

CFD ANALYSIS OF HYDRODYNAMIC LUBRICATION OF SLIDE CONICAL BEARING WITH CONSIDERATION OF THE BEARING SHAFT AND SLEEVE SURFACE ROUGHNESS

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Abstract

In this work is shown the result of CFD simulation of hydrodynamic conical bearing lubrication with consideration of the effect of the bearing shaft and sleeve surface roughness. The oil flow in a bearing lubrication gap largely depend on the condition of the cooperating surfaces of a bearing. Surface irregularities are formed already at the manufacturing process and furthermore the quality of the surface may change during operation of a bearing. In this work, as a parameter describing surface condition, the K_s roughness height parameter was taken (i.e. sand-grain roughness height). The hydrodynamic pressure distribution in lubrication gaps of investigated bearings were calculated by using the commercial CFD software ANSYS Academic Research for fluid flow phenomenon (Fluent). Calculations were conducted for bearings without misalignment. 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 bearings have smooth surfaces and there is no slip on surfaces. This paper presents results for bearings with different rotational speeds and of different bearing gap heights.

Keywords: conical bearing, surface roughness, hydrodynamic lubrication, CFD simulation, non-Newtonian oil, pressure distribution, sand-grain roughness

1. Introduction

A consideration of the influence of roughness on the parameters of co-operating surfaces requires the use of statistical models. In the case of hydrodynamic lubrication of slide bearings, analytical considerations can be performed, by applying the expectancy operator and deriving the stochastic Reynolds-type equation [5]. The presently available CFD software allows simulating wide range of processes and taking into account the impact of many effects. In this work is shown the result of CFD simulation of hydrodynamic conical bearing lubrication with consideration of the effect of the bearing shaft and sleeve surface roughness. The hydrodynamic pressure distribution in lubrication gaps of investigated bearings were calculated by using the commercial CFD software ANSYS Academic Research for fluid flow phenomenon (Fluent). In this software, for turbulent flows, roughness can be modelled as uniform sand-grain roughness described by roughness height parameter K_s . The conversion of this parameter value to the value of arithmetic average of surface roughness parameter R_a can be conducted by using the algorithm described in paper [1], where:

$$K_s = 5.863 \cdot R_a . \quad (1)$$

For determining the effect of roughness, the hydrodynamic pressure distributions and the axial and radial components of load carrying capacity were compared for bearings with different values of relative eccentricity ratios, different rotational speeds and different surface roughness height parameter K_s .

2. Results

In the simulations, there was assumed: the incompressible flow of lubricating oil, the vibrations of bearing shaft are negligible, the pressure at the side surfaces of the bearing gap is equal to the atmospheric pressure (i.e. 101 325 Pa), a negligible heat conduction effect of the bearing sleeve and shaft material, the process of lubrication is isothermal. The calculations concerned the Gumbel boundary condition. [6, 13]. Fig. 1 shows the geometry of investigated bearing. There is no misalignment between the axis of bearing sleeve and the axis of shaft (the cone generating line of the bearing shaft is parallel to the cone generating line of the bearing sleeve). The bearing length is $L = 50$ mm and the radius of the shaft at lowest cross-section is $R = 50$ mm. The radial clearance is $\varepsilon = R' - R = 0.025$ mm (R' is a radius of bearing sleeve at the lowest cross-section) and the $\alpha = 80^\circ$.

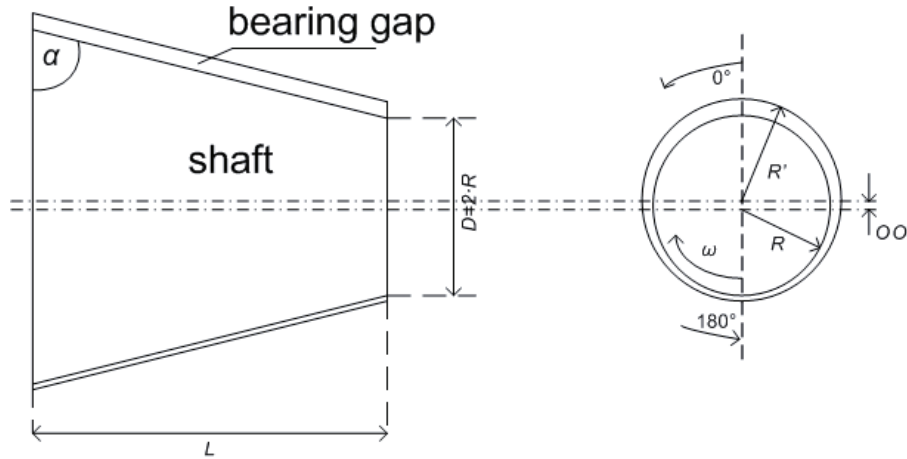


Fig. 1. The investigated bearing geometry

The relative eccentricity ratio λ is defined as [5]:

$$\lambda = \frac{OO'}{\varepsilon}, \quad (2)$$

where OO' is the bearing eccentricity.

