CFD ANALYSIS OF PRESSURE DISTRIBUTION IN SLIDE CONICAL BEARING LUBRICATED WITH NON-NEWTONIAN OIL

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Abstract

The aim of this work is to simulate and determine a hydrodynamic pressure distribution in a lubrication gap of slide conical bearing gap, assuming that the lubricating oil has non-Newtonian properties. Investigations were 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 motor oils. The calculated hydrodynamic pressure distributions were compared with the data obtained for corresponding bearings, but assuming that lubricating oil has Newtonian properties. In this research, the Gümbel boundary condition (also known as half-Sommerfeld condition) was imposed. Moreover, there was assumed: a steady-state operating conditions of a bearing, laminar, incompressible flow of lubricating oil, no slip on bearing surfaces, negligible heat conduction effect of bearing material, pressure on the side surfaces of bearing gap is equal to atmospheric pressure, no oil supply and oil outflow from bearing gap. This paper presents results for bearings with different rotational speeds and of different bearing gap heights. 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 calculated for investigated bearings.

Keywords: *slide conical bearing, CFD, non-Newtonian oil, pressure distribution*

1. Introduction

A slide conical bearing is capable of carrying longitudinal and transverse loads and also makes it possible, for bearing which carry transverse loads, to adjust the height of the lubrication gap.

This work presents the results of CFD analysis of the hydrodynamic pressure distribution in slide conical bearing lubricated with non-Newtonian oil. The commercial software ANSYS Workbench with CFD module ANSYS Fluent was used in this investigation. It was assumed that the oil shear stress varies from shear rate according to the Ostwald-de Waele relationship (power law lubricant) [3, 4], which can be written as:

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

where τ is the shear stress (Pa), $\dot{\gamma}$ is the shear rate (s^{-1}), K is the *flow consistency index* ($Pa \cdot s^n$) and n is the dimensionless *flow behaviour index*.

The coefficients for that model were determined for the experimental results obtained for motor oils presented in previous paper [1]. In this work, there was assumed, that bearings are lubricated with oil, which has Superol CC-40 properties at temperature t = 90°C. The least squares approximation method was used to determine the values of K and n indices for Ostwald-de Waele relationship. The values of measured shear stress and viscosity versus shear rate and also respectively fitted curve are presented in Fig. 1. The determined values of indices in this case are $K = 0.01147$ $Pa \cdot s^n$, $n = 0.9932$.

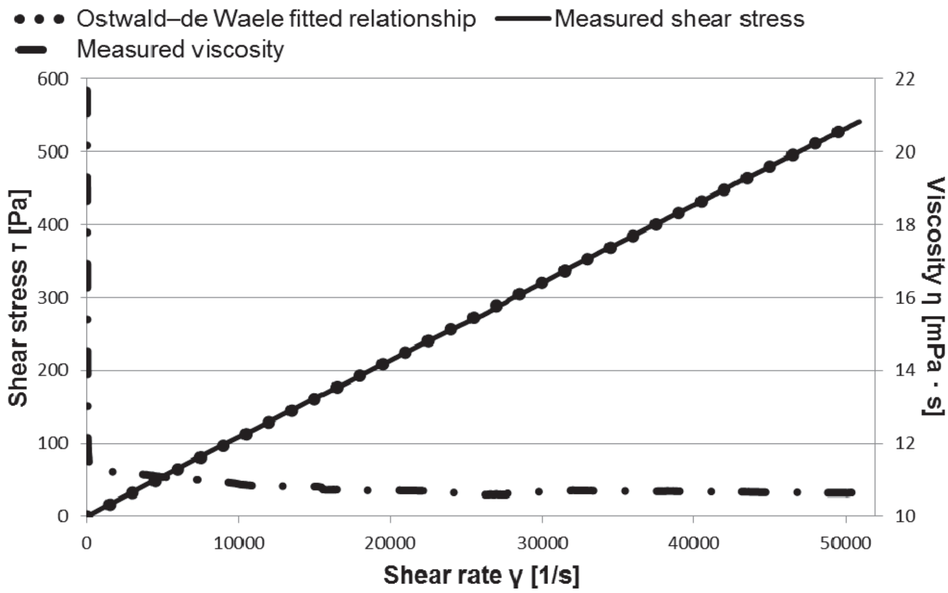

Fig. 1. Shear stress and viscosity vs. shear rate for Superol CC-40 at temperature $t = 90°C$. $K = 0.01147\ Pa·s^n$, $n = 0.9932$, $\eta_0 = 21.7\ mPa·s$, $\eta_\infty = 10.6\ mPa·s$

