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CFD ANALYSIS OF EFFECT OF MISALIGNMENT PLANE POSITION ON HYDRODYNAMIC LUBRICATION OF SLIDE CONICAL BEARING

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Abstract

The numerical calculations of the hydrodynamic lubrication of slide bearings can be carried out by modelling the oil flow for a given value of height of bearing lubrication gap. On the basis of the assumed height of the lubrication gap, the values of hydrodynamic pressures, load carrying capacities, friction forces, temperatures, can be determined. The bearing lubrication gap height can be influenced by many effects, e.g. misalignment between the shaft axis and the axis of the sleeve, vibrations, varying load, change in the viscosity value of lubricating oil caused by changes in temperature, pressure, shear rate or by oil ageing, wear of journal and sleeve surfaces. This article presents the results of numerical simulations concerning the influence of the misalignment between the axis of shaft and the axis of sleeve of the sliding conical bearing on its hydrodynamic lubrication, by taking into account the position of the plane in which the misalignment occurs. In this study, there was defined an angle between the plane in which the misalignment occurs and the plane in which lies the line of centres of corresponding bearing without misalignment. In this research, to investigate the impact of the position of the plane in which the misalignment occurs, the CFD software, designed to solve general flow phenomena, was used. It was assumed, that the bearings operate in a steady state conditions, the flow in the bearing lubrication gap is laminar and non-isothermal. A lubricating oil has shear properties as the Ostwald-de Waele fluid.

Keywords: *slide bearing, hydrodynamic lubrication, conical bearing, misalignment, CFD, pressure distribution*

1. Introduction

This article presents the results of numerical simulations concerning the influence of the misalignment [3] between the axis of shaft and the axis of sleeve of the sliding conical bearing on its hydrodynamic lubrication. The paper [2] discusses the influence of the value of the defined misalignment angle ζ on the conical bearing lubrication. This work introduces an additional parameter ξ, i.e. an angle describing the position of the plane in which the misalignment occurs.

The formation of a wedge, the height of lubrication gap, the values of the hydrodynamic pressure and the temperature of lubricating oil of a slide bearing, at the fixed rotational speed, are mainly dependent on the given load. Considering the required design parameters, such as geometric dimensions, rotational speed, load, by setting values for disposable parameters, such as oil viscosity or radial clearance, engineers are able to design or fit suitable bearings for a particular case. Unfortunately, the case is no longer so obvious if bearings of a more complex geometry, than the journal bearings, are considered, or when the characteristics of the lubricant strongly depend on various factors, for example, they are strongly non-Newtonian or are ferro-liquids working in the external magnetic field [4].

The numerical calculations of the hydrodynamic lubrication of slide bearings can be carried out by modelling the oil flow for a given value of height of bearing lubrication gap. On the basis of the assumed height of the lubrication gap, the values of hydrodynamic pressures, load carrying capacities, friction forces, temperatures, can be determined. In the simplest case of oil flow model in journal or conical slide bearing, the lubrication gap height h is only a function of the circumferential coordinate φ. In practice, the lubrication gap height is a function of time and varies due to vibrations, changes in the value and direction of load, shaft deflection, change in the lubricating oil viscosity value or wear of journal and sleeve surfaces. One of the parameters, that can be used to describe changes in the height of a lubrication gap, is the angle ξ determining the position of a plane, on which lie the misaligned axes of the shaft and the sleeve. Therefore, in this research, the height of the lubrication gap of conical slide bearing, was considered as a $h = h(\varphi, \zeta, \xi)$ function.

2. Bearing model

The conical slide bearing shown in Fig. 1, was concerned. On the left side of Fig. 1, there is a cross-section at plane, where the bearing has lowest diameter, on the right side of the Fig. 1; there is a cross-section along the axis of the bearing sleeve. It was assumed, that the change of the angle ζ value is made by rotating the bearing shaft at a point P_o, i.e. the point, at which the axis of the bearing shaft pierces the plane of lowest cross-section of the sleeve. This rotation was made in the plane *S'*, for which $\xi = 0°$, i.e. the plane, on which lies the axis of the sleeve and the *line of centres* – line passing through centres of the shaft and sleeve in the cross-section plane. The value of ξ angle was changed by rotating the plane *S'* relative to the sleeve axis. The calculations were made for the following angles (see Fig. 1) $\xi = 0°, 45°, 90°, 135°, 180°, 225°, 270°, 315°$, at the considered angles of misalignment $\zeta = 0.001°, 0.004°, 0.007°$ (for a greater values of this angle, there occurred the contact between the bearing shaft and sleeve surfaces).

