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CFD ANALYSIS OF INFLUENCE OF AXIAL POSITION OF SHAFT ON HYDRODYNAMIC LUBRICATION OF SLIDE CONICAL BEARING

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Abstract

During the operation of a slide bearing, the position of its shaft or sleeve varies due to many factors, such as vibrations, load changes, changes in the lubricating viscosity. The vibrations or varying load can cause, that the position of the bearing shaft, measured along its axis of rotation, changes. This is particularly important for sliding bearing with conical geometry. Due to the geometry of this kind of bearing, i.e. where the radius of this bearing (of the shaft and sleeve) has not a constant value, as in the case of a journal bearing, it is more difficult to obtain proper values and describe its hydrodynamic lubrication.

This article shows the results of hydrodynamic lubrication of the slide conical bearing, for which the changes in the position of the bearing shaft in the longitudinal direction, i.e. along its axis of rotation, were taken into account. The commercial CFD software, designed for solving general for flow phenomena problems, was used in the simulations. This article shows the results of simulations, assuming that the lubricating oil behaves as a generalized Newtonian fluid. The hydrodynamic pressure distributions, load carrying capacities and friction torques were calculated for the concerned bearing.

The aim of this work is to show how the operating parameters of the slide conical bearing can be influenced, by only changes of the position of the shaft along the axis of its rotation.

Keywords: *hydrodynamic lubrication, conical bearing, slide bearing, radial clearance, CFD*

1. Introduction

During the operation of a slide bearing, the position of its journal (shaft) or sleeve varies due to many factors [2], such as vibrations, load changes, changes in the lubricating oil temperature, and hence, changes in its viscosity. When modelling or designing slide bearing, considering in simulations additional effects (e.g. shaft misalignment, changes of oil viscosity due to temperature, pressure, shear rate, vibrations, roughness of bearing surfaces) can contribute to obtaining more accurate, closer to the measurable, results of pressure distributions, load carrying capacities or friction coefficients. Taking into account the subsequent phenomena and physical characteristics creates the need to use or prepare a more general mathematical model and to perform more complex, numerical calculations. This is particularly important when the hydrodynamic lubrication of a sliding conical bearing is investigated, i.e. the bearing, which geometry allows to carry both, the radial and axial loads (like its equivalent, the conical rolling bearing, tapered roller bearing) and it can be used in back-to-back pairs to support axial forces in either direction. Unfortunately, due to the geometry of this bearing, i.e. the radius of this bearing (of shaft and sleeve) has not a constant value, as in the case of a journal bearing, it is more difficult to obtain and describe its lubrication with the Reynolds type equation.

The results of hydrodynamic lubrication of the slide conical bearing are presented in this work, taking into account the changes in the position of the bearing shaft in the longitudinal direction, i.e. along its axis of rotation. These changes can be caused by vibrations or varying load. The commercial CFD software – Fluent, form the Ansys Workbench platform, was used for the simulation (this software can be used in general for flow phenomena, not only for bearings), in which the equations of momentum, continuity and energy are solved, by application of the finite volume method. The aim of this work is to show how the operating parameters of the slide conical bearing can be influenced only by changing the position of the shaft along the axis of its rotation and also to verify the adopted methodology of investigations of the hydrodynamic lubrication of slide conical bearings.

2. Model

This study concerns the slide conical bearing shown in Fig. 1. On the left side of Fig. 1 is a cross-section at axial direction and on the right side of the Fig. 1 is a radial cross-section.

