E_a [J/kmol] to the thermodynamic constant R = 8314 J/(kmol·K) and T_α [K] is a reference temperature for which $H(T)$ = 1.

The no-slip condition for oil layer at bearing surfaces was adopted. The pressure of oil at the boundaries was equal to the ambient pressure p_{amb} = 100 [kPa]. The location of the end of the lubricating wedge in the angle of wrap direction is determined by the Gümbel condition [5]. The supplying oil, the bearing shaft surface and sleeve outer surface, have a constant temperature of 90°C. The heat is generated due to viscous heating in oil.

It was assumed in simulation, that the lubricating oil has properties as Shell Helix Ultra AV-L at a temperature 90°C, which rheological properties were examined and shown in the paper [1]. The results from work [2] were used, where by applying the least squares approximation procedure from the Statsoft Statistica software and fitting the curves described by the equations (3) and (4) to the experimental data presented in paper [9], the values of needed coefficients were determined. The obtained values are as follows: K = 0.01242 [Pa·s^n], n = 0.9792 [–], T_α = 363 [K] and α_T = 3323 [1/K].

The model assumes that the outer surface temperature of the sleeve is constant, and the simulations have been carried out for different values of the heat conduction coefficient of the sleeve material. The sleeve wall thickness is δ = 2.5 [mm], while its material has the density value of ρ = 2719 [kg/m^3] and the specific heat of c_p = 871 [J/(kg·K)].

The calculations were carried out by application of the CFD software – ANSYS Fluent (with the pre- and post-processing tools) from the ANSYS Workbench 2 platform. The pressure based simple algorithm was applied (Green-Gaus node based, second order pressure, the momentum second order upwind, for the energy: Power-law model).

3. Results

The values of load carrying capacities in radial and axial directions, for bearing with λ = 0.5 and n = 1500 rpm, are shown in Fig. 2. In this case, only between κ = 10 [W/(m·K)] and κ = 50 [W/(m·K)] there is a 3.7% difference in the value of the C_t and C_l, whereas the differences, for greater κ, are less than 1%. Fig. 3 shows, how the change in the thermal conduction coefficient of the pan material influences the values of the maximum p_{max} and average p_{avg} hydrodynamic pressure in the lubrication gap of investigated bearing. The comparison of hydrodynamic pressure distributions for κ = 10 [W/(m·K)] and κ = 200 [W/(m·K)], in the cross-section and longitudinal section, at the point where the maximum pressure in the lubrication gap occurs, are shown in Fig. 4.

The results show, that the differences in the load carrying capacities obtained for various kappa, are relatively low, for bearing with λ = 0.5 and n = 1500 rpm.

Figures 5-7 show the results for investigated bearing, when the speed of its shaft was of n = 5000 rpm. The values of C_t and C_l for various κ are shown in Fig. 5, p_{max} and p_{avg} are shown in Fig. 6 and the comparison of hydrodynamic pressure distributions for κ = 10 [W/(m·K)] and κ = 200 [W/(m·K)], in the cross-section and longitudinal section, at the point where the maximum pressure in the lubrication gap occurs, are shown in Fig. 7.

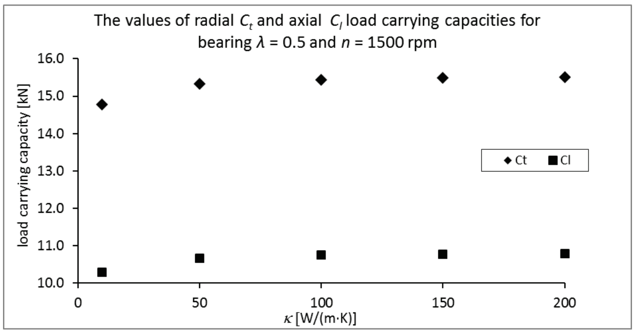

Fig. 2. The influence of the heat conduction coefficient value of the sleeve material on the radial C_t and axial C_l components of load carrying capacity, for the bearing with λ = 0.5 and n = 1500 rpm