The obtained hydrodynamic pressure distributions were compared with the results for the corresponding bearings, but with the assumption, that the lubricating oil in these bearing has Newtonian properties and value of viscosity for that oil is equal to that value at lowest measured shear rate (i.e. η_0). In this research calculations were performed for the Gümbel [2, 5] (also known as half-Sommerfeld) condition. Other assumptions adopted in the simulations are: a steady-state operating conditions of a bearing, laminar, incompressible flow of lubricating oil, no slip on bearing surfaces, negligible heat conduction effect of bearing sleeve and shaft material, process of lubrication is isothermal, vibrations of bearing shaft are negligible, pressure on the side surfaces of bearing gap is equal to atmospheric pressure, no oil supply and oil outflow from bearing gap. This paper presents results for bearings with different rotational speeds and different heights of bearing gap. Calculations were made for bearings without misalignment, i.e. where the cone generating line of bearing shaft is parallel to the cone generating line of bearing sleeve.

2. Results

The geometry of investigated bearing is shown in Fig. 2. Studies were performed for bearing with a length of $L = 50$ mm. The radius of shaft at lowest cross-section of the bearing is $R = 50$ mm. The radial clearance is $\varepsilon = R' - R = 0.025$ mm, where R' is a radius of bearing sleeve.

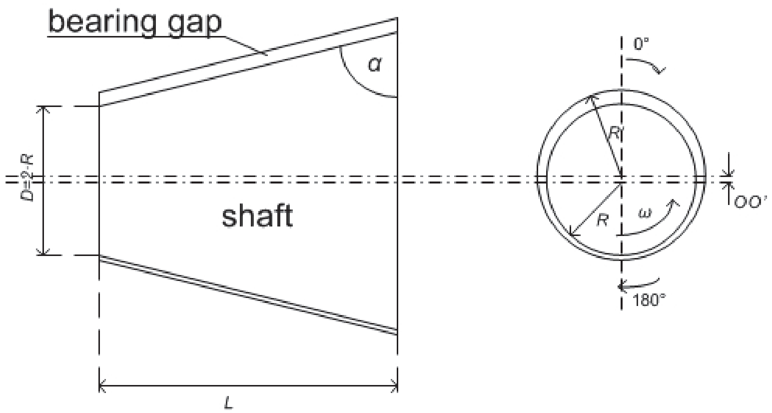

Fig. 2. The geometry of the investigated bearing

The angle is $\alpha = 80°$. The relative eccentricity ratio λ is defined as [2]:

$$\lambda = \frac{OO'}{\varepsilon}, \tag{2}$$

where OO' is the bearing eccentricity.

The geometry, meshing, setup solution and post-processing of each case, was made with the use of ANSYS Workbench software with CFD Fluent module. There was considered the Gümbel boundary conditions, therefore only half of bearing lubrication gap (i.e. from 0° to 180°) was investigated. The pressure in the second half of lubrication gap, according to Gümbel conditions, is equal to atmospheric pressure. The constant viscosity values η_0 for Newtonian lubricants, and values of K, n, η_0, η_∞ for power-law lubricants, were entered during Fluent solution setup.

The contour graphs present hydrodynamic pressure distribution in investigated bearings in the absolute pressure scale. Fig. 3-5 presents results for the bearing with the relative eccentricity $\lambda = 0.4$, but with different rotational speeds of bearing shaft. In Fig. 3 is shown hydrodynamic pressure distribution in lubrication gap of the bearing, which operates at the speed $n_r = 500$ rpm. Fig. 3a shows results for bearing lubricated with Non-Newtonian which behaves as Superol CC-40 at 90°C, while Fig. 3b is hydrodynamic pressure distribution of corresponding bearing, but lubricated with Newtonian oil, at $\eta = 21.7$ mPa·s. Results for bearings operating at $n_r = 3200$ rpm and at $n_r = 7200$ rpm are presented in an analogous manner in Fig. 4a, 4b, 5a and 5b.