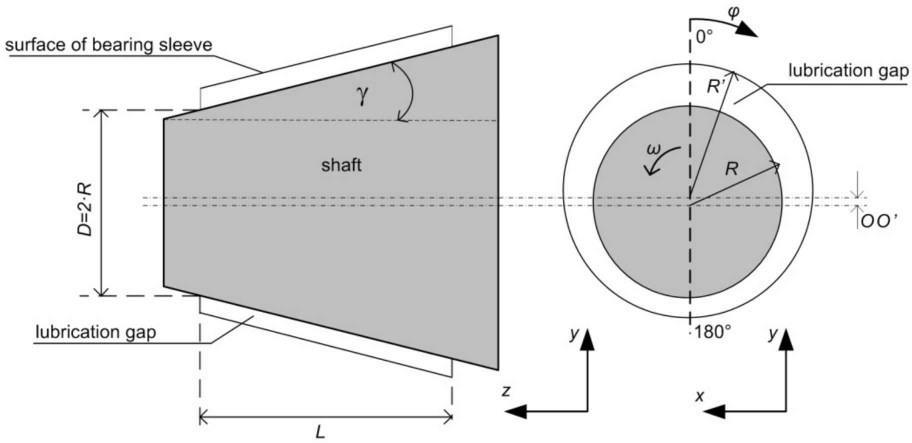

Fig. 1. The concerned slide conical bearing geometry

The bearing length, measured along the axis of the shaft, was of L = 50 [mm], while the radius of the shaft lowest cross-section (perpendicular the shaft axis) R = 50 [mm]. The radial clearance is defined as $\varepsilon = R' - R$ (R' is the radius of the sleeve at its lowest cross-section) and at the nominal position of was ε = 0.025 [mm]. The bearing operates in a steady state. The sleeve is stationary, the rotational speed of the shaft ω = 1500 [rpm]. The oil is incompressible and the flow is laminar, but non-isothermal (the parameters of the lubricating oil: density 850 [kg/m^3], specific heat 1006 [J/(kg·K)], heat conduction coefficient 0.025 [W/(m·K)].). The temperature of the shaft and sleeve surfaces and the temperature of supplying oil was 90°C. The sleeve is made of aluminium and conducts heat from bearing gap to the surroundings (the parameters of the sleeve material: density ρ = 2719 [kg/m3], specific heat c_p = 871 [J/(kg·K)], heat conduction coefficient κ = 202 [W/(m·K)], sleeve thickness δ = 5 [mm]). There was no slip of oil at bearing surfaces, which were smooth, rigid, and without deformations. The boundary condition for the side surfaces of bearing gap and at the ends of oil film, assumes that pressure was equal to the ambient pressure. The locations of the end of oil film is defined by the Gümbel [2] (half-Sommerfeld) boundary condition.

To describe non-Newtonian lubricating oil viscosity in relation to shear rate and temperature, the following equation was used:

$$\eta(\theta,T) = \eta_1(\theta)\cdot H(T) = K\cdot\theta^{n-1}\cdot\exp\left[\alpha_T\cdot\left(\frac{1}{T}-\frac{1}{T_\alpha}\right)\right], \tag{1}$$

where:

$$\eta_1(\theta) = K\cdot\theta^{n-1}, \tag{2}$$

is shear depended viscosity according to the Ostwald-de Waele [3-5] model, $H(T)$ is a function introducing the effect of the temperature on the viscosity of the oil, θ [s^{-1}] is the shear rate,

K = 0.01242 [Pa·s^n] is the flow consistency index and n = 0.9792 [-] is the flow behaviour (power-law) index. The coefficients for this model were determined by fitting the curve described by this model, to the experimental data, presented in paper [1], with the least squares approximation method (Statsoft Statistica software). It was assumed, that the lubricating oil has a properties as Shell Helix Ultra AV-L at a temperature 90°C. The parameter α_T=E_a/R is the ratio of the activation energy E_a = 5096 [J/kmol] to the thermodynamic constant R = 8314 J/(kmol·K) and T_α [K] is a reference temperature for which $H(T)$ = 1. In simulations was assumed, that the relative eccentricity:

$$\lambda = \frac{OO'}{\varepsilon} = 0.5, \tag{3}$$

where OO' [mm] is the absolute value of eccentricity (the nominal distance between the axis of shaft and axis of sleeve measured in perpendicular direction to these axes).

In Fig. 2 is presented a method of describing the offset of a shaft in relation to the sleeve position, measured along its axis of rotation.