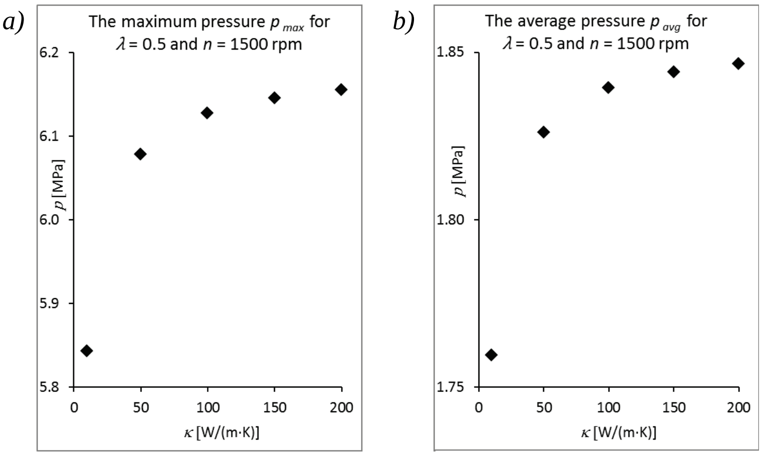

Fig. 3. The values of maximum p_{max} (a) and average p_{avg} (b) oil pressure, for the bearing with λ = 0.5 and n = 1500 rpm

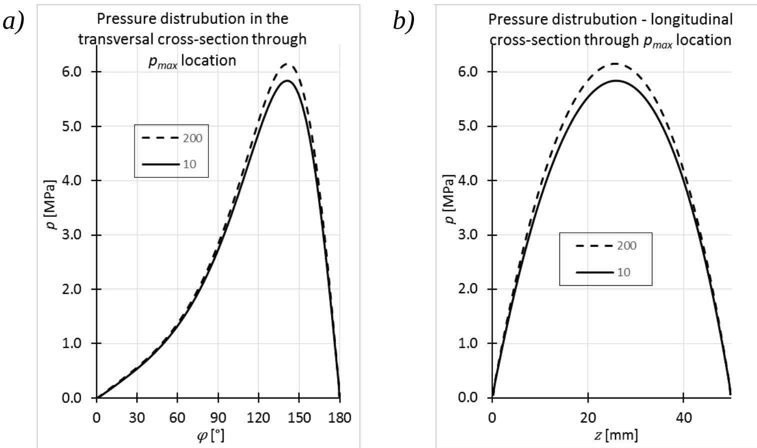

Fig. 4. The comparison of hydrodynamic pressure distributions in the radial (a) and longitudinal (b) cross-section for the bearing with λ = 0.5 and n = 1500 rpm. The results for sleeve material heat conduction coefficient κ = 10 [W/(m·K)] – the solid line, while the results for κ = 200 [W/(m·K)] – the dashed line

At higher rotational speeds, the influence of the sleeve material heat conduction coefficient is essential, because more heat is generated due to viscous heating. The greater κ value facilitates the removal of this heat to the environment and the temperature increments of lubricating oil that cause the decrease in viscosity values, are lower. In this case, by changing the κ value from 10 [W/(m·K)] to 50 [W/(m·K)], the C_t increased by 10.8% and C_l by 8.6%, while further change of κ to 100 [W/(m·K)] gave an increase in the C_t by 2.6% and C_l by 5.6%, relative to κ = 50 [W/(m·K)]. For greater κ the differences were on the order of 1% and lower.

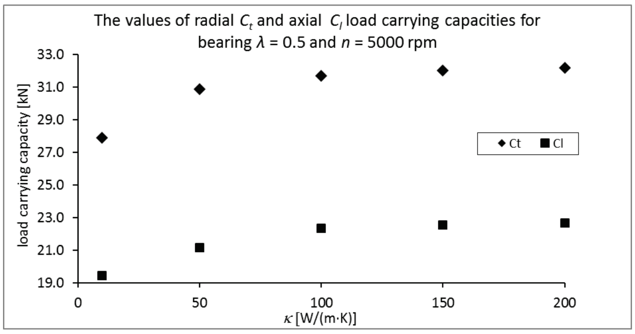

Fig. 5. The influence of the heat conduction coefficient value of the sleeve material on the radial C_t and axial C_l components of load carrying capacity, for the bearing with λ = 0.5 and n = 5000 rpm