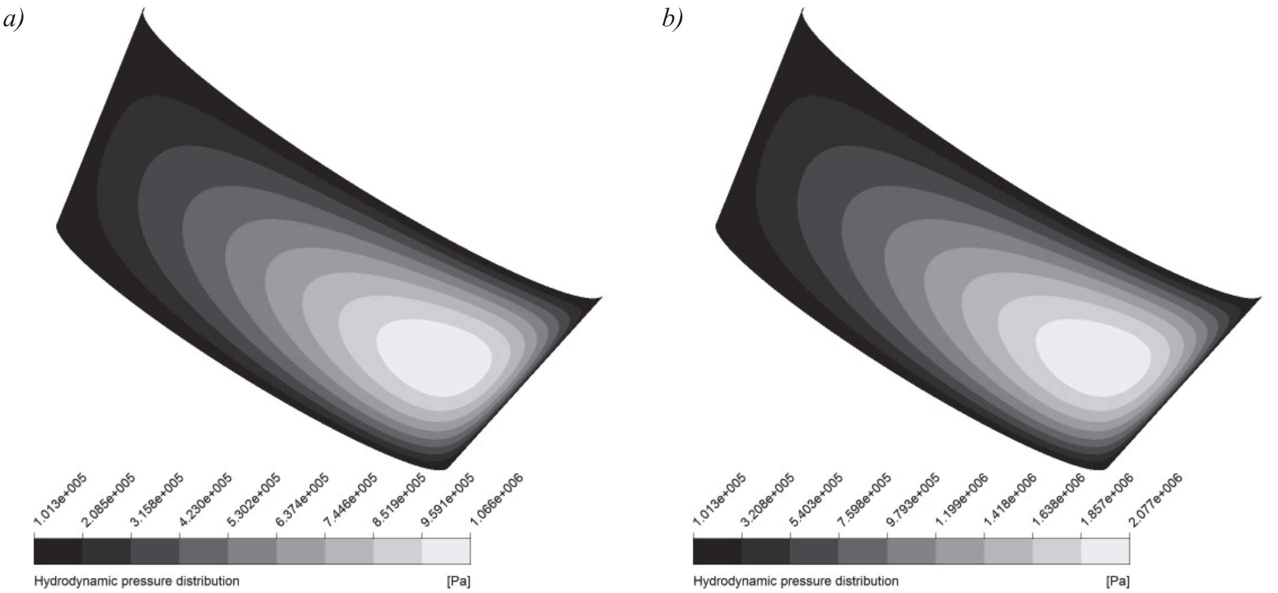

Fig. 3. Hydrodynamic pressure distribution in conical slide bearing; $\lambda = 0.4$, $n_r = 500$ rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

One can observe, that the values of hydrodynamic pressure in the case, where bearing was lubricated with Newtonian oil, were grater, than for corresponding bearing, but lubricated with non-Newtonian oil. This is due to a decrease of lubricating oil viscosity value for increasing values of shear rates. Tab. 1 contain comparison of values of maximum hydrodynamic pressure generated in investigated bearing lubrication gap, in the absolute scale of pressure, and also the values of longitudinal C_w and transverse C_r components of bearing load carrying capacity, for bearing with relative eccentricity $\lambda = 0.4$.

The performed calculations show, that the increased values of shear rates, which occur during the considered bearing lubrication, cause the major decreases of dynamic viscosity values, hence the bearing load carrying capacity decreases.

The hydrodynamic pressure distributions for bearings with $\lambda = 0.5$, lubricated with non-Newtonian oil are presented in Fig. 6a, 7a, 8a, while for bearings lubricated with Newtonian oil are shown

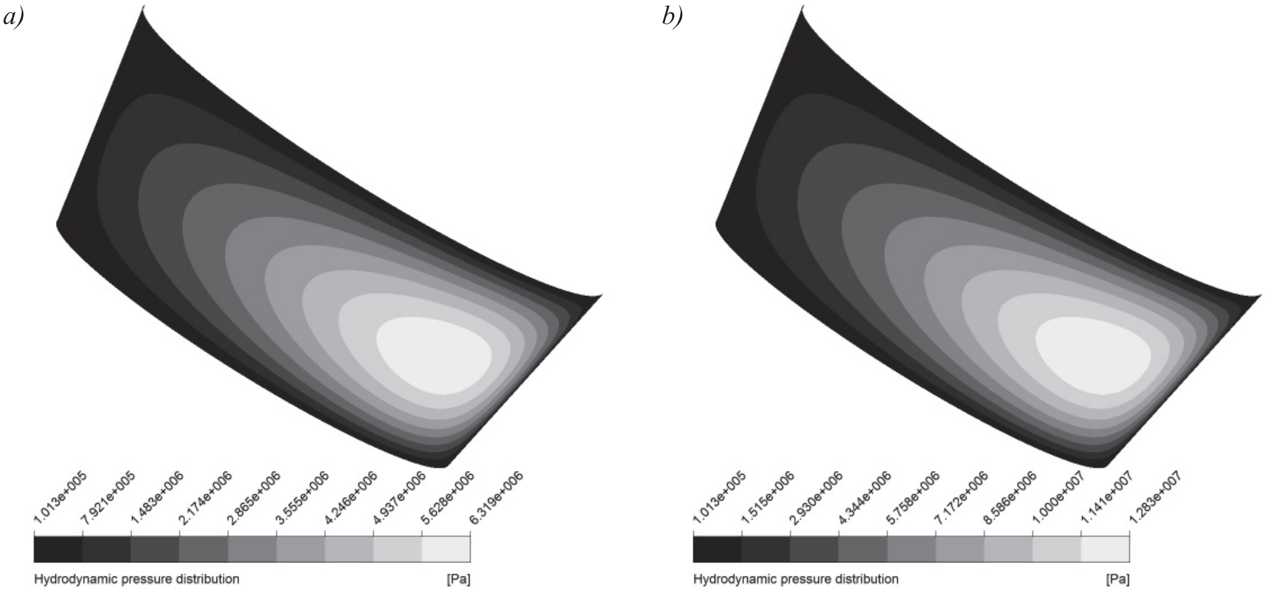

Fig. 4. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.4, n_r = 3200 rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