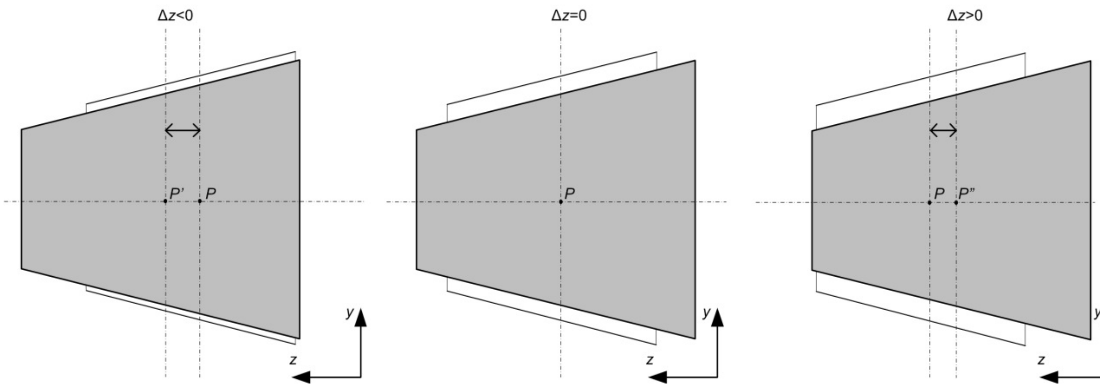

Fig. 2. The geometry of investigated conical bearing

The nominal position of the shaft is, when Δz. For cases, where shaft is approaching the sleeve, the Δz < 0, otherwise Δz > 0. The position of the shaft was changed every 0.01 [mm]. The calculations were carried out for following values of the opening angle of the shaft cone γ (opening angle of the sleeve cone): 10°, 15°, 30° and the minimum distance between the shaft and sleeve, measured along the axis of rotation, was depended on the value of cone opening angle γ.

The Fluent CFD module from the ANSYS Workbench 2 platform was used to prepare the geometry of the bearing and mesh, then to calculate the solution. The pressure based coupling algorithm was applied (Green-Gauss node based, second order pressure, the momentum second order upwind, the energy second order upwind).

3. Results

Figure 3 and 4 show the hydrodynamic pressure distributions in the lubrication gap for bearings with γ = 10°, which were determined in a plane containing the sleeve axis and the position of the maximum value of hydrodynamic pressure – Fig. 3 shows the results for Δz values from -0.06 [mm] to 0.00 [mm] while Fig. 4 show the results for Δz from 0.00 [mm] to +0.07 [mm]. In Fig. 5 and 6 are shown the hydrodynamic pressure distributions in the lubrication gap for bearings with γ = 10°, which were determined in a plane perpendicular to the shaft axis and passing through a location of maximum value of hydrodynamic pressure – Fig. 5 shows the results for Δz values from -0.06 [mm] to 0.00 [mm] while Fig. 6 show the results for Δz from 0.00 [mm] to +0.07 [mm]. The graphs do not show results for Δz = -0.07 [mm]. The reason for this is outlined later in this article.

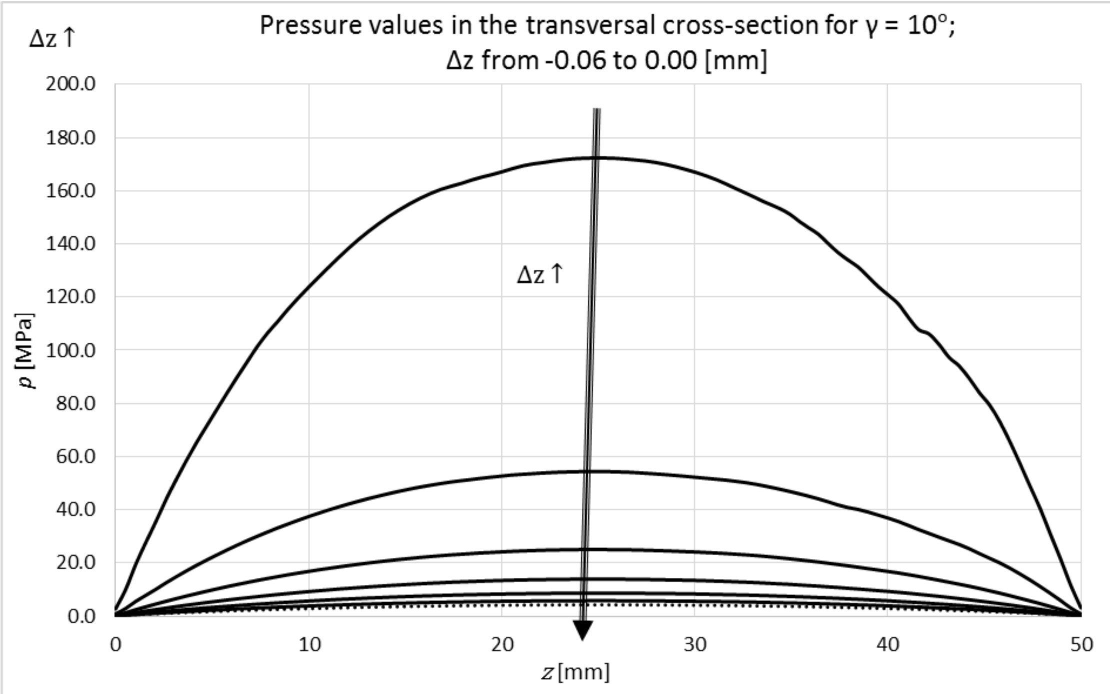

Fig. 3. The hydrodynamic pressure distributions in the longitudinal section containing maximum pressure in the lubrication gap of bearing with $\gamma = 10°$, for Δz from -0.06[mm] to 0.00 [mm] (the arrow indicates direction of the increment of Δz). The dotted line is the pressure distribution for the bearing with $\Delta z = 0.00$ [mm]

