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CFD ANALYSIS OF THE IMPACT OF A CONE OPENING ANGLE PARAMETER ON THE HYDRODYNAMIC LUBRICATION OF THE CONICAL SLIDE BEARING

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Abstract

The height of the oil lubrication gap is the primary quantity that determines in simulations the operating parameters of a hydrodynamic slide bearing. It is influenced by multiple effects, such as vibrations during operation, varying load, misalignment between the shaft axis and the axis of the bearing sleeve, the roughness of the journal and sleeve surfaces, 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 etc. It is important to take into account such effects considering hydrodynamic lubrication simulations and design of the slide bearings.

The one of the factors influencing the height of the oil lubrication gap of the conical slide bearing is the difference between the opening angle of the cone of bearing shaft and opening angle of the cone of bearing sleeve. The aim of this work is to investigate the impact of the difference between the values of these angles on the hydrodynamic lubrication of the conical slide bearing. The commercial CFD software Ansys Fluent, from the Ansys Workbench 2 platform, was used to determine the hydrodynamic pressure distributions, load carrying capacities and friction torques of the simulated bearings. It was assumed, that the bearings operate in a steady state conditions, the flow in the bearing lubrication gap is laminar and non-isothermal, there is no misalignment between the axis of bearing journal and axis of bearing sleeve, the surfaces of the journal and sleeve are smooth and lubricating oil acts as a liquid described by the Ostwald-de Waele power law model.

Keywords: *hydrodynamic lubrication, slide bearing, conical bearing, cone opening angle, CFD, pressure distribution*

1. Introduction

The main parameters, which are imposed when designing hydrodynamic slide bearings, are the radial clearance and the viscosity of lubricating oil. An appropriate setting of their values at the given load and rotational speed of the slide bearing journal (or shaft), is essential to achieve the lubricated friction. The viscosity is determined by selection of the lubricating oil, but the viscosity does not have to be constant and can vary depending on the temperature [2, 6], pressure [3], shear rate [2, 3, 6], effect of aging [8], and for ferro-oils also depends on the magnetic field [2]. For given values of the radial clearance, viscosity, load and shaft rotational speed, the oil wedge in the bearing lubrication gap is forming due to the eccentricity of the rotation axis of the bearing shaft in relation to the axis of the bearing sleeve. For a fixed value of eccentricity of the bearing, there is a distribution of the height h of the lubricating gap, which in the simplest case, is only a function of the circumferential coordinate φ, i. e.:

$$h = h(\varphi) . \quad (1)$$

In fact, the value of h depend on many factors, such as vibrations during operation of slide bearing, varying load, shaft deflection caused by the load and, due to this, the misalignment between the shaft axis and the axis of the bearing sleeve, the roughness of the journal and sleeve surfaces, change in the viscosity value or wear of journal and sleeve surfaces. Therefore, when carrying out simulations on hydrodynamic lubrication of the slide bearings and numerical designation of their operating parameters, it is required to model appropriately the height of the lubrication gap.

The one of the factors influencing the height of the oil lubrication gap of the conical slide bearing is the difference between the opening angle of the cone of bearing shaft and opening angle of the cone of bearing sleeve. This paper shows results of CFD simulations of an impact of the difference $\Delta\alpha$ between the values of these angles on the hydrodynamic lubrication of the conical slide bearing. It was assumed in this investigation, that the height of lubricating gap depends only on the circumferential coordinate φ and on the value of $\Delta\alpha$.

2. Modelling of lubricating oil flow

In this research, the conical slide bearing shown in Fig. 1 was concerned.