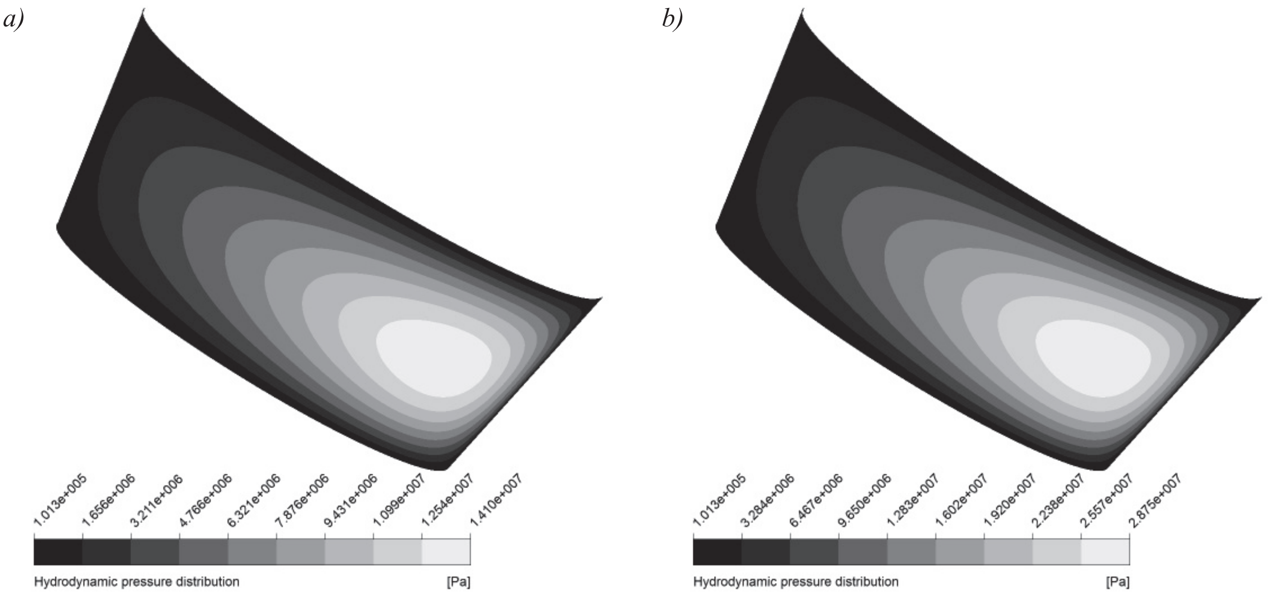

Fig. 5. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.4, n_r = 7200 rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

Tab. 1. The maximum value of hydrodynamic pressure in the absolute scale and the values of longitudinal C_w and transverse C_r components of bearing load carrying capacity, for bearing with eccentricity ratio λ = 0.4

Speed, n_r [rpm]	Maximum pressure in bearing gap, p_{max} [Pa]		Load carrying capacities			
	non-Newtonian oil	Newtonian oil	non-Newtonian oil		Newtonian oil	
			C_w [N]	C_r [N]	C_w [N]	C_r [N]
500	$1.066 \cdot 10^6$	$2.077 \cdot 10^6$	488	2 254	998	4 615
3 200	$6.319 \cdot 10^6$	$1.283 \cdot 10^7$	3 138	14 531	6 428	29 739
7 200	$1.410 \cdot 10^7$	$2.875 \cdot 10^7$	7 048	32 729	14 456	61 965

in Fig. 6b, 7b and 8b. Tab. 2 contain values of maximum hydrodynamic pressure in lubrication gap, in the absolute scale of pressure, and also the values of longitudinal C_w and transverse C_r components of bearing load carrying capacity, for bearing with eccentricity ratio λ = 0.5.