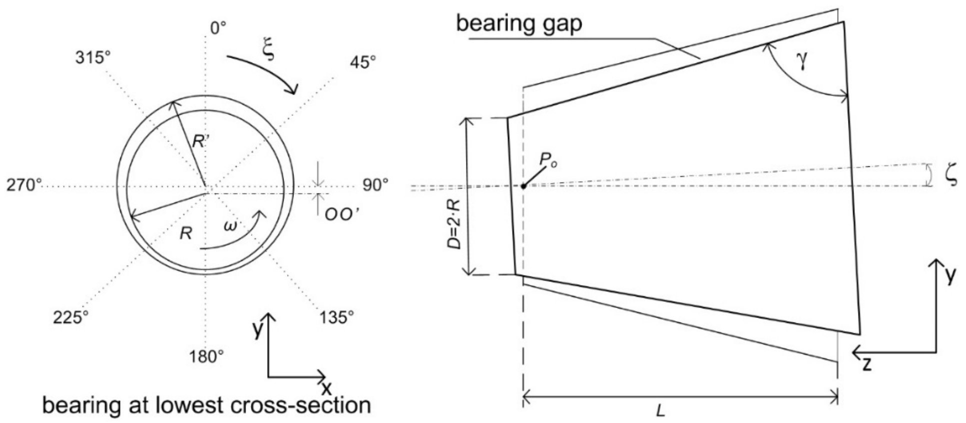

Fig. 1. The geometry of investigated conical bearing

The assumptions and conditions adopted in the simulations are:
- the bearing length $L = 50$ [mm] (measured along the axis of the sleeve),
- the shaft cross-section, perpendicular to shaft axis, has the lowest radius of $R = 50$ [mm],
- the opening angle of the shaft cone and the opening angle of the sleeve cone was 10°, therefore $\gamma = 80°$ (Fig. 1),
- the radial clearance $\varepsilon = R' - R = 0.025$ [mm], where R' is the radius of the sleeve at its lowest cross-section,
- the bearing operates in a steady state condition– constant rotational speed, no vibrations, laminar and incompressible flow of oil,
- the flow of lubricating oil was 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 surface and also the supplying oil was 90°C
- the stationary sleeve, made of aluminium, conducts heat from bearing gap to the surroundings (the parameters of the sleeve material: density $\rho = 2719$ [kg/m^3], specific heat $c_p = 871$ [J/(kg·K)], heat conduction coefficient $\kappa = 202$ [W/(m·K)], sleeve thickness $\delta = 5$ [mm],

– the rotational speed of the shaft ω = 1500 [rpm],
– there was no slip of oil at bearing surfaces, which were smooth, rigid, and without deformations,
– the pressure on the side surfaces of bearing gap was equal to the ambient pressure,
– there was imposed the Gümbel (half-Sommerfeld) boundary condition [4],
– the calculations were made for the relative eccentricity [4]:

$$\lambda = \frac{\mathrm{OO'}}{\varepsilon} = 0.5\,, \tag{1}$$

where OO' [mm] is the absolute value of eccentricity (the nominal distance between the axis of shaft and axis of sleeve),
– the Ostwald-de Waele [5, 6] model was imposed:

$$\tau = K \cdot \theta^{n}\,, \tag{2}$$

where τ [Pa] it the shear stress, 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 effects of shear rate and temperature on the viscosity of the lubricating oil where included according to the formula:

$$\eta(\theta,T) = \eta_1(\theta)\cdot H(T) = K\cdot\theta^{n-1}\cdot\exp\left[\alpha_T\cdot\left(\frac{1}{T}-\frac{1}{T_\alpha}\right)\right], \tag{3}$$

where η_1 [Pa·s] is the viscosity, dependent on shear rate due to the Ostwald-de Waele relationship (2) [5, 6], and $H(T)$ is a function describing the effect of the temperature on the viscosity of the oil. The parameter $\alpha_T = E_a/R$ 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.

