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CFD ANALYSIS OF NON-NEWTONIAN AND NON-ISOTHERMAL LUBRICATION OF HYDRODYNAMIC CONICAL BEARING

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Abstract

In this work is shown the result of CFD simulation of hydrodynamic conical bearing lubrication with consideration of non-isothermal oil flow in a bearing lubrication gap and also with assumption, that oil has non-Newtonian properties. The determination of hydrodynamic pressure distribution in bearing gap was carried out by using the commercial CFD software ANSYS Academic Research for fluid flow phenomenon (Fluent). Calculations were performed for bearings without misalignment, i.e. where the cone generating line of bearing shaft is parallel to the cone generating line of bearing sleeve. The Ostwald-de Waele model for non-Newtonian fluids was adopted in this simulation. The coefficients of Ostwald-de Waele relationship were determined by application of the least squares approximation method and fitting curves described by this model to the experimental data, obtained for some motor oils, presented in previous work. The calculated hydrodynamic pressure distributions were compared with the data obtained for corresponding bearings, but assuming that the flow in the bearing lubrication gap is isothermal. Some other simplifying assumptions are: a steady-state operating conditions of a bearing, incompressible flow of lubricating oil, no slip on bearing surfaces, pressure on the side surfaces of bearing gap is equal to atmospheric pressure. This paper presents results for bearings with different rotational speeds and of different bearing gap heights.

Keywords: *conical bearing, hydrodynamic lubrication, CFD simulation, non-Newtonian oil, non-isothermal flow, pressure distribution*

1. Introduction

The viscosity of oil greatly depends on temperature. The influence of temperature and shear rate on dynamic viscosity of some selected oils and ferro-oils was shown in papers [1, 2]. Change in the viscosity values of the oil influences the operating parameters of a slide bearing. Therefore taking into account, in numerical simulations, the effect of temperature changes caused by viscous heating, can significantly affect the correctness of the obtained results.

In this work is shown the result of CFD simulation of hydrodynamic lubrication in slide conical bearing with consideration of non-isothermal oil flow in a bearing lubrication gap and also with assumption, that oil has non-Newtonian properties. The determination of hydrodynamic pressure distribution in bearing gap was carried out by using the commercial CFD software ANSYS Academic Research for fluid flow phenomenon (Fluent). The Ostwald-de Waele power-law lubricant model for non-Newtonian fluids was adopted in this simulation. The Ostwald-de Waele relationship [4, 5] can be written as:

$$\tau = K \cdot \dot{\gamma}^{n}, \tag{1}$$

where: τ is the shear stress in [Pa], $\dot{\gamma}$ is the shear rate in [s^{-1}], K is the *flow consistency index* in [$Pa \cdot s^{n}$] and n is the dimensionless *flow behaviour index*. The coefficients for that model were determined with the least squares approximation method and fitting the curve to the experimental results obtained for motor oils presented in previous paper [1].

If considering the non-isothermal and non-Newtonian flow in ANYS Fluent, the temperature dependence and the shear rate dependence on the viscosity can be included and the dynamic viscosity η [Pa·s] of such fluid can be described as:

$$\eta = \eta_1(\dot{\gamma}) \cdot H(T), \tag{2}$$

where η_1 [Pa·s] is viscosity dependent on shear rate and $H(T)$ [-] is the temperature dependence (from the Arrhenius law):

$$H(T) = \exp\left[\alpha \cdot \left(\frac{1}{T} - \frac{1}{T_\alpha}\right)\right], \tag{3}$$

where $\alpha = E_a/R$ 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 $H(T) = 1$.

It was assumed, that bearings are lubricated with the oil with tribological properties as Shell Helix Ultra AV-L [1]. The determined values of indices in this case are $K = 0.01242$ [Pa·s^n], $n = 0.9792$, $\alpha = 5096$ [1/K], $T_\alpha = 90$ [K]. The values of α and T_α were determined (using the Statsoft Statistica 9.1 software and the least squares approximation method) by fitting the Eq. (3) to the results presented in the paper [1]. While considering isothermal flow, the parameter α was set to zero, i.e. the oil had constant temperature and properties as Shell Helix Ultra AV-L at $T = T_\alpha$ = 90 [K]. The obtained hydrodynamic pressure distributions and calculated bearing load carrying capacities were compared with the data obtained for corresponding bearings, but assuming that the flow in the bearing lubrication gap is also non-Newtonian but isothermal. This paper presents results for bearings with different rotational speeds and of different bearing gap heights.