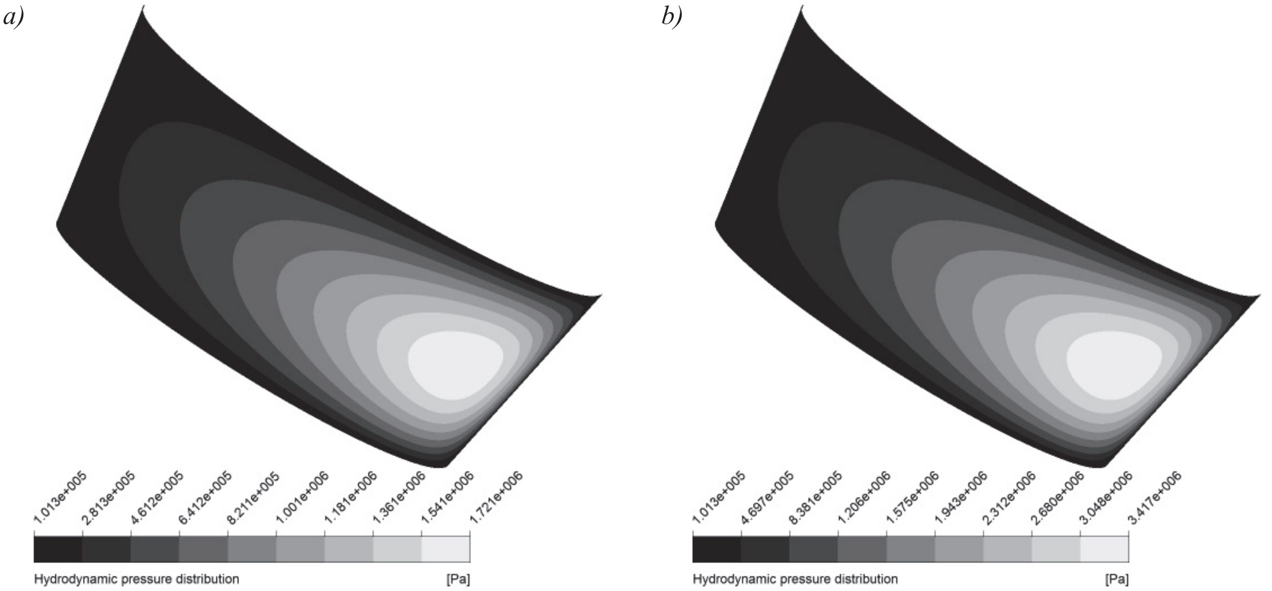

Fig. 6. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.5, n_r = 500 rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

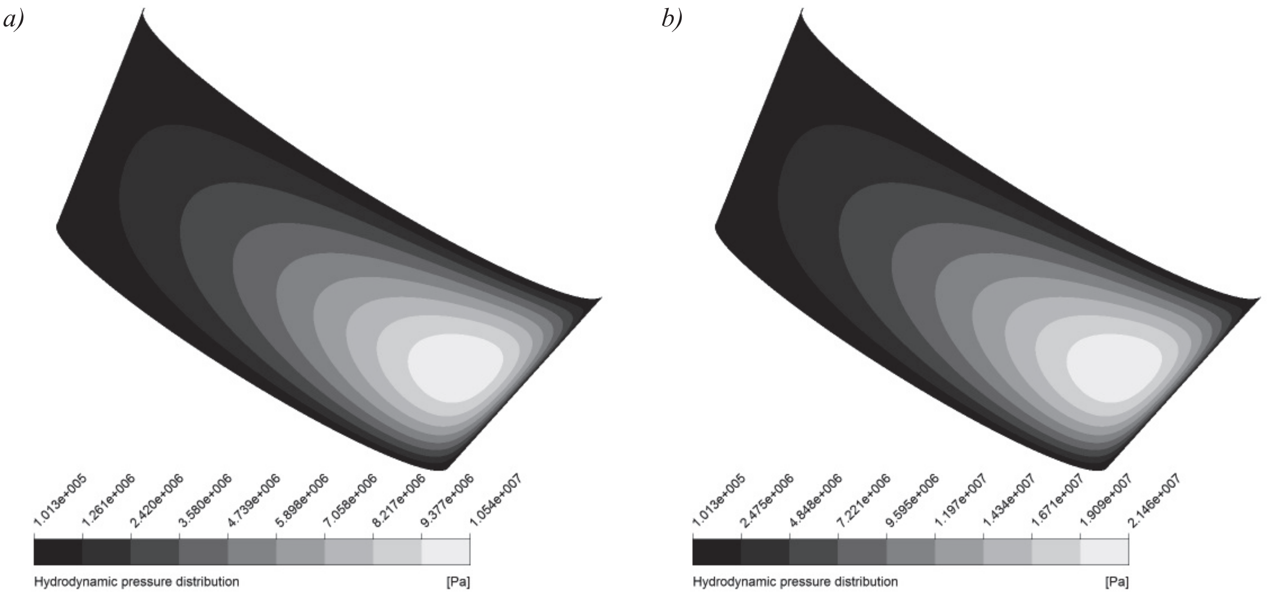

Fig. 7. Hydrodynamic pressure distribution in slide conical bearing; λ = 0.5, n_r = 3200 rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