There was assumed, that lubricating oil is the power-law fluid described by the Ostwald-de Waele relationship [9, 10]:

$$\tau = K \cdot \dot{\gamma}^n, \quad (3)$$

where τ [Pa] is shear stress, $\dot{\gamma}$ [s^{-1}] is shear rate, K [$Pa \cdot s^n$] is the *flow consistency index* in $Pa \cdot s^n$ and n [-] is the flow behaviour index. The coefficients for that model were determined with the least squares approximation method and fitting the curve described by the Ostwald-de Waele equation to the experimental data obtained for engine oil Shell Helix Ultra AV-L at $t = 60$ [$^\circ C$], which was presented in the previous paper [3]. The calculated values are $K = 0.03419$ [$Pa \cdot s^n$] and $n = 0.09792$.

The pre-processing, solution and post-processing were made with the ANSYS Workbench software with CFD Fluent module. The prepared mesh was with about 4 500 000 of nodes. The solutions parameters were: the pressure based SIMPLE algorithm (Green-Gaus node based), the pressure - second order and the momentum - second order upwind. The calculations were made for bearings with different K_s values of roughness height parameter (0.1 μm , 0.2 μm , 0.4 μm) and were compared with the data obtained for bearings with smooth surfaces of bearing shaft and sleeve (i.e. while $K_s = 0.0$ μm). In simulations, the k-omega model was adopted.

Table 1 show the comparison of the maximum values of lubricating oil absolute pressure in the bearing lubrication gap for different relative eccentricities and rotational speeds.

Tab. 1. The maximum value of hydrodynamic pressure in the absolute scale for different values of the surface roughness parameter K_s

| Relative eccentricity | Shaft speed | Maximum pressure value in the bearing gap p_{max} (absolute) [Pa] | | | | |
|-----------------------|-------------|---|---|---|---|----------------------|
| | | The bearing with smooth surfaces (no slip) | The bearing with rough surfaces $K_s = 0.1 \mu\text{m}$ | The bearing with rough surfaces $K_s = 0.2 \mu\text{m}$ | The bearing with rough surfaces $K_s = 0.4 \mu\text{m}$ | |
| $\lambda [-]$ | n_r [rpm] | | | | | |
| | | 500 | $1.89169 \cdot 10^6$ | $1.86794 \cdot 10^6$ | $1.84483 \cdot 10^6$ | $1.80034 \cdot 10^6$ |
| | | 3 200 | $1.08244 \cdot 10^7$ | $1.06821 \cdot 10^7$ | $1.05437 \cdot 10^7$ | $1.01772 \cdot 10^7$ |
| 0.4 | 7 200 | | $2.34878 \cdot 10^7$ | $2.31781 \cdot 10^7$ | $2.28764 \cdot 10^7$ | $2.25431 \cdot 10^7$ |
| | | 500 | $5.31425 \cdot 10^6$ | $5.21765 \cdot 10^6$ | $5.12461 \cdot 10^6$ | $4.94837 \cdot 10^6$ |
| | | 3 200 | $3.13234 \cdot 10^7$ | $3.07451 \cdot 10^7$ | $3.01880 \cdot 10^7$ | $2.91328 \cdot 10^7$ |
| 0.6 | 7 200 | | $6.81959 \cdot 10^7$ | $6.69354 \cdot 10^7$ | $6.57214 \cdot 10^7$ | $6.34221 \cdot 10^7$ |

The obtained values show, that increasing the height roughness causes the slight decrease in maximum value of pressure generated in bearing lubrication gap. How this affects the axial C_w and radial C_r load carrying capacities is shown in the Tab. 2., where are listed those values for bearing with smooth surfaces along with the relative decrease of load carrying capacities due to increasing value of surface roughness.