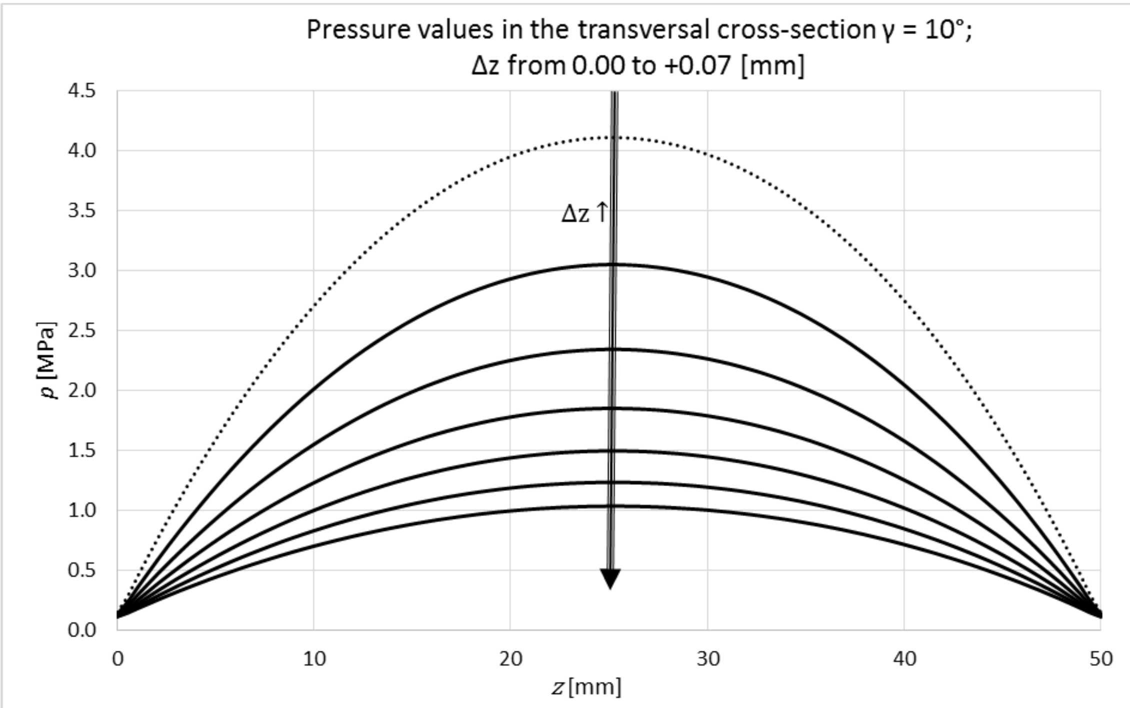

Fig. 4. The hydrodynamic pressure distributions in the longitudinal section containing maximum pressure in the gap of bearing with $\gamma = 10°$, for Δz from 0.00 [mm] to 0.07 [mm] (the arrow shows direction of the increment of Δz). The dotted line is the pressure distribution for the bearing with $\Delta z = 0.00$ [mm]

Figure 7 shows the values of the maximum pressure and the average pressure values generated in the lubrication gap for bearing with $\gamma = 10°$ and for concerned values of Δz, while Fig. 8 shows the values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque M_z for varying values of Δz for this bearing.