The ANSYS Workbench 2 platform and the Fluent CFD module were 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

In Figs. 2 and 3 are shown the hydrodynamic pressure distributions in the lubrication gap of concerned bearing, for the misalignment ζ = 0.004°. These pressure distributions were determined along a line passing through a position, at which there is maximum value of pressure – in Fig. 2 is the transversal cross-section (the axial coordinate has a constant value) while in Fig. 3 is the longitudinal section (the angular coordinate has a constant value). The length z [mm] is the distance measured from the lowest cross-section of the bearing (front of the bearing) along the axis of the bearing sleeve. The pressures are in an absolute scale (the ambient pressure in calculations: $p_{amb} = 1.013\cdot10^5$ [Pa]). The dashed line is the pressure distribution for the bearing without misalignment (when ζ = 0.000°). Other distributions are described by the corresponding value of the ξ angle – the solid lines show the distributions for ξ = 0°, 45°, 90°, 135°, 180°, while the dotted lines represent distributions for ξ = 225°, 270°, 315°.

The most significant differences, with respect to the bearing without misalignment, occurred for the ξ angles, for which the misalignments lies in the plane equivalent to the line of centres plane, while the lowest changes were observed for the perpendicular plane, but noting, that for

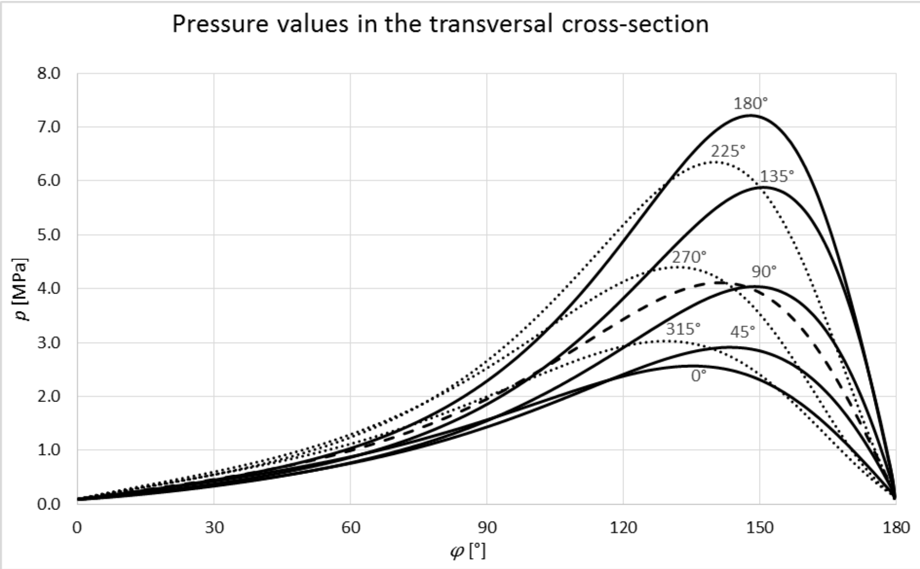

Fig. 2. The hydrodynamic pressure distributions in the lubrication gap of bearing, for the misalignment $\zeta = 0.004°$ – the transversal cross-section along the location of the maximum value of hydrodynamic pressure: dashed line – pressure distribution for the bearing without misalignment; the solid lines for $\xi = 0°, 45°, 90°, 135°, 180°$, the dotted for $\xi = 225°, 270°, 315°$