Tab. 2. The relative decrease of axial C_w and radial C_r components of bearing load carrying capacity depending on roughness height of the bearing sleeve and shaft surfaces

| Relative eccentricity | Shaft speed | Load carrying capacity changes | | | | | | | |
|-----------------------|-------------|--|--------|---|-------|-------------------------|-------|-------------------------|-------|
| | | The bearing with smooth surfaces (no slip) [N] | | The bearing with rough surfaces – the relative decrease of load carrying capacity [%] | | | | | |
| $\lambda [-]$ | n_r [rpm] | C_w | C_r | $K_s = 0.1 \mu\text{m}$ | | $K_s = 0.2 \mu\text{m}$ | | $K_s = 0.4 \mu\text{m}$ | |
| | | | | C_w | C_r | C_w | C_r | C_w | C_r |
| 0.4 | 500 | 912 | 4208 | 1.21 | 1.21 | 2.39 | 2.42 | 4.61 | 4,71 |
| | 3 200 | 5456 | 25209 | 1.21 | 1.22 | 2.37 | 2.40 | 4.62 | 4.69 |
| | 7 200 | 11883 | 55002 | 1.20 | 1.21 | 2.36 | 2.40 | 4.60 | 4.68 |
| 0.6 | 500 | 2080 | 10019 | 1.59 | 1.62 | 3.08 | 3.17 | 5.96 | 6.14 |
| | 3 200 | 12453 | 60024 | 1.57 | 1.61 | 3.08 | 3.17 | 5.96 | 6.14 |
| | 7 200 | 27150 | 130976 | 1.57 | 1.61 | 3.08 | 3.17 | 5.95 | 6.13 |

Except that the relative decrease of load bearing capacities C_w and C_r depends on the roughness height, it can be observed, that also the relative eccentricity also influences those values. The greater value of relative eccentricity represents the lower value of bearing lubrication gap and the roughness height has relatively greater importance. For the examined values of rotational speeds, it can be noticed, that the rotational speed changes in terms of roughness, had no effect on load carrying capacities relative decrease.

The contour graphs in Fig. 2 and 3 present the hydrodynamic pressure distributions in the absolute pressure scale for bearing with rotational shaft speed $n_r = 7200$ rpm. In Fig. 2 is shown the hydrodynamic pressure distribution for the conical slide bearing with $\lambda = 0.4$ while in Fig. 3 with $\lambda = 0.6$. Fig. 2a and 3a show results for bearing with smooth surfaces, Fig. 2b and 3b show results for bearing with rough surfaces and $K_s = 0.1 \mu\text{m}$, Fig. 2c and 3c are for $K_s = 0.2 \mu\text{m}$ and Fig. 2d and 3d are for $K_s = 0.4 \mu\text{m}$. The white lines are just contours of some specific pressures – isobars (each white line is for different pressure – there is no link with the scale).

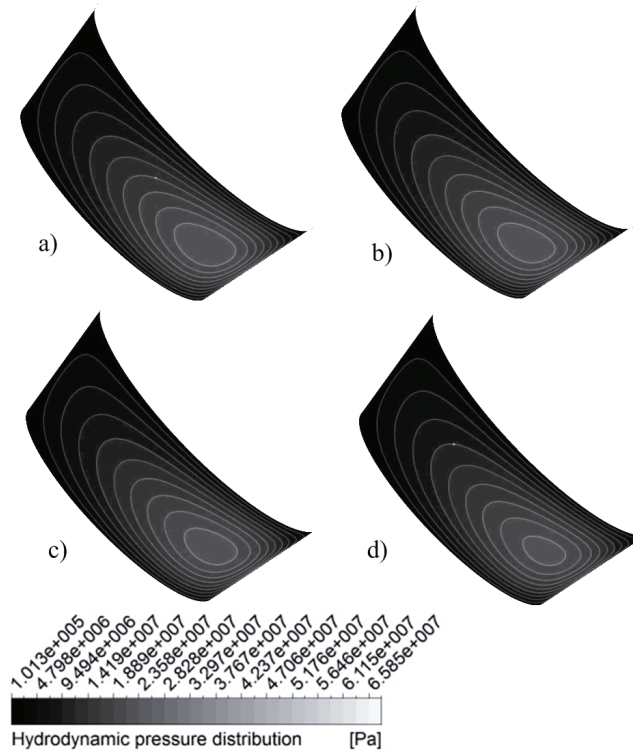


Fig. 2. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.4$, $n_r = 7200$ rpm a) with smooth surfaces, b) with shaft and sleeve surface roughness parameter $K_s = 0.1 \mu\text{m}$, c) with shaft and sleeve surface roughness parameter $K_s = 0.2 \mu\text{m}$, d) with shaft and sleeve surface roughness parameter $K_s = 0.4 \mu\text{m}$