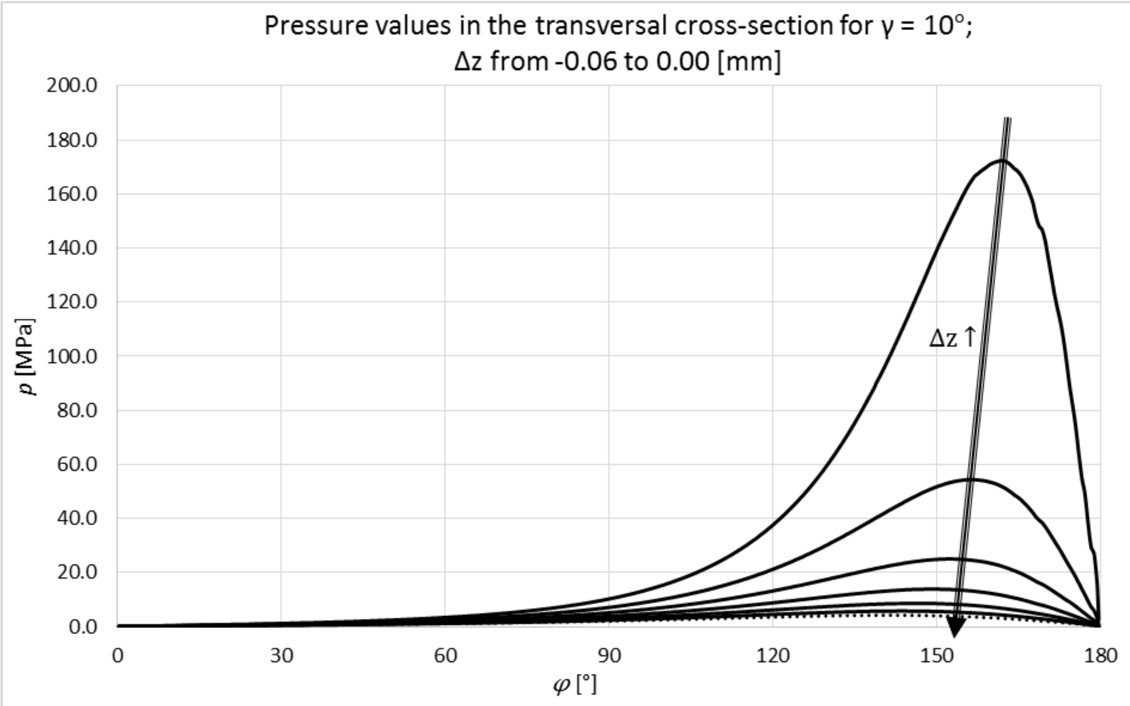

Fig. 5. The hydrodynamic pressure distributions in the cross-section containing maximum hydrodynamic pressure in the lubrication gap of bearing with $\gamma = 10°$, for Δz from -0.06 [mm] to 0.00 [mm] (the arrow indicates the direction of the increment of Δz). The dotted line is the pressure distribution for the bearing with $\Delta z = 0.00$ [mm]

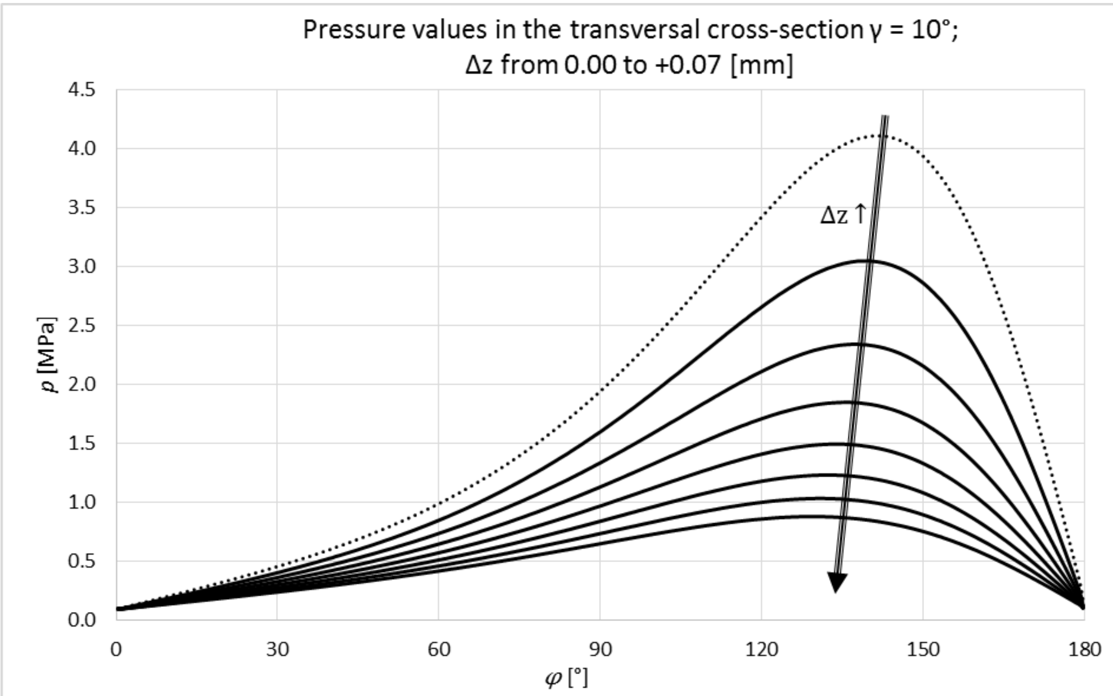

Fig. 6. The hydrodynamic pressure distributions in the cross-section containing maximum hydrodynamic pressure in the lubrication gap of bearing with $\gamma = 10°$, for Δz from 0.00[mm] to 0.07 [mm] (the arrow indicates the direction of the increment of Δz). The dotted line is the pressure distribution for the bearing with $\Delta z = 0.00$ [mm]