Tab. 2. The maximum value of hydrodynamic pressure in the absolute scale and the values of longitudinal C_w and transverse C_r components of bearing load carrying capacity, for bearing with eccentricity ratio λ = 0.5

Speed, n_r [rpm]	Maximum pressure in bearing gap, p_{max} [Pa]		Load carrying capacity			
	non-Newtonian oil	Newtonian oil	non-Newtonian oil		Newtonian oil	
			Cw [N]	Cr [N]	Cw [N]	Cr [N]
500	$1.721 \cdot 10^6$	$3.417 \cdot 10^6$	730	3 443	1 495	6 147
3 200	$1.054 \cdot 10^7$	$2.146 \cdot 10^7$	4 702	22 198	9 628	45 427
7 200	$2.359 \cdot 10^7$	$4.818 \cdot 10^7$	10 571	50 006	21 661	102 289

In Fig. 9a, 10a, and 11a are presented results for bearings with relative eccentricity ratio λ = 0.6, lubricated with non-Newtonian oil, while in Fig. 9b, 10b and 11b results for the same bearings, but lubricated with Newtonian oil.

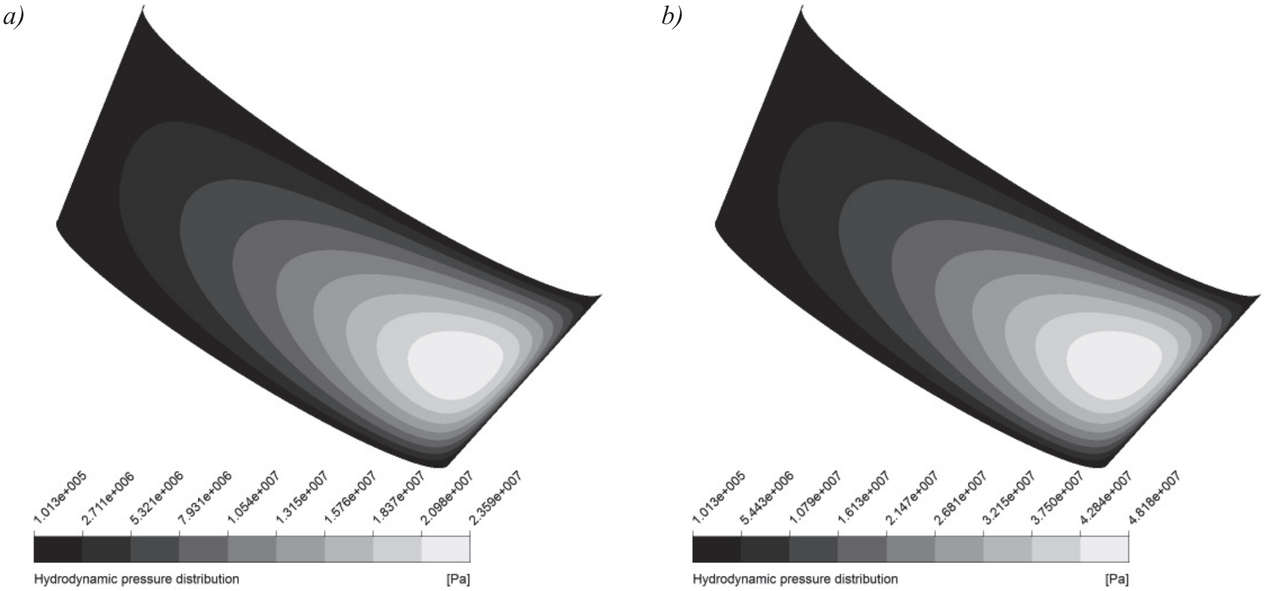

Fig. 8. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.5$, $n_r = 7200$ rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