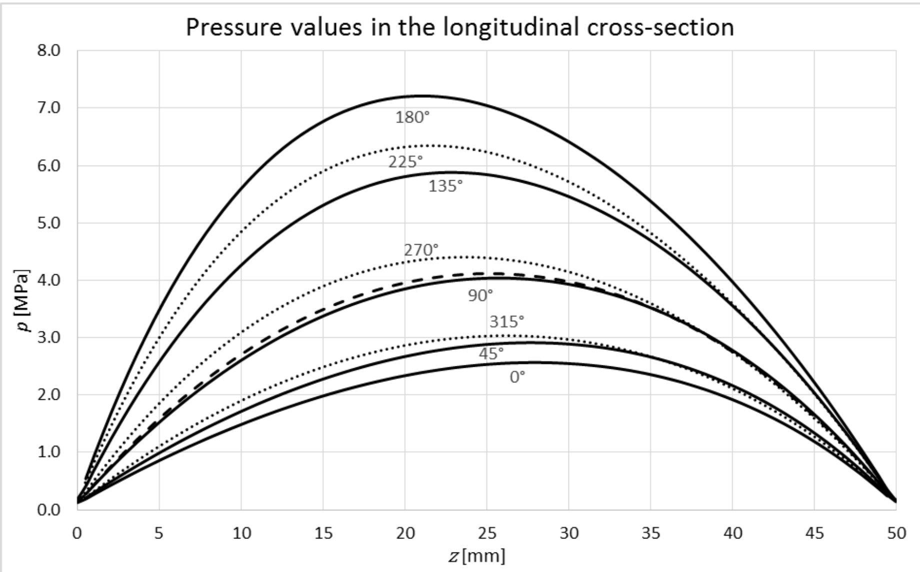

Fig. 3. The hydrodynamic pressure distributions in the lubrication gap of bearing, for the misalignment $\zeta = 0.004°$ – the longitudinal section along the location of the maximum value of hydrodynamic pressure: dashed line – pressure distribution for the bearing without misalignment; the solid lines for $\xi = 0°, 45°, 90°, 135°, 180°$, the dotted for $\xi = 225°, 270°, 315°$

$\xi = 270°$, the values of hydrodynamic pressures where slightly greater, than for $\xi = 90°$ – this is due to the angle ξ measurement method, i.e. in the opposite direction than the shaft rotation, therefore for an angle $\xi = 90°$ the height of the gap in the area of occurrence of maximum pressures slightly increased, while for $\xi = 270°$ the height of the gap in this area decreased. The graphs also show, that the position of the maximum pressure in the lubrication gap was changing, according to the variations of ξ angle. In Fig. 4 is shown the location of the maximum value of hydrodynamic pressure in the lubrication gap, with respect to angular coordinate φ [°] and distance z [mm], measured from the front of bearing. Fig. 5 shows the values of the maximum pressure and the average pressure generated in lubrication gap for bearing with $\zeta = 0.004°$ and for concerned values of ξ angle, while in Fig. 6 are shown the calculated values of the radial C_t and axial C_l

components of load carrying capacities, and also the friction torques M_z, i.e. the values corresponding to the minimum torque required to maintain the rotation of the bearing shaft. The greatest difference in the position of the maximum pressure in the lubrication gap, relative to the axial variable z, occurs between the cases, where $\xi = 0°$ and $\xi = 180°$ (i.e. 7 mm), while in relation to angular coordinate φ, the greatest difference is between the planes of misalignment, for which $\xi = 135°$ and $\xi = 315°$ (i.e. ~22 °).

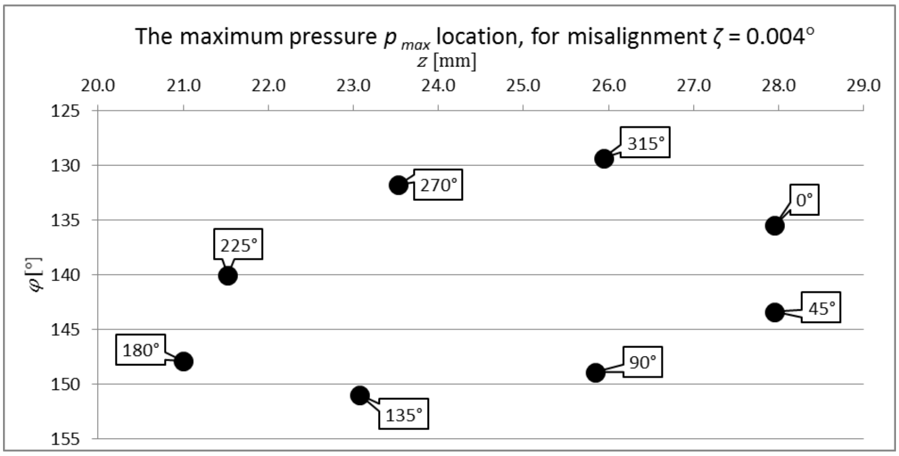

Fig. 4. The maximum pressure location for varying values of angle ξ, while $\zeta = 0.004°$