For very small distances of the shaft surface from the sleeve surface, the very high relative increases of the calculated values of pressures and bearing load carrying capacities were observed, while changing the value of Δz.

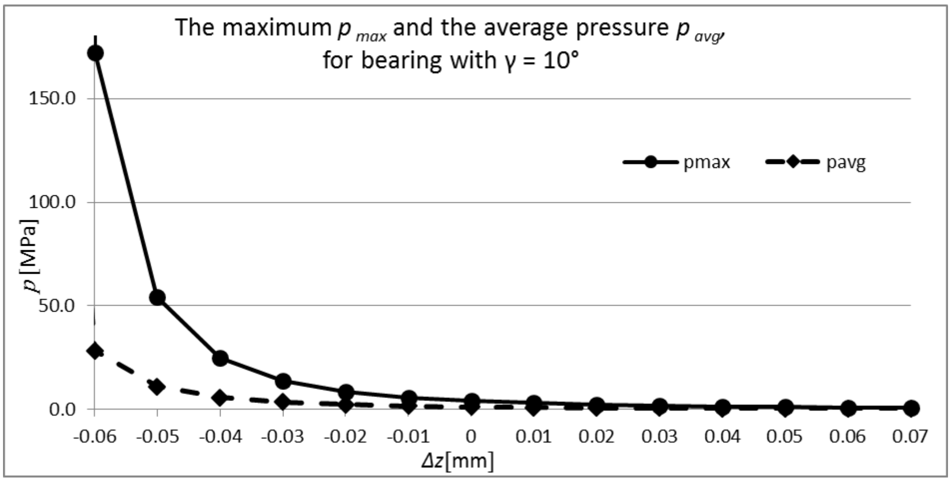

Fig. 7. The values of maximum p_{max} and average p_{avg} pressures in gap of bearing with γ= 10° for varying values of Δz

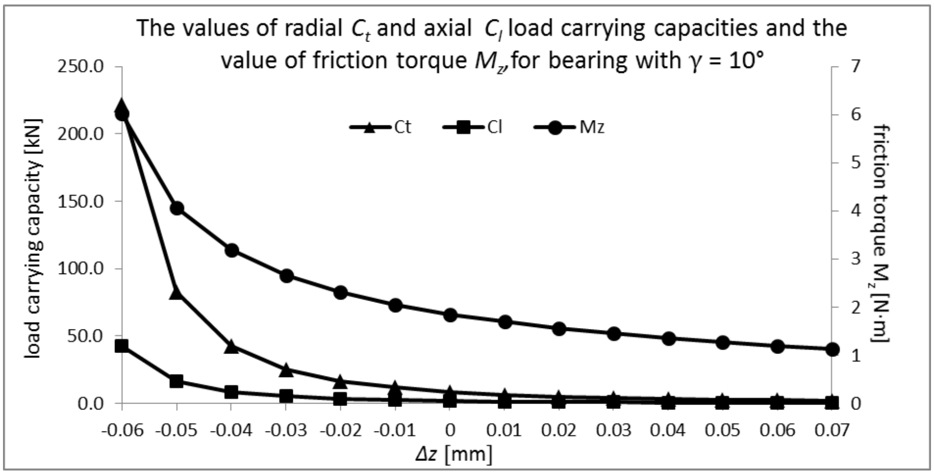

Fig. 8. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of Δz – the bearing with γ = 10°