2. Results

The calculations were conducted for the bearings without misalignment (where the cone generating line of bearing shaft is parallel to the cone generating line of the bearing sleeve). The geometry of investigated bearing is shown in Fig. 1. Studies were performed for bearing with a length of L = 50 [mm] and the radius of shaft at lowest cross-section of the bearing R = 50 mm. The radial clearance was $\varepsilon = R' - R$ = 0.025 [mm], where R' is a radius of bearing sleeve and $\alpha = 80°$.

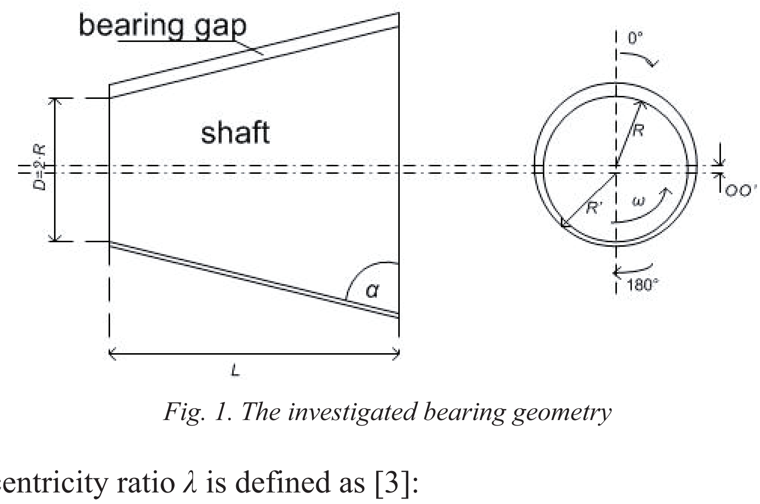


Fig. 1. The investigated bearing geometry

The relative eccentricity ratio λ is defined as [3]:

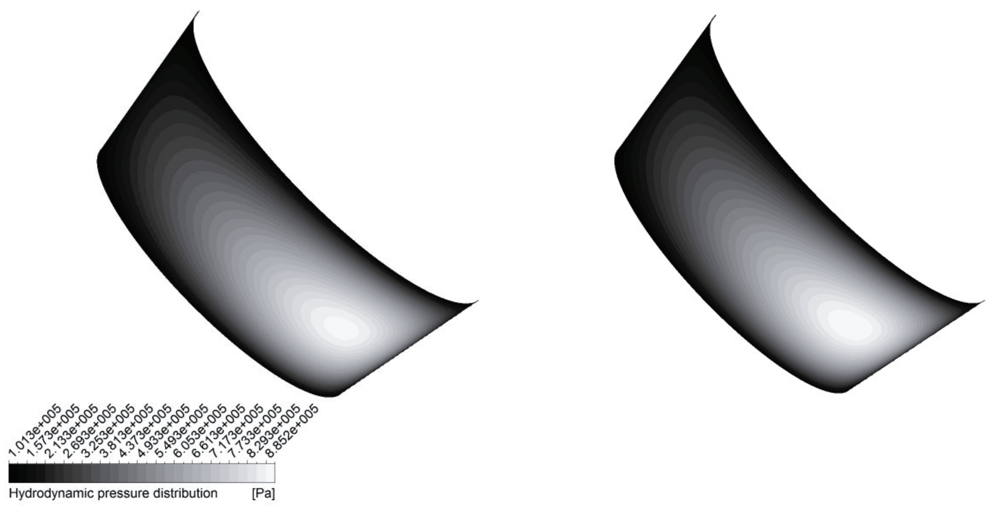
$$\lambda = \frac{OO'}{\varepsilon}, \tag{4}$$

where OO' is the bearing eccentricity.