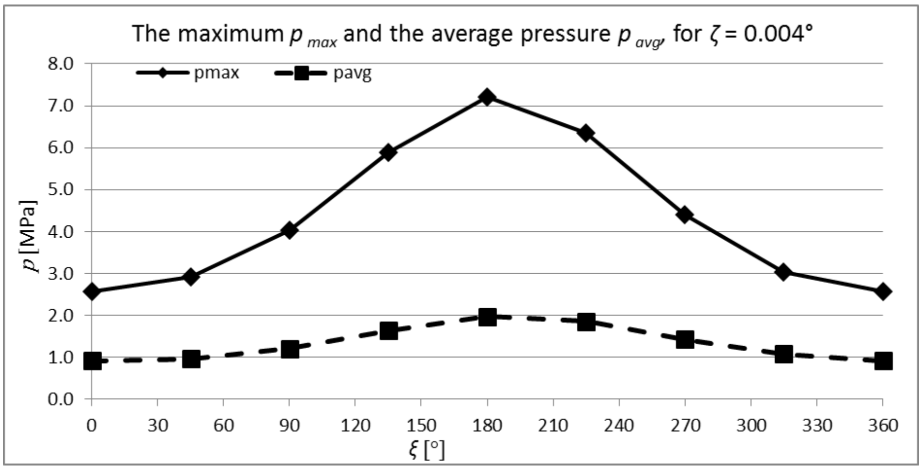

Fig. 5. The values of maximum p_{max} and average p_{avg} pressures for varying values of angle ξ, while $\zeta = 0.004°$

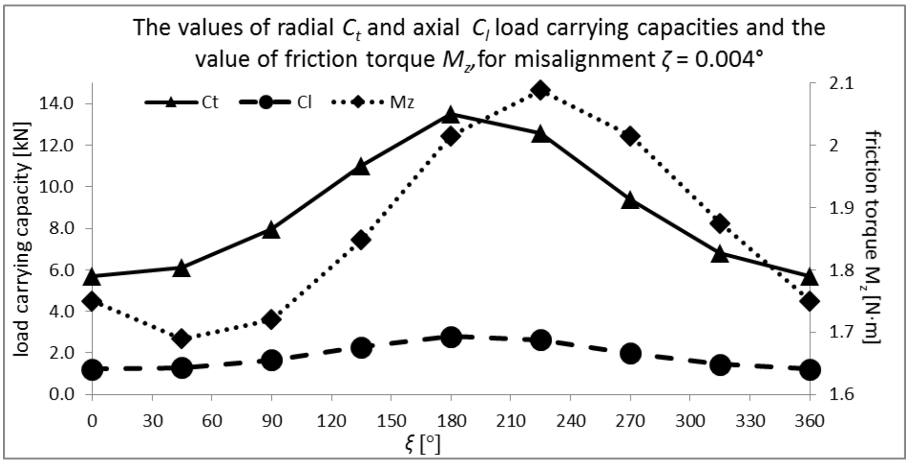

Fig. 6. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of angle ξ, when bearing misalignment $\zeta = 0.004°$

Figure 7 show the changes of maximum hydrodynamic pressure location in lubrication gap of the bearing, for which the defined misalignment $\zeta = 0.001°$, for its different positioning of the misalignment plane, described by the ξ angle. Fig. 8 shows the values of the maximum and the average pressures in lubrication gap for this bearing, for varying values of ξ angle, and Fig. 9 shows the values of the radial C_t, axial C_l load carrying capacities, and also the values of friction torque M_z.