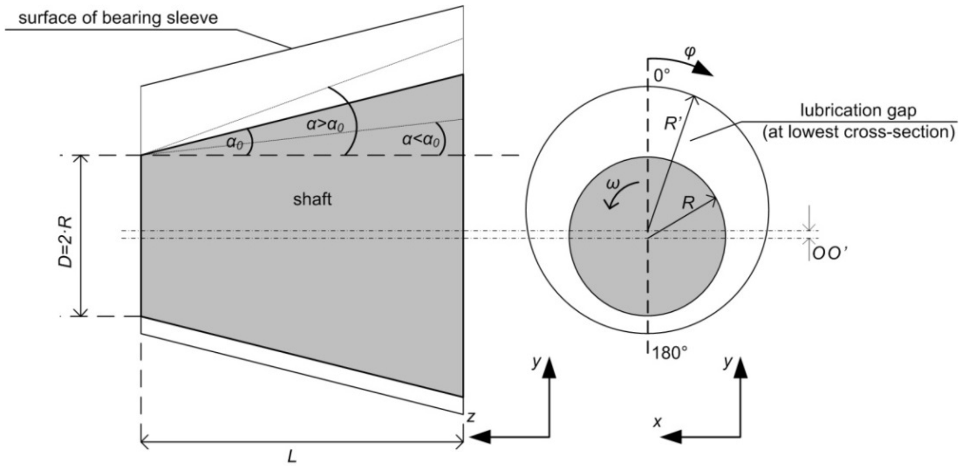

Fig. 1. The geometry of investigated conical bearing

The length bearing: L = 50 mm. The radius of the shaft at the lowest cross-section (considered as the front of the bearing): R = 50 mm. The radial clearance: $\varepsilon = R' - R$ = 0.025 mm, where R' is the radius of bearing sleeve at bearing front. There is no misalignment between the symmetry axes of bearing shaft and sleeve. The nominal opening angle of the cones of the shaft and sleeve: α_0 = 10°.

The opening angle of the cone of bearing sleeve remained constant and only the value of cone opening angle α of bearing shaft was changed. There was assumed, that:

$$\Delta\alpha = \alpha - \alpha_0, \tag{2}$$

and we have:

- $\Delta\alpha < 0$ for $\alpha < \alpha_0$,
- $\Delta\alpha = 0$ for $\alpha = \alpha_0$,
- $\Delta\alpha > 0$ for $\alpha > \alpha_0$.

The values of $\Delta\alpha$ varied, in this work, between – 0.010° and + 0.010°, every 0.001°. The relative eccentricity ratio λ is defined as [6]:

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

where OO' is the bearing eccentricity. For the examined bearing λ = 0.6.

The investigations were carried out for bearing, which operates in a steady state, i.e. when the value of rotational speed is constant, without vibrations, with laminar and incompressible flow of lubricating oil and constant value of relative eccentricity. Furthermore, there was assumed that surfaces of the bearing shaft and sleeve are smooth, rigid and without deformations, there is no slip of lubricating oil at bearing surfaces, the pressure on the side surfaces of bearing gap is equal to the ambient pressure, the temperature of the surface of bearing shaft and also the supplying oil is 80°C. The Gümbel boundary condition [5] (half-Sommerfeld condition), for the end of the oil film, was imposed in simulations. The rotational speed of the shaft was n_r = 3200 [rpm]. The bearing sleeve was made of steel and conducts heat from bearing lubrication gap to the surroundings (parameters of the bearing sleeve material were, as follows: the density ρ = 8030 [kg/m^3], the specific heat c_p = 503 [J/(kg·K)], the heat conduction coefficient κ = 16.27 [W/(m·K)], the sleeve thickness δ = 1 [mm]). The study includes the effects of shear rate and temperature on the viscosity of the lubricating oil according to the formula:

$$\eta(\dot{\gamma}, T) = \eta_1(\dot{\gamma}) \cdot H(T), \tag{4}$$

where η_1 [Pa·s] viscosity is dependent on shear rate, while

$$H(T) = \exp\left[\alpha_T \cdot \left(\frac{1}{T} - \frac{1}{T_\alpha}\right)\right], \tag{5}$$

is a factor introducing the effect of 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 Ostwald-de Waele power law lubricant model for non-Newtonian fluids [6, 9] was adopted in this research, thus the oil viscosity changes depending on the shear rate is represented as follows:

$$\eta_1(\dot{\gamma}) = K \cdot \dot{\gamma}^{n-1}, \tag{6}$$

where 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 that model were determined with the least squares approximation method and fitting the curve described by this model, to the experimental data, as in paper [1]. Other parameters of the lubricating oil are the density 850 [kg/m^3], the specific heat 1006 [J/(kg·K)], the heat conduction coefficient 0.025 [W/(m·K)].

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-Gaus node based, second order pressure, the momentum second order upwind, the energy second order upwind).

3. Results

The examples of calculated hydrodynamic pressure distributions, for various values of $\Delta\alpha$, are shown in Fig. 2. The value δ_p of distance between the front of bearing and the location of maximum hydrodynamic pressure value, measured parallel to the axis of the shaft, was indicated at hydrodynamic pressure contours. For $\Delta\alpha < 0$, i.e. the decrease of the value of the opening angle of the shaft, at a constant angle of the opening angle of the sleeve, causes that the position of maximum pressure value approaches towards the front of bearing (because the lubrication gap narrows in this direction), while for $\Delta\alpha > 0$ and increase of the value of $\Delta\alpha$ results in increase of δ_p. On each of the graphs is specified the maximum temperature generated in the lubrication gap.