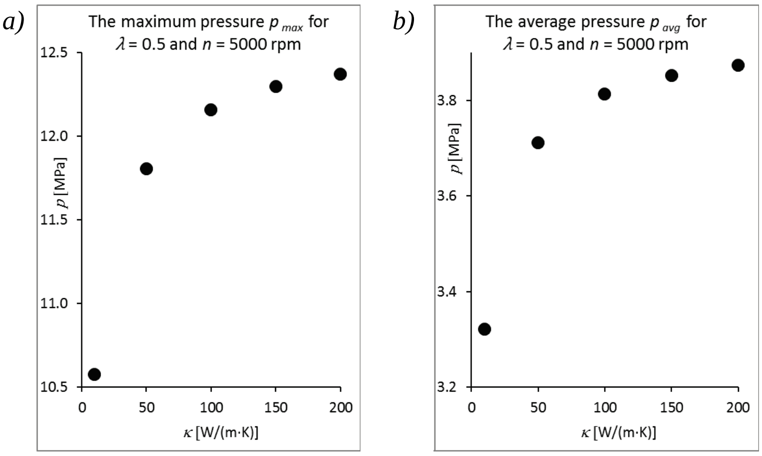

Fig. 6. The values of maximum p_{max} (a) and average p_{avg} (b) oil pressure, for the bearing with λ = 0.5 and n = 5000 rpm

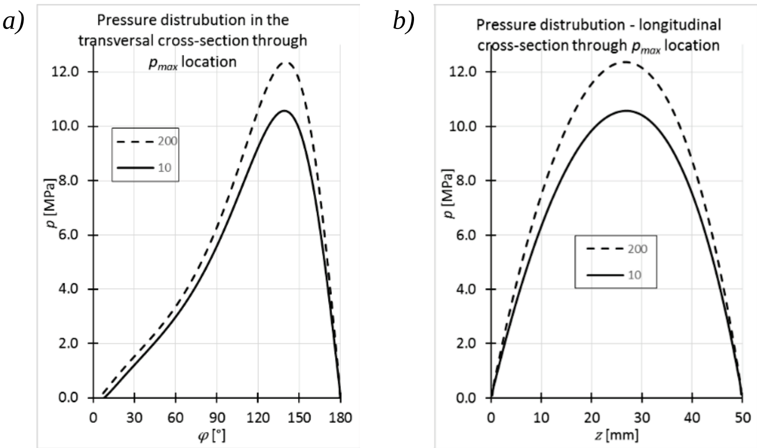

Fig. 7. The comparison of hydrodynamic pressure distributions in the radial (a) and longitudinal (b) cross-section for the bearing with λ = 0.5 and n = 5000 rpm. The results for sleeve material heat conduction coefficient κ = 10 [W/(m·K)] – the solid line, while the results for κ = 200 [W/(m·K)] – the dashed line

Figures 9-11 show the simulation results for a bearing with n = 5000 rpm, but where λ = 0.7. The C_t and C_l depending on κ, are shown in Fig. 8, the values of p_{max} and p_{avg} are shown in Fig. 9, while the comparison of hydrodynamic pressure distributions for κ = 10 [W/(m·K)] and κ = 200 [W/(m·K)], in the cross-section and longitudinal section thorough the p_{max} location, are shown in Fig. 10.

In this case, when κ = 50 [W/(m·K)], there was a 11.1% increase in the C_t value and 11.2% increase in the C_l value, in relation to the bearing, for which the sleeve material has κ = 10 [W/(m·K)].

Increasing the heat conduction coefficient of sleeve material to $\kappa = 100$ [W/(m·K)] resulted in a further increase in C_t by 3.4% and in C_l by 3.3%. At $\kappa = 150$ [W/(m·K)] there was 1.4% gain of C_t and C_l increased by 1.3%, in relation to $\kappa = 100$ [W/(m·K)]. Despite, that the shaft rotational speed is the same, the relative changes in the load carrying capacities, for bearing with $\lambda = 0.7$ differ from those for the bearing with $\lambda = 0.5$. Due to change in lubrication gap height distribution and reduction in the minimum gap height value, there is a significant increase in hydrodynamic pressures, but also the change of the lubricant layer thickness affects the amount of heat generated due to viscous heating and transferred to the bearing surfaces.