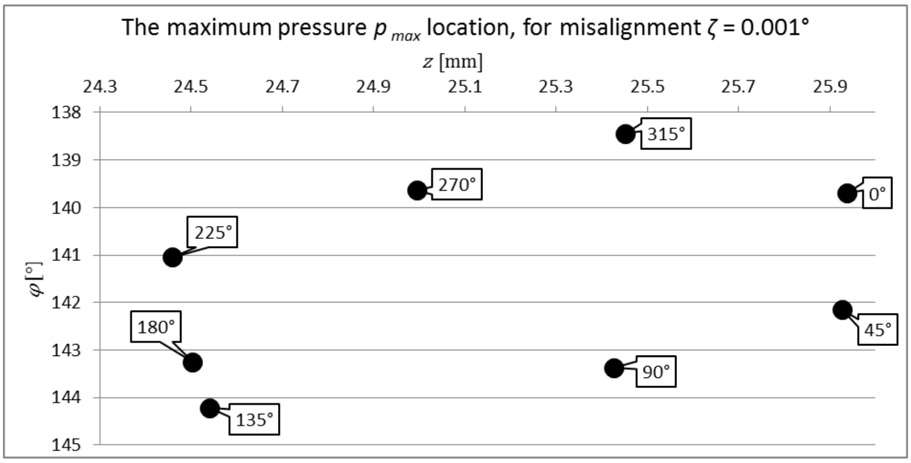

Fig. 7. The maximum pressure location for varying values of angle ξ, while $\zeta = 0.001°$

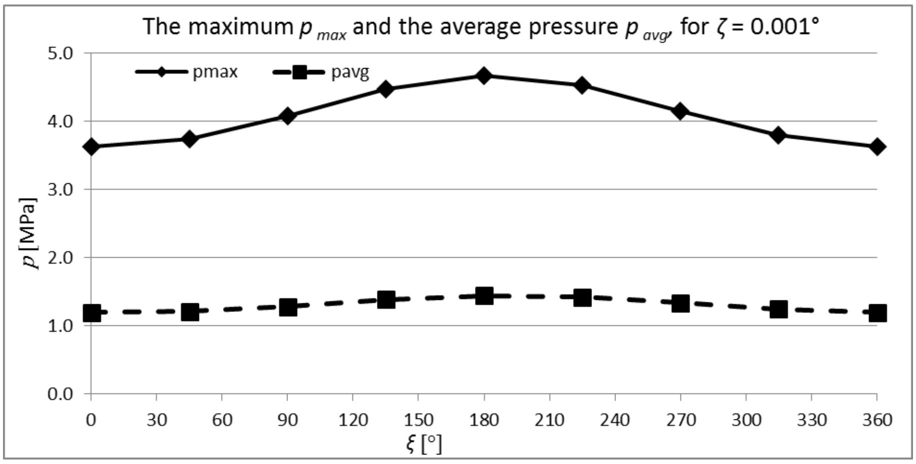

Fig. 8. The values of maximum p_{max} and average p_{avg} pressures for varying values of angle ξ, while $\zeta = 0.001°$

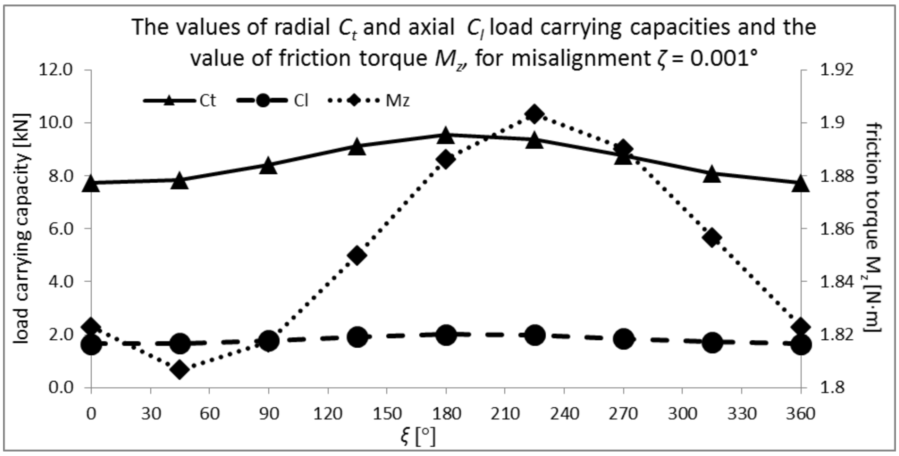

Fig. 9. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of angle ξ, when bearing misalignment $\zeta = 0.001°$