The greater the value of $\Delta\alpha$, the changes in the maximum temperature generated in the lubricating oil by viscous heating are more relevant.

In Tab. 1 there are presented the values of maximum hydrodynamic pressure p_{max} in the bearing lubrication gap, the transversal C_t and longitudinal C_l load carrying capacities, the frictional angular momentum M_l, the distance δ_p of between the front of bearing and the location of maximum hydrodynamic pressure value, the value of φ_p, i.e. the angle where is the value of maximum hydrodynamic pressure and the value of angle Φ_{Ct}, at which operates the resultant transverse force C_t.

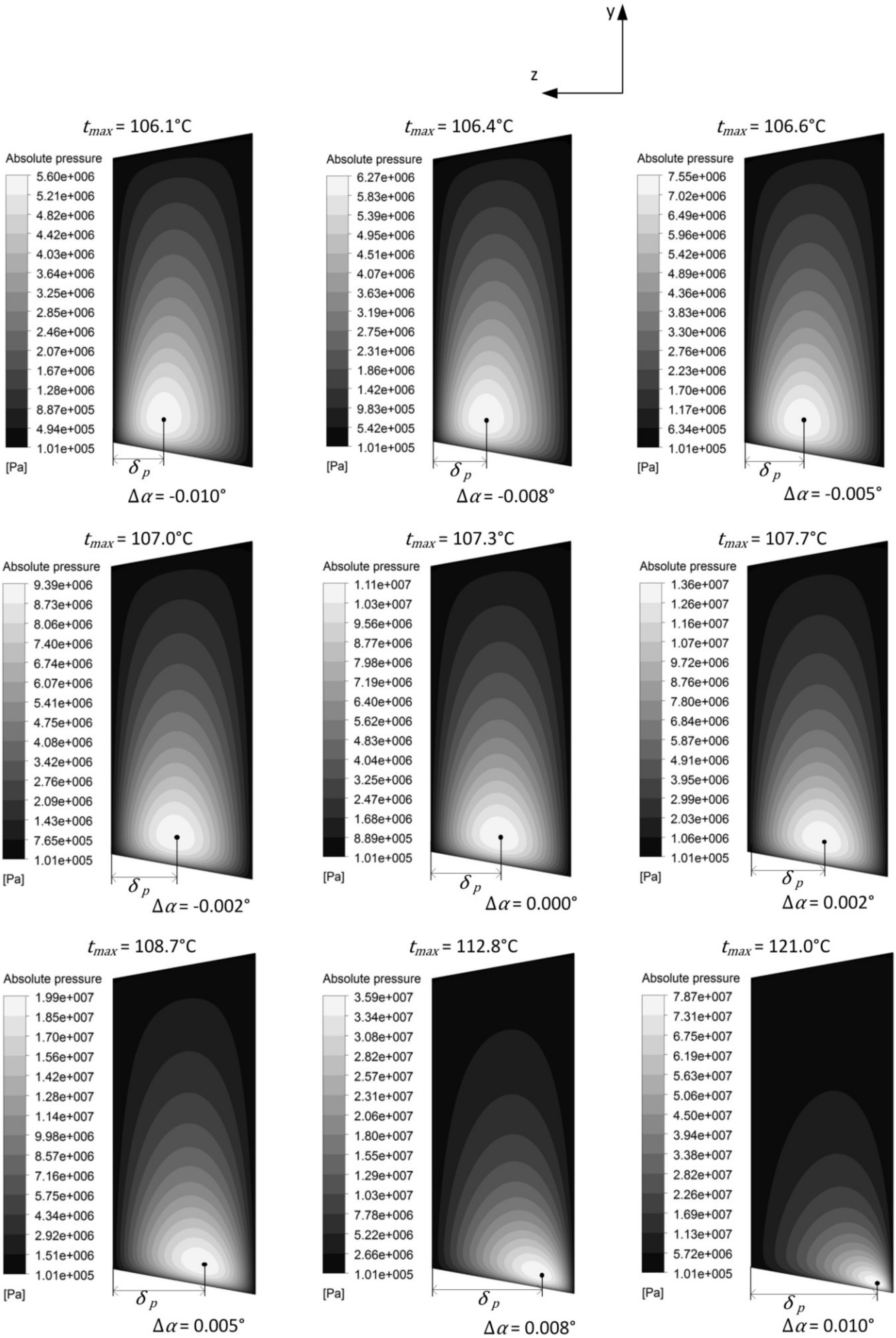

Fig. 2. The calculated hydrodynamic pressure distributions for varying values of Δα