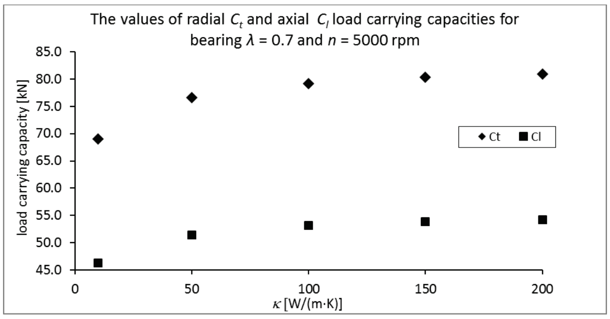

Fig. 8. The influence of the heat conduction coefficient of the sleeve material on the radial C_t and axial C_l components of load carrying capacity, for the bearing with λ = 0.7 and n = 5000 rpm

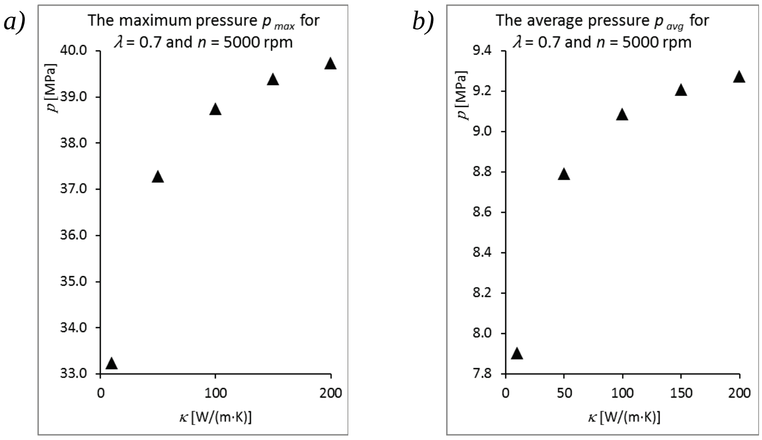

Fig. 9. The values of maximum p_{max} (a) and average p_{avg} (b) oil pressure, for the bearing with λ = 0.7 and n = 5000 rpm

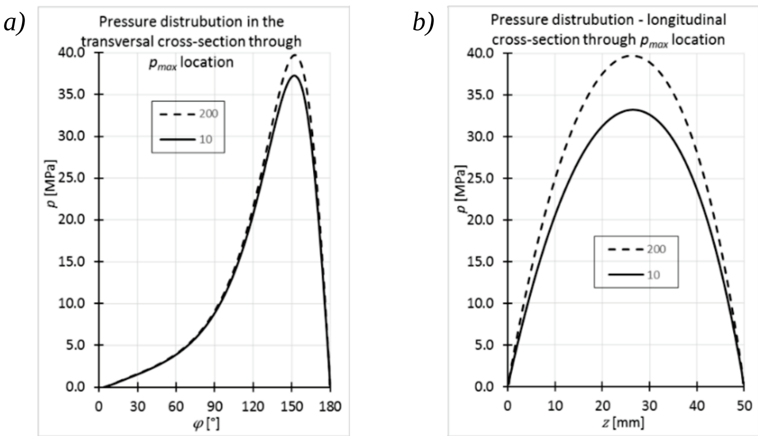

Fig. 10. The comparison of hydrodynamic pressure distributions in the radial (a) and longitudinal (b) cross-section for the bearing with λ = 0.7 and n = 5000 rpm. The results for sleeve material heat conduction coefficient κ = 10 [W/(m·K)] – the solid line, while the results for κ = 200 [W/(m·K)] – the dashed line

For the greater temperatures of the lubricating oil, more heat is transferred to the environment. Part of that heat is conducted through the bearing sleeve. The calculated values of heat dissipated through the considered sleeve (i.e. for sleeve within the range of the wrap angle from 0° to 180°) are shown in Fig. 11.