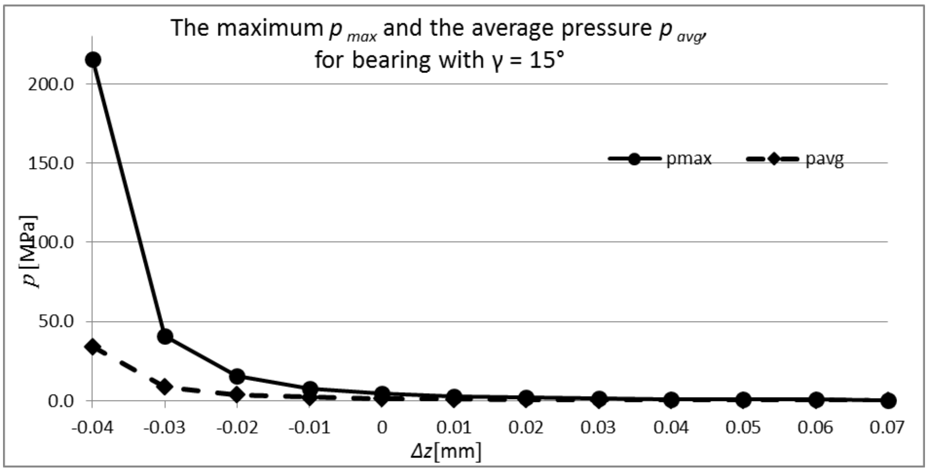

Fig. 9. The values of maximum p_{max} and average p_{avg} pressures in gap of bearing with γ= 15° for varying values of Δz

The maximum and the average pressure values in lubrication gap of bearing with γ = 15°, for considered values of Δz (in this case from -0.04 [mm]) are shown in Fig. 9. The radial C_t and axial C_l components of load carrying capacity and the value of friction torque M_z due to Δz changes, for bearing with γ = 15°, are presented in Fig. 10. The results for bearing, in which angle γ = 30°, are shown in Fig. 11 and 12 (in this case for Δz from -0.02 [mm]).

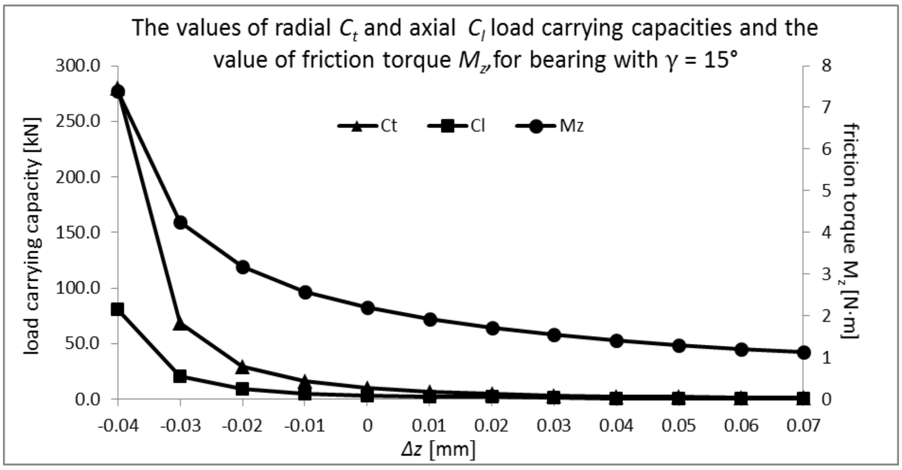

Fig. 10. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of Δz – the bearing with $\gamma = 15°$

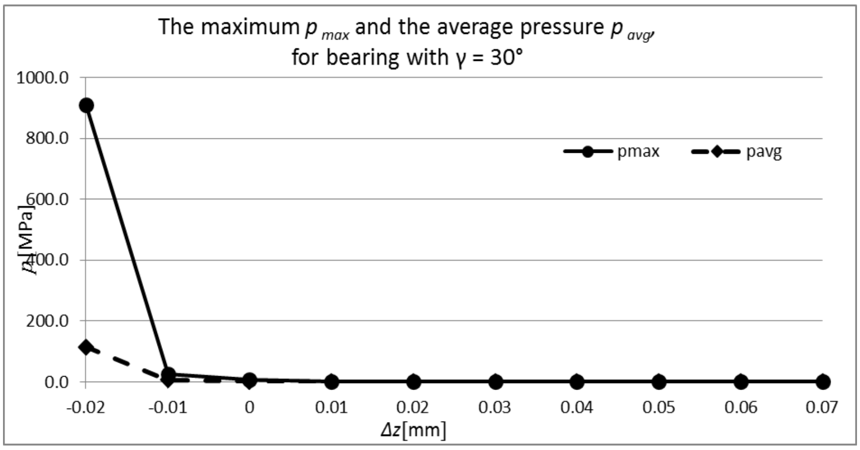

Fig. 11. The values of maximum p_{max} and average p_{avg} pressures of bearing with $\gamma = 30°$ for varying values of Δz