For the bearing with misalignment $\zeta = 0.007°$, the maximum hydrodynamic pressure location in lubrication gap, according to ξ angle, was shown in Fig. 10, while the maximum and average pressure values are shown in Fig. 11. The radial C_t and axial C_l components of load carrying capacity, and also the values of friction torque M_z. can be found in Fig. 12.

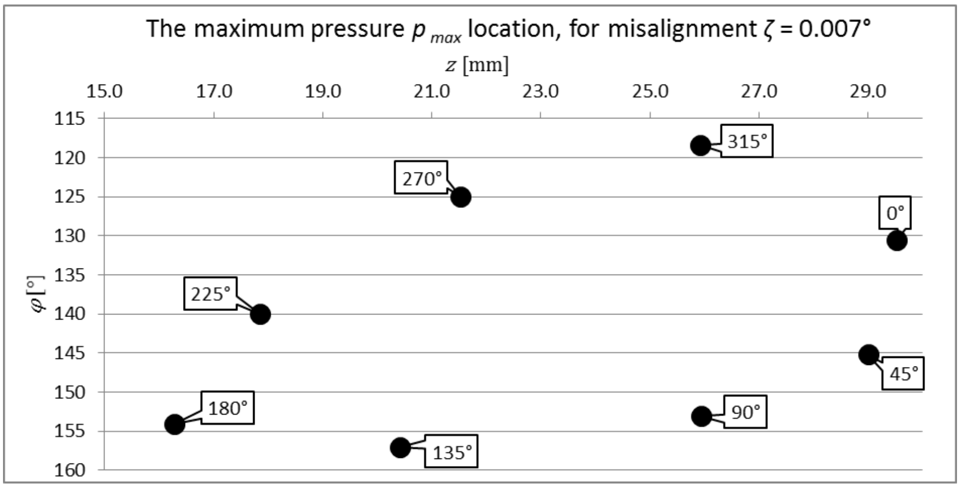

Fig. 10. The maximum pressure location for varying values of angle ξ, while $\zeta = 0.007°$

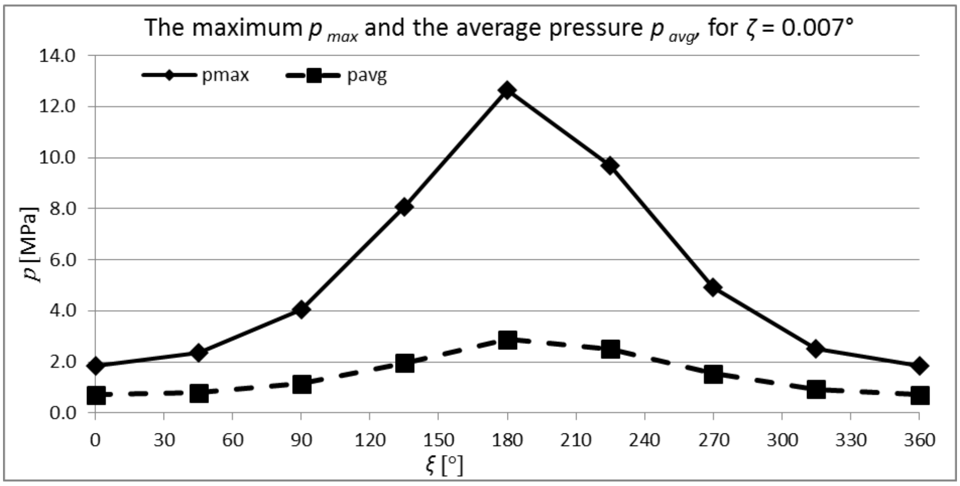

Fig. 11. The values of maximum p_{max} and average p_{avg} pressures for varying values of angle ξ, while $\zeta = 0.007°$