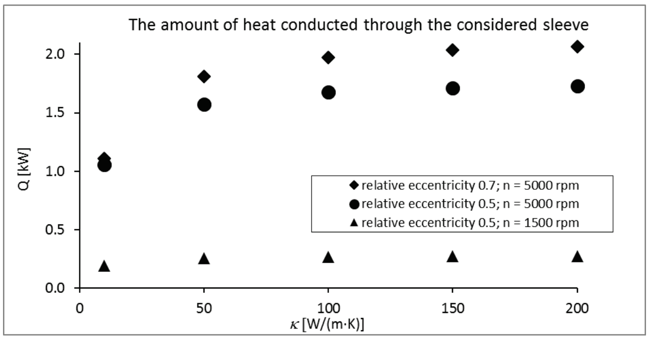

Fig. 11. The amount of heat conducted through the bearing sleeve material for investigated values of its heat conduction coefficient. The calculated values for the half of the bearing sleeve (i.e. from 0° to 180° of wrap angle)

Figure 12 shows the maximum temperature increase in the lubrication gap. In Fig. 13 are shown the values of friction torque at bearing shaft, i.e. the torque needed to sustain the rotation, for the concerned values of sleeve material heat conduction coefficient.

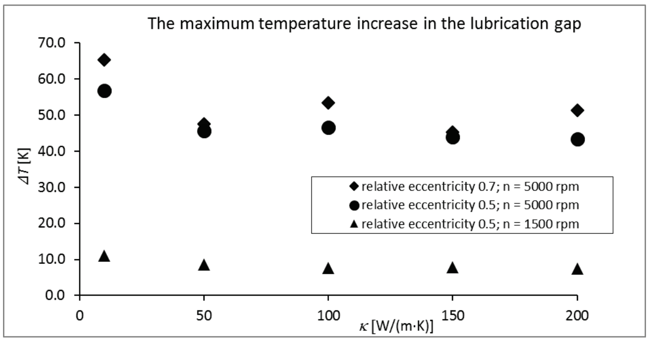

Fig. 12. The calculated maximum temperature increase ΔT in the lubrication gap of the investigated bearing

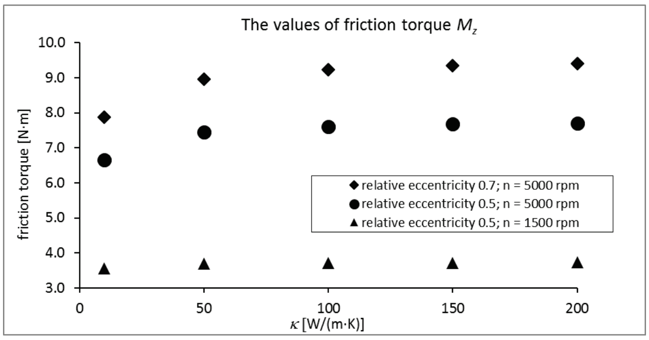

Fig. 13. The influence of the sleeve material heat conduction coefficient on the friction torque at bearing shaft

4. Discussion and conclusions

In this study, the numerical analysis of hydrodynamic lubrication of the conical slide bearing, with different values of heat conduction coefficient of the sleeve material, was carried out. The fixed working conditions were adopted, among others, the constant temperature of the surface of the bearing shaft and the outer surface of the sleeve. Only the value of the sleeve material heat conduction coefficient was changed in the calculations, causing, under certain conditions, the significant changes in the calculated bearing operating parameters, by influencing the amount of heat discharged from the lubrication gap through the sleeve to the environment, and hence, obtaining changes in the values of oil temperature, i.e. the main parameter affecting the lubricating oil viscosity. In particular, this is important for relatively high rotational speeds, when more heat is generated due to viscous heating. It has been shown that the influence of the heat conduction coefficient value of the sleeve material, on the calculated bearing parameters, is nonlinear. Depending on the given conditions, above a certain value, further increase in κ value does not bring any significant changes in bearing parameters.

In further work, it is planned to examine the influence of temperature on the thermal conductivity coefficient of the bearing material and oil, at turbulent flow, to investigate, how it is important in simulations of hydrodynamic lubrication of a conical slide bearing.

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