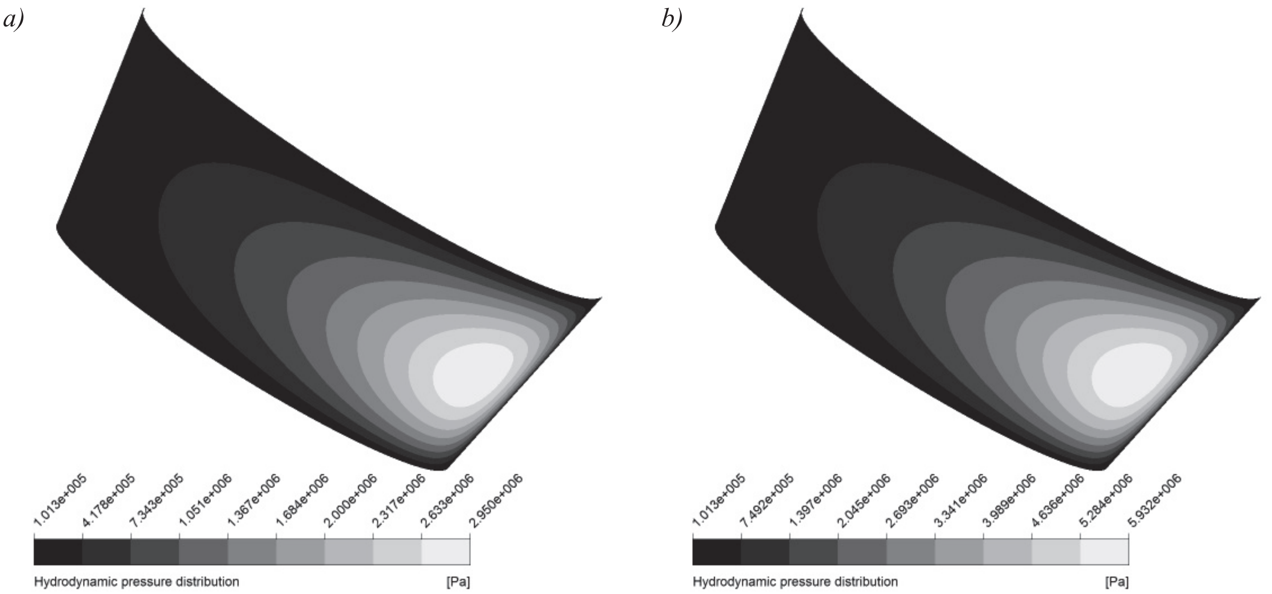

Fig. 9. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.6$, $n_r = 500$ rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

In Tab. 3 are shown values of maximum hydrodynamic pressure of lubricating oil and also the values of longitudinal C_w and transverse C_r components of bearing lift capacity, for bearing with relative eccentricity ratio $\lambda = 0.6$.

Tab. 3. The maximum value of hydrodynamic pressure in the absolute scale and the values of longitudinal C_w and transverse C_r components of bearing load carrying capacity, for bearing with eccentricity ratio $\lambda = 0.6$

Speed, n_r [rpm]	Maximum pressure in bearing gap $p_{\max}$ [Pa]		Load carrying capacity			
	non-Newtonian oil	Newtonian oil	non-Newtonian oil		Newtonian oil	
			Cw [N]	Cr [N]	Cw [N]	Cr [N]
500	$2.950\cdot10^6$	$5.932\cdot10^6$	1 124	5 428	2 301	11 113
3 200	$1.845\cdot10^7$	$3.767\cdot10^7$	7 240	34 994	14 824	71 615
7 200	$4.141\cdot10^7$	$8.466\cdot10^7$	16 289	78 831	33 359	161 253

a) b)

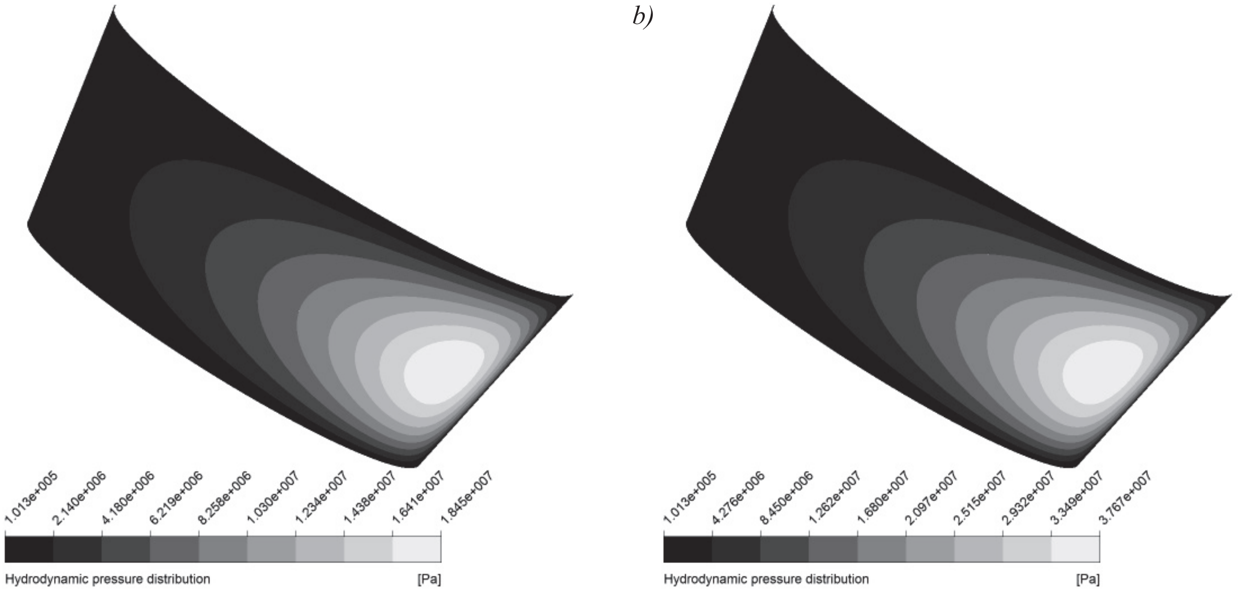

Fig. 10. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.6$, $n_r = 3200$ rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

a) b)