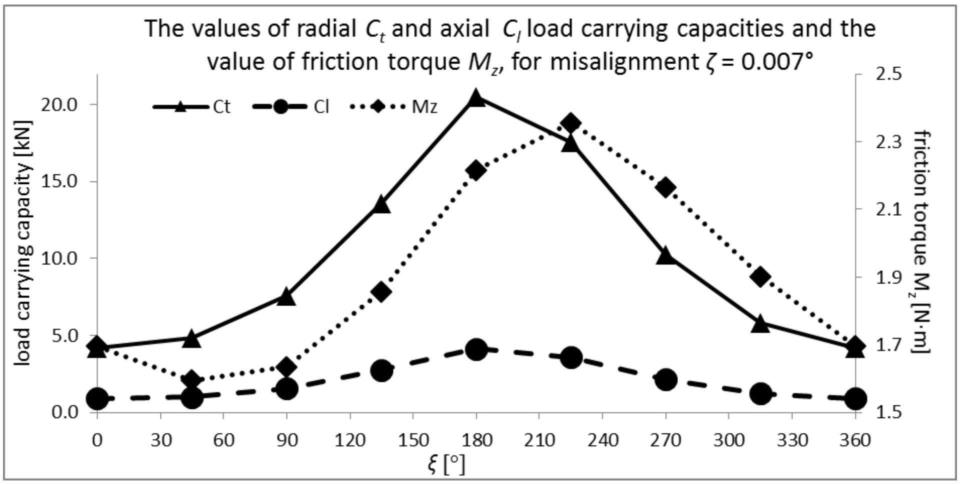

Fig. 12. The values of radial C_t and axial C_l components of load carrying capacity and the value of friction torque for varying values of angle ξ, when bearing misalignment $\zeta = 0.007°$

4. Discussion and conclusions

In this article is presented the significance of taking into account the influence of misalignment between the axis of the shaft and the axis of the sleeve of conical slide bearing, on its hydrodynamic lubrication. In work [2] was presented the effect of the shaft axis misalignment with respect to the sleeve axis, by defining the angle between these axes, but this angle of misalignment lies only in one plane – the plane of the line of centres. In this research, the generalized considerations were made by introducing the rotation of the plane, in which the misalignment occurs. The rotation of the plane was determined by the ξ angle, measured according to the plane of the line of centres. This is a still simplification, because, in general, the axis of rotation of the shaft does not have to lie in one plane with the axis of the bearing sleeve. However, for the imposed conditions and for predefined distribution of the height of bearing lubrication gap, the operating parameters of the bearing were determined for the specific positions of the shaft. The greater the value of the angle of misalignment ζ causes the greater differences in the resulting values, while changing the ξ angle of the misalignment plane. Tab. 1 specifies the maximum differences Δ^{max} of the calculated values for the investigated misalignments (i.e. assumed values of ζ angle) and investigated planes of the misalignment (i.e. assumed values of ξ angle).

The following conclusions were drawn from the research:

- the numerical determination of the operating parameters of the slide bearings can be carried out at a predetermined height of the lubrication gap,
- the commercial CFD software for general flow phenomena is a good tool for solving the hydrodynamic theory of lubrication problems,
- the research showed, that not only the value of misalignment angle between the shaft and sleeve axes, has a significant impact on the hydrodynamic lubrication and operating parameters of the slide conical bearing, but also the plane, in which it occurs has an major influence on the obtained values, therefore it should be taken into account in simulations.

Tab. 1. The maximum differences Δ^{max} of calculated values, for the assumed values of ζ and ξ angles

ζ [°]	$\Delta^{max} p_{max}$ [MPa]	$\Delta^{max} p_{avg}$ [MPa]	Δ^{max} C_t [kN]	Δ^{max} C_l [kN]	Δ^{max} M_z [Nm]	$\Delta^{max} \varphi$ [°]	$\Delta^{max} z$ [mm]
0.001	1.0	0.2	9.5	0.4	0.1	5.8	1.5
0.004	4.6	1.0	13.5	1.6	0.40	21.6	7.0
0.007	10.8	2.2	16.3	3.2	0.76	38.7	13.3

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