The assumptions adopted in the simulations are: a steady-state operating conditions of a bearing, laminar and incompressible flow of lubricating oil, no slip on bearing surfaces, vibrations of bearing shaft are negligible, pressure on the side surfaces of bearing gap is equal to atmospheric pressure, the temperature of bearing shaft and supplying oil is 90 [K], bearing sleeve is made of steel and conducts heat from bearing lubrication gap to the surroundings. The parameters of bearing sleeve material are density ρ = 8030 [kg/m^3], specific heat c_p = 503 [J/(kg·K)], heat conduction coefficient λ = 16.27 [W/(m·K)], sleeve thickness δ = 1 [mm], while 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 calculations included the Gümbel [3, 6] (also known as half-Sommerfeld) boundary condition. The results are presented in the form of contours of hydrodynamic pressure values. The tables contain values of maximum oil pressure generated in bearing gap and also bearing load carrying capacities (slide conical bearing is capable for carrying radial C_r and axial C_w forces).

The geometry, meshing, setup solution and post-processing were made with the ANSYS Workbench software with CFD Fluent module. In each case the number of generated mesh nodes was about 4 000 000. The pressure based SIMPLE algorithm (Green-Gaus node based, the pressure second order, the momentum second order upwind, the energy second order upwind) was adopted in the calculations. The contour graphs present the hydrodynamic pressure distributions in the absolute pressure scale.

Figure 2-4 present results for the bearing with the relative eccentricity λ = 0.4. In Fig. 2 is shown hydrodynamic pressure distribution in lubrication gap of the bearing, which operates at the speed n_r = 500 [rpm]. Fig. 2a shows results for bearing lubricated with non-Newtonian oil and with assumption, that there occurs the viscous heating, while Fig 2b is hydrodynamic pressure distribution of corresponding bearing lubricated with non-Newtonian oil, but when it is assumed, that the flow is isothermal (there is no viscous heating). Results for the bearings, which operate at n_r = 3200 [rpm] and at n_r = 7200 [rpm] are presented in an analogous manner in Fig. 3a, 3b, 4a and 4b.

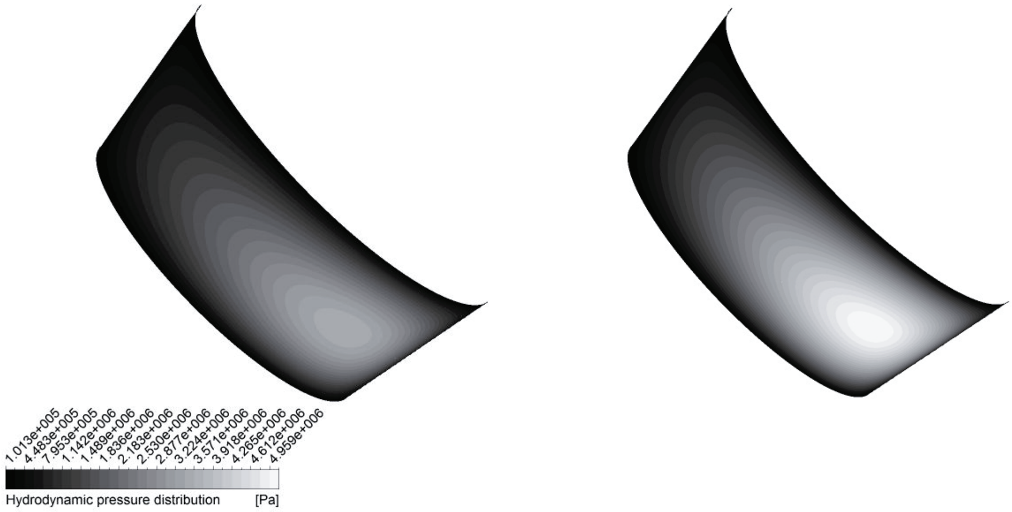


a) *b)*

Fig. 2. Hydrodynamic pressure distribution in conical slide bearing; λ = 0.4, n_r = 500 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication

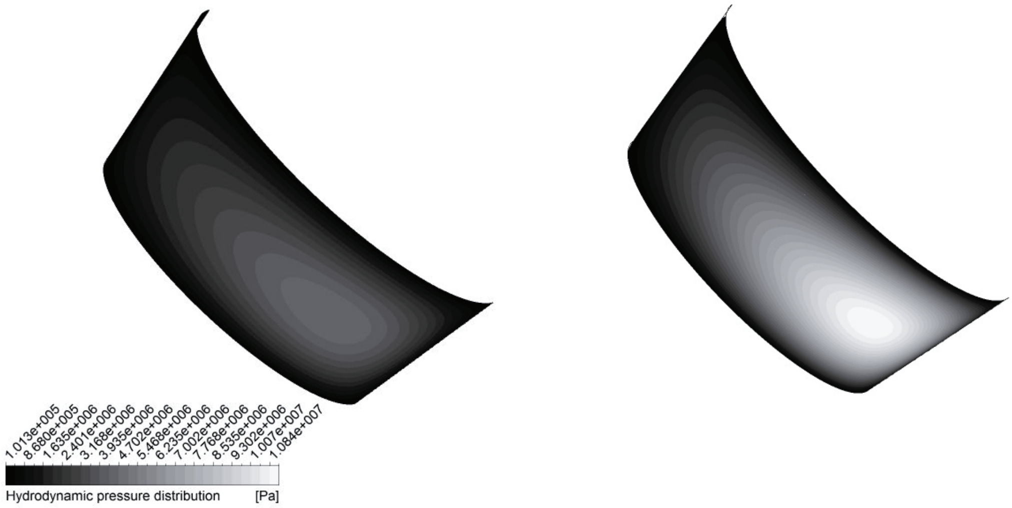
One can observe that the values of hydrodynamic pressure in the case, where the viscous heating effect was considered, were lower than for corresponding bearing, but with the assumption, than lubrication is isothermal. The decrease of lubricating oil viscosity value is due to the increase of temperature caused by the viscous heating effect; therefore, the hydrodynamic pressure values are lower. The relative decrease, (max. value of pressure) is ~1.5 [%] in the case

of this bearing, but when the shaft rotational speed is 3200 [rpm], the viscous heating causes the more significant decrease in the pressure values, i.e. ~34 [%] while for speed 7200 [rpm] it is even more than 62 [%]. The higher the rotational speed of the shaft causes grater values of shear rate, so more heat is generated thorough the friction between particles of oil. Tab. 1 show the comparison of maximum values of pressure generated in the bearing lubrication gap, for different relative eccentricities and rotational speeds and also the relative drop of maximum values of pressure for investigated bearings, according to corresponding bearings, but with isothermal lubrication.
Tab. 2 shows the comparison of the values of radial and axial components of load carrying capacity of bearings with different relative eccentricity and shaft rotational speeds.



a) *B)*

Fig. 3. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.4, n_r = 3200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication



a) *b)*

Fig. 4. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.4, n_r = 7200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication

In Fig. 5, 6, 7 are shown the hydrodynamic pressure distributions for bearings with $\lambda = 0.5$. Fig. 5a, 6a, 7a show results obtained, when considering viscous heating effect, while results for corresponding bearings with isothermal lubrication are shown in Fig. 5b, 6b and 7b.

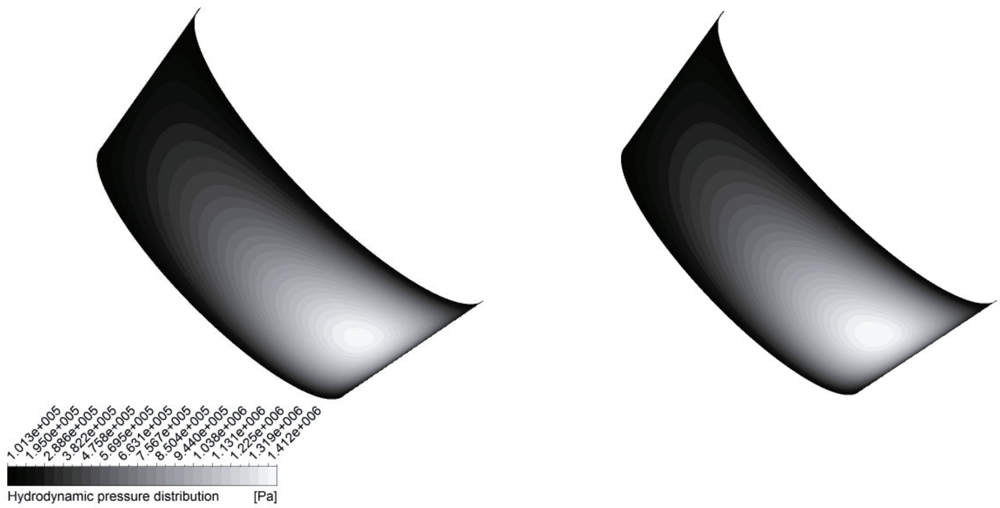
In Fig. 8a, 9a, and 10a are presented results for bearings with considered viscous heating for relative eccentricity ratio $\lambda = 0.6$ and in Fig. 8b, 9b and 10b show results for corresponding bearings, but when isothermal flow is considered.