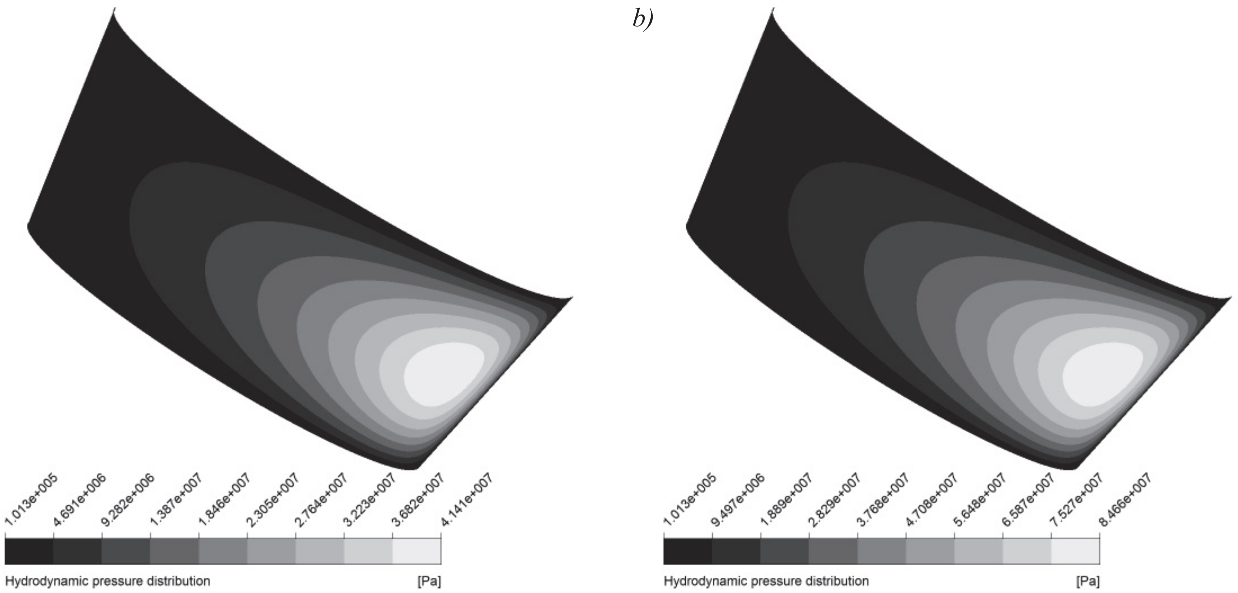

Fig. 11. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.6$, $n_r = 7200$ rpm a) lubricated with non-Newtonian oil, b) lubricated with Newtonian oil

Due to presented results, there can be observed, that consideration of non-Newtonian properties of lubricating oil, can cause significant changes in values of hydrodynamic pressure and according to that values of bearing lift capacities.

Conclusions

This paper presents the results of CFD analysis of the hydrodynamic pressure distribution in slide conical bearing lubricated with non-Newtonian oil and comparison with corresponding bearing, but lubricated with Newtonian oil. It was shown, that taking into account changes of lubricating oil viscosity as a function of shear rate, could greatly affect the value of hydrodynamic pressure of oil in slide conical bearing lubrication gap.

In this simulation was assumed, that shear stress in lubricating oil depends on shear rate accordingly to the Ostwald-de Waele relationship. The coefficients of that model were determined by fitting the curve described by this relation, to experimental data, obtained for mineral motor oil. It should be noted, that flow curve adopted for this simulation, was determined only in shear rate range from 10 to 50 000 1/s. The maximum η_0 and minimum η_∞ viscosity values correspond to these shear rates. If the share rate of lubricant exceeded these limits, then the used software substitutes the value of viscosity by the appropriate limit of viscosity value. Moreover there were imposed some of simplifying assumptions, e.g. laminar and isothermal flow of lubricant. However, even with occurring in investigated bearing share rates and conditions, one can observe, that considering non-Newtonian properties of lubricating oil cause significant impact on slide conical bearing operating parameters.

The adopted method allows in a simple way, without going into the mathematical aspects of continuity and momentum equations, to obtain accurate data. Further research will take into account such aspects as the effect of pressure and temperature on the properties of the lubricating oil and also lubrication with the ferro-oils.

References

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