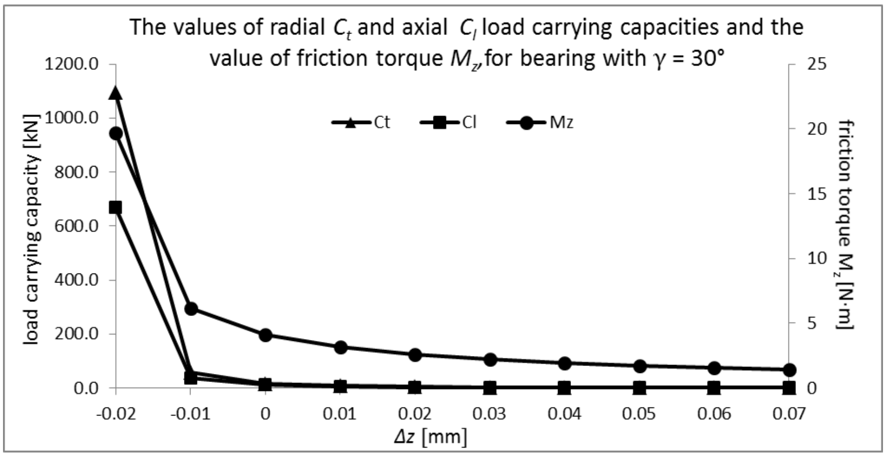

Fig. 12. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of Δz – the bearing with $\gamma = 30°$

4. Discussion and conclusions

The operational parameters of slide conical bearing, with taking into account the changes in the position of the bearing shaft along its axis of rotation, were investigated in this work. The position of the shaft was changed by every 0.01 [mm]. The smallest distance between the shaft and sleeve surfaces was dependent on the γ, i.e. the opening angle of the shaft and sleeve cone. For the bearing, for which $\gamma = 10°$, the shaft and sleeve were in contact, when $\Delta z = -0.08$ [mm], for bearing with γ = 15°, the contact was for Δz = -0.05 [mm], while for bearing with γ = 30°, its shaft and sleeve were in contact, when Δz = -0.03 [mm]. For the all three concerned values of cone opening angle γ, calculations were made for the maximum of Δz = +0.07 [mm]. The influence of the shaft position changes along its axis of rotation is particularly important for simulations of bearing with the conical geometry, because the movement of the shaft along this line causes not just a change in the value of relative eccentricity, but also a change in the defined value of radial clearance ε. Such a phenomenon does not occur, in the case of simulations of slide journal bearings (cylindrical geometry) – to achieve this in such case, it would be necessary to use a shaft or a sleeve of different diameter.

The values obtained for γ = 10° and Δz = +0.07 [mm] (p_{max} = $7.39\cdot10^3$ [MPa], p_{avg} = $5.03\cdot10^2$ [MPa], C_t = 4207 [kN], C_l = 758, M_z = 24 [N·m]) are not shown on the charts, due to the relatively large increase in calculated parameters, as in case of γ = 15° and Δz = +0.04 [mm] (Figs. 9 and 10) or in case of γ = 30° and Δz = +0.02 (Fig. 11 and 12). Such large relative increments of calculated quantities result from the very small height of the lubrication gap. Moreover, the adoption of the Gümbel condition causes, that in simulations, the oil film end is always at constant value of angular coordinate: φ = 180°, at which there are very high pressure gradients. It seems reasonable to apply in these cases the Reynolds boundary condition [2], which assumes that the pressure gradients at the end of the film are close to zero, and that the position of the end of the film may change. This unfortunately involves, in the adopted method, the introduction of a new computational loop in which the geometry mesh must be reprocessed at every step.

The main conclusions of the study are:

- the variations of the position of the shaft, with respect to the sleeve, measured along the axis of rotation, have a significant influence on the slide conical bearing operating parameters,
- by changing the position of the shaft along its axis of rotation, a change in the value of defined radial clearance is obtained,
- the use of commercial CFD software, designed for concerning general cases of flow phenomena, which instead of the Reynolds type equation of hydrodynamic lubrication, carry out simulations basing on the momentum and continuity equations, allows, for the defined lubrication gap, to determine the desired sliding bearing operating parameters,
- replacing the Gümbel boundary condition with the Reynolds condition of the oil film end, can contribute in obtaining more accurate results, which are in relation to those measured on actual bearings, especially for bearings simulations, that concern relatively low heights of lubrication gaps, such as, at cases, where the radial clearance value is low or the value of relative eccentricity is high.

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