Tab. 1. The maximum value of hydrodynamic pressure in the absolute scale and the values of relative drop of maximum pressure, when considering the viscous heating effect

Relative eccentricity	Shaft speed	Maximum pressure value in the bearing gap p_{max} [Pa]		Relative drop of max. pressure
λ [-]	n_r [rpm]	Considering viscous heating	Isothermal flow of oil	Δp_{max} [%]
0.6	500	$2.44264 \cdot 10^6$	$2.47904 \cdot 10^6$	1.47
	3 200	$9.81624 \cdot 10^6$	$1.48553 \cdot 10^7$	33.92
	7 200	$1.22737 \cdot 10^7$	$3.27572 \cdot 10^7$	62.53
0.5	500	$1.43842 \cdot 10^6$	$1.45897 \cdot 10^6$	1.41
	3 200	$5.78412 \cdot 10^6$	$8.51594 \cdot 10^6$	32.08
	7 200	$7.58763 \cdot 10^6$	$1.87294 \cdot 10^7$	59.49
0.4	500	901519	913243	1.28
	3 200	$3.56356 \cdot 10^6$	$5.13230 \cdot 10^6$	30.57
	7 200	$4.69754 \cdot 10^6$	$1.12186 \cdot 10^7$	58.13

Tab. 2. The values of axial C_w and radial C_r components of bearing load carrying capacity

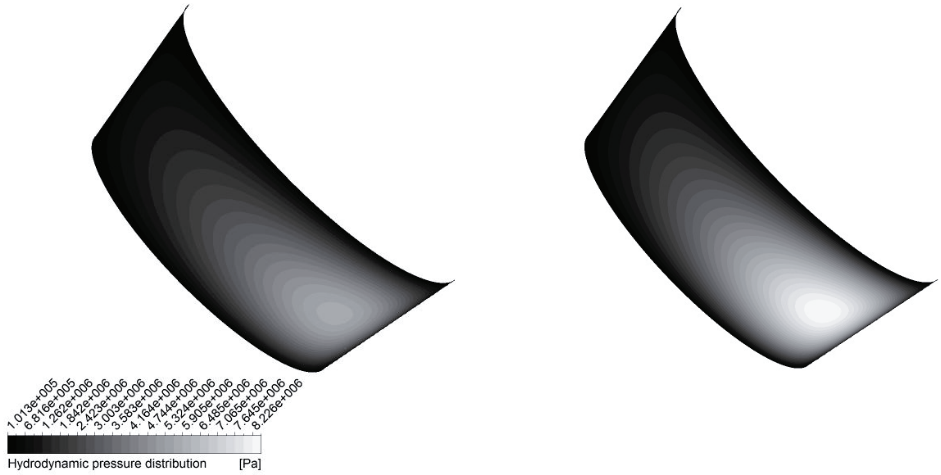
Relative eccentricity	Shaft speed	Load carrying capacity			
λ [-]	n_r [rpm]	Considering viscous heating		Isothermal flow of oil	
		C_w [N]	C_r [N]	C_w [N]	C_r [N]
0.6	500	931	4489	945	4557
	3 200	4012	19139	5859	28283
	7 200	5404	25635	12953	62660
0.5	500	607	2857	616	2901
	3 200	2672	12471	3813	17984
	7 200	3727	17449	8425	39864
0.4	500	407	1877	412	1905
	3 200	1816	8327	2550	11805
	7 200	2547	11306	5619	26110