Tab. 1. The values of maximum hydrodynamic pressure p_{max} in the bearing lubrication gap, the transversal Ct and longitudinal C_l load carrying capacities, the frictional angular momentum M_l, the distance δ_p of between the front of bearing and the location of maximum hydrodynamic pressure value, the value of the angle φ_p, where is the maximum hydrodynamic pressure and the value of angle Φ_{Ct}, at which operates the resultant transverse force C_t

$\Delta\alpha$ [°]·10^3	p_{max} [MPa]	C_t [kN]	C_l [kN]	M_l [N·m]	δ_p [mm]	φ_p [°]	Φ_{Ct} [°]
-10	5.60	11.35	2.37	2.90	17.9	140.9	301.0
-9	5.93	12.00	2.51	2.95	18.4	140.9	301.3
-8	6.27	12.77	2.67	3.00	18.9	141.6	301.6
-7	6.66	13.51	2.82	3.05	19.4	141.6	302.1
-6	7.08	14.38	3.00	3.09	19.9	142.2	302.6
-5	7.55	15.32	3.20	3.15	20.5	142.8	303.0
-4	8.09	16.34	3.41	3.21	21.0	143.4	303.6
-3	8.70	17.51	3.65	3.27	22.0	144.0	304.1
-2	9.39	18.83	3.92	3.34	22.6	144.0	304.7
-1	10.20	20.32	4.23	3.42	23.6	144.6	305.4
0	11.13	22.01	4.58	3.50	24.6	145.8	306.1
1	12.24	23.95	4.97	3.58	25.5	146.3	306.8
2	13.58	26.19	5.43	3.68	26.5	147.3	307.7
3	15.20	28.83	5.97	3.79	28.0	148.4	308.6
4	17.24	31.95	6.60	3.91	29.5	149.4	309.6
5	19.86	35.72	7.35	4.05	31.1	150.4	310.8
6	23.37	40.37	8.28	4.20	32.7	152.0	312.1
7	28.34	46.27	9.45	4.39	34.8	153.5	313.6
8	35.93	54.08	10.99	4.61	37.4	156.1	315.4
9	49.25	65.10	13.15	4.89	40.6	159.2	317.7
10	78.74	82.45	16.50	5.29	43.7	162.9	320.8

The difference $\Delta\alpha$ between the cone-opening angle of the shaft and the cone-opening angle of the sleeve has a major impact on the operating parameters of the slide bearing. The relative change in the maximum pressures and load carrying capacities are at the highest tested values of the $\Delta\alpha$. Moreover, when changing the $\Delta\alpha$, there also occurs a change of the position of the maximum pressure value and the direction of the resultant bearing force. With an increase in the $\Delta\alpha$ value, the direction of the resultant lateral force approaches to the line that connects centres of the shaft and sleeve (i.e. *line of centres*). In Fig. 3 there are shown the curves of the maximum pressure and load carrying capacities, according to the $\Delta\alpha$ value, which changes may be described as exponential function. Fig. 4 shows the change in the position of the maximum value of the pressure in the lubricating gap with respect to angular coordinate and distance from the front of bearing – the increase of the $\Delta\alpha$ causes the increase of the distance δ_p and increase of the value of angular coordinate φ_p.

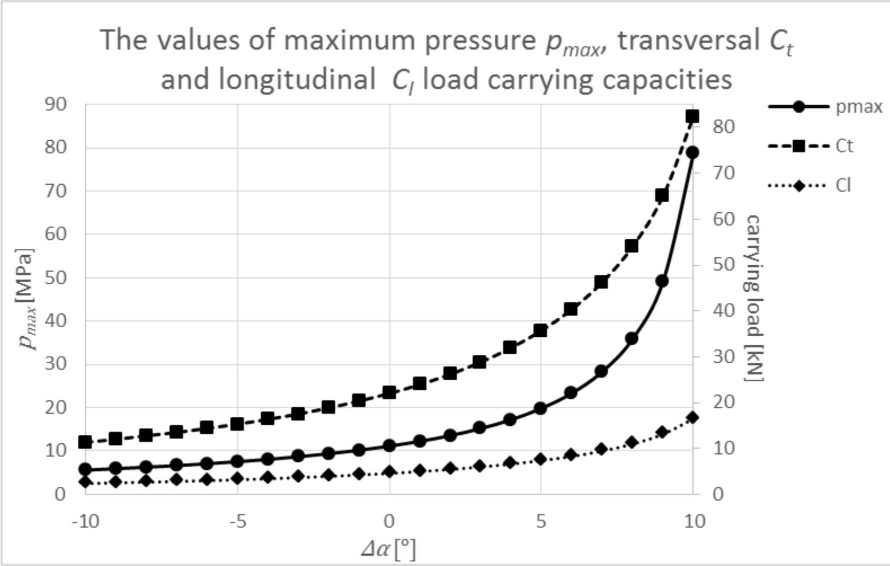

Fig. 3. The calculated values of maximum hydrodynamic pressure p_{max}, the transversal C_t and longitudinal C_l load carrying capacities for varying $\Delta\alpha$

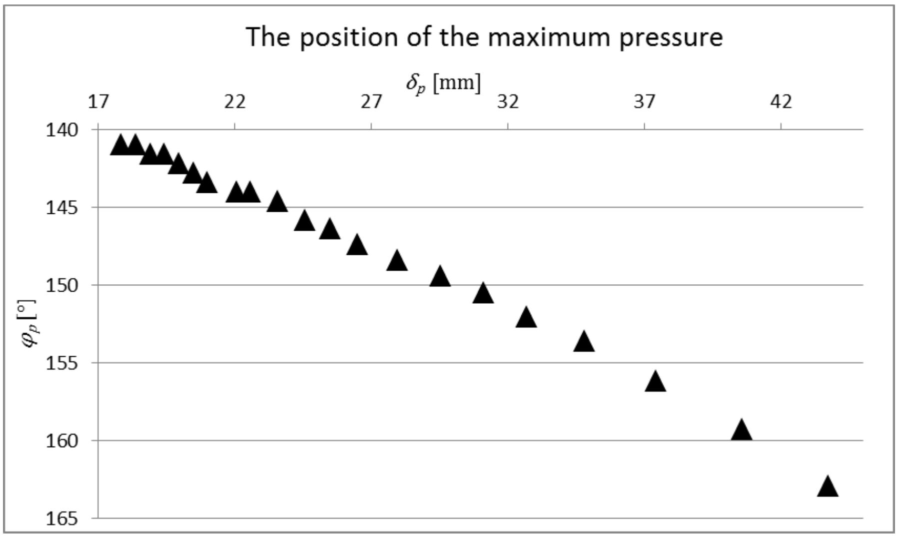

Fig. 4. The position of the maximum hydrodynamic pressure for varying values of $\Delta\alpha$ (increase of the $\Delta\alpha$ causes increase of the distance δ_p and value of angular coordinate φ_p

4. Conclusions

1. In this research, the operational parameters of bearing were calculated for a predefined distribution of the height of the oil gap. In fact, the height of lubrication gap is determined by given load, rotational speed, viscosity of the used oil. However, this study gives insight in how the change of height of the oil gap, in this case due to the difference between the opening angles of the cone of the bearing shaft and cone of the bearing sleeve, affects the hydrodynamic pressure distribution generated in the gap and operating parameters of the bearing.
2. It was shown, that for the investigated values of the difference $\Delta\alpha$ between the opening angles of cones of the shaft and sleeve, changes in the values of the calculated operating parameters of the slide conical bearing are significant and become particularly important for larger values of $\Delta\alpha$, i.e. when the surface of the shaft approaches the surface of the sleeve. Then, also the temperature of lubricating oil increases considerably and the torque required maintaining rotation of the shaft is greater.

3. It is noted that the change of the $\Delta\alpha$ value causes change in the position of the maximum pressure generated in the lubrication gap. Moreover, the direction of the generated radial component C_t changing, and due to this, and to the change of the value of axial component C_l, the direction of the resultant bearing force also varies.
4. The application of CFD methods is a useful way, to consider the hydrodynamic lubrication of the slide conical bearings. The presented calculations were performed for a bearing, which operates in a steady state. The author believes that greater opportunities and more accurate results, closer to reality, can be obtained using FSI methods (*Fluid-Structure Interaction*) [3], where not only the equations for the flow are solved, but also an interaction of the fluid with a solid is taken into account (e.g. to include in calculations the vibration of the unevenly loaded shaft or deformation of the surfaces of the shaft and sleeve).

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