a) *b)*

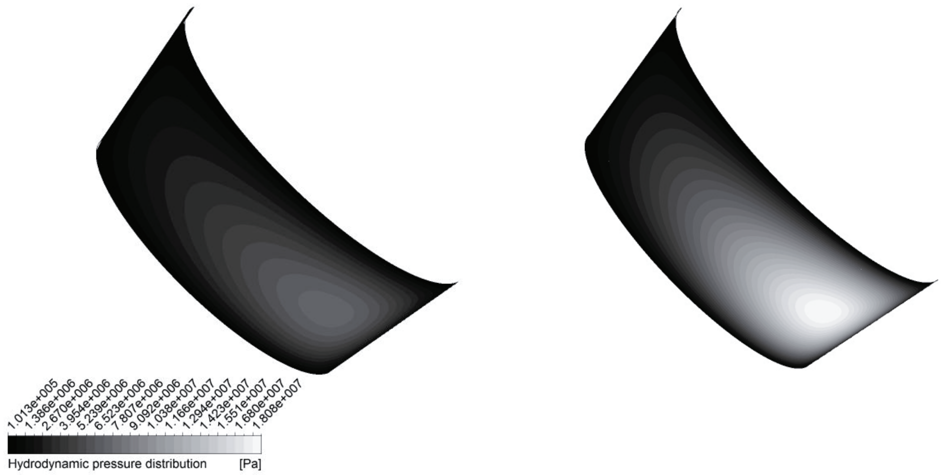
Fig. 5. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.5, n_r = 500 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication

The obtained results show that the inclusion of viscous heating in simulations of hydrodynamic lubrication of slide conical bearings has a significant impact on the derived values. The investigations show that the relative reduction of the pressure generated in the oil gap is the greater, the higher the rotational speed of the shaft and the greater the value of bearing relative eccentricity.



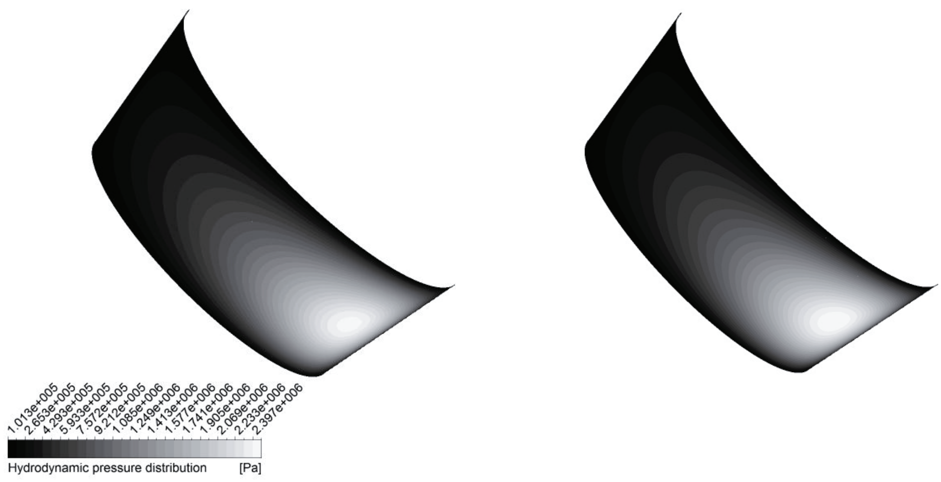
a) *b)*

Fig. 6. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.5, n_r = 3200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication



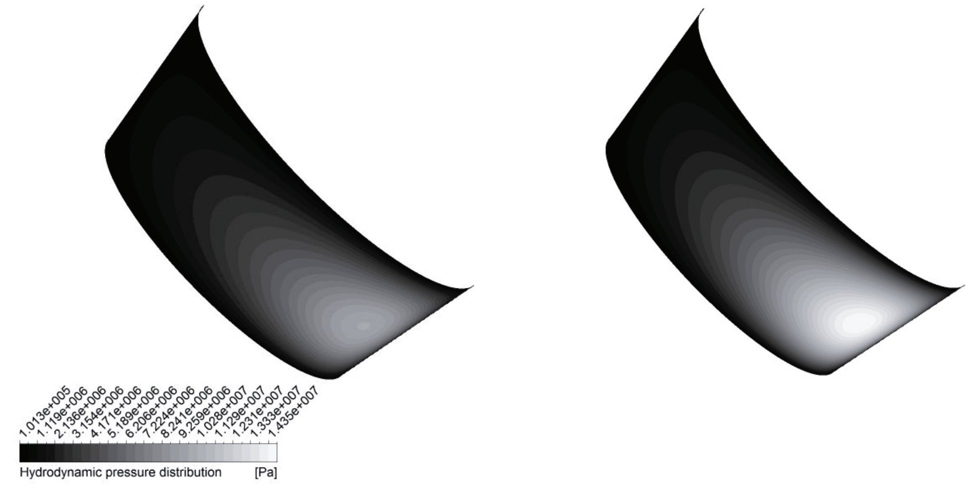
a) *B)*

Fig. 7. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.5, n_r = 7200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication



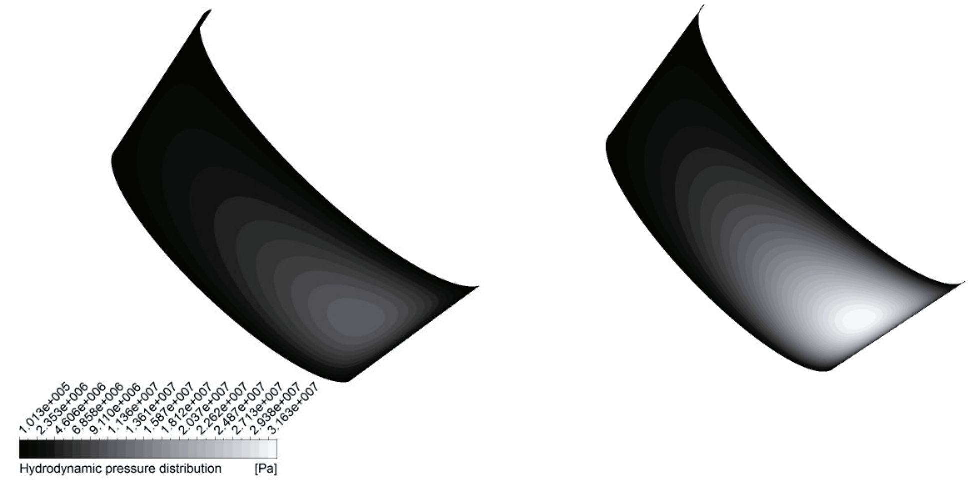
a) *b)*

Fig. 8. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.6, n_r = 500 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication



a) *b)*

Fig. 9. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.6, n_r = 3200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication



a) *b)*

Fig. 10. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.6, n_r = 7200 rpm a) lubricated with non-Newtonian oil and including viscous heating, b) non-Newtonian isothermal lubrication

In Fig. 11 is shown the temperature distribution in bearing gap for L = 25 mm and where the gap height has its lowest value, for different rotational speeds.

Conclusions

This research shows, that taking into account the effects of temperature changes caused by the viscous heating, can significantly affect the results. The obtained data show, that the inclusion of viscous heating in simulations of hydrodynamic lubrication of slide conical bearings has an important impact on the obtained values. For relatively low speeds, the changes in the pressures and the load bearing capacities are not great, however, it has been shown, that for this bearing at speeds above 7000 rpm, there can be observed a decrease in values of these parameters by more than 50 percent. This effect is compounded by increase of the value of relative eccentricity, that is, for example, by increasing the bearing load.

Taking into account the increasing number of effects in numerical simulations may significantly affect the accuracy of the results, but it should be noted that this also increases the load on the computer and the calculation time as well.

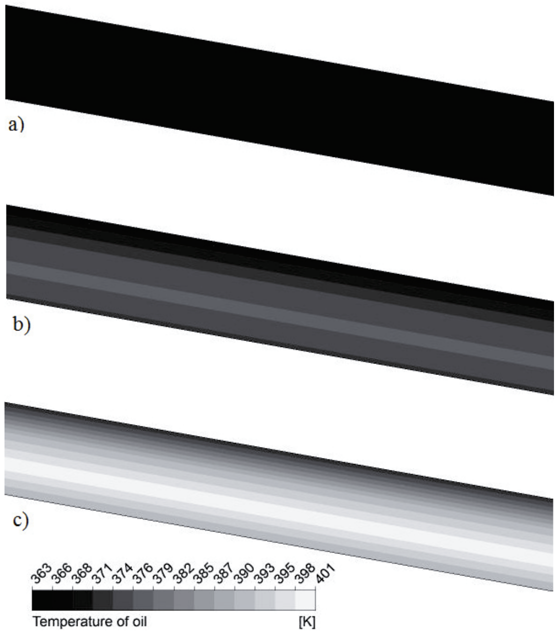


Fig. 11. Temperature distribution in slide conical bearing gap with relative eccentricity λ = 0.6, for L= 25 mm and the lowest value of gap height: a) n_r = 500 rpm, b) n_r = 3200 rpm, c) n_r = 7200 rpm

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