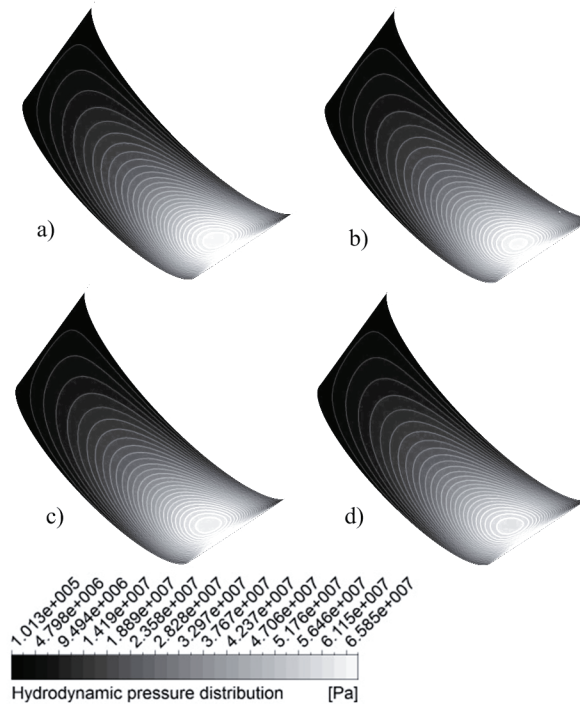


Fig. 3. Hydrodynamic pressure distribution in slide conical bearing; $\lambda = 0.6$, $n_r = 7200$ rpm a) with smooth surfaces, b) with shaft and sleeve surface roughness parameter $K_s = 0.1 \mu\text{m}$, c) with shaft and sleeve surface roughness parameter $K_s = 0.2 \mu\text{m}$, d) with shaft and sleeve surface roughness parameter $K_s = 0.4 \mu\text{m}$

The results of this simulation show, that the inclusion of the bearing shaft and sleeve surface roughness affects bearing performance. The maximum relative reduction in the load bearing capacity was of the order of 6% for the bearing with $\lambda = 0.4$ and $n_r = 7200$ rpm while sand-graing

roughness parameter was $K_s = 0.4 \mu\text{m}$. Substituting this value into the Eq. 1, we receive the equivalent value of surface roughness parameter $R_a \approx 0.068 \mu\text{m}$. In paper [7] for slide journal bearings and in paper [8] for slide journal micro-bearings used in computer fans is shown, that measured values of R_a for that bearings varies in the range from $0.030 \mu\text{m}$ to $0.500 \mu\text{m}$. Moreover, some of bearings surfaces presented in these papers, have grooves and micro-grooves, therefore, in future work is planned, to conduct CFD simulation with investigation of the impact of manufactured grooves and micro-grooves on the operating parameters of a slide bearing.

3. Discussion

In this paper are presented the results of CFD simulation of hydrodynamic lubrication of slide conical bearings, with the consideration of the bearing shaft and sleeve surface roughness effect. The conducted considerations show, that the increase of the surface roughness height parameter K_s value causes the decrease in values of axial and radial bearing load carrying capacities. Despite the fact, that in this simulation, the surface roughness was modelled as the uniform sand-grain roughness, this gave some vision on the effect of roughness on slide conical bearing operating parameters. There are some studies [2] showing that with increasing roughness values, the value of load carrying capacity also increases due to the lower values of bearing gap height. There are also some papers [11] describing decreasing values of load carrying capacity according to roughness height (e.g. cases, where the metallic contact can occur). In paper [12] is shown, that changes of load carrying capacity depends on how the roughness is modelled (i.e. positively or negatively skewed). There is planned to investigate the effect of roughness on the friction force of slide conical bearings by applying the CFD simulations. Moreover, manufacturers produce bearings with surfaces covered with grooves and there are some attempts, to determine the influence of grooves and micro-grooves on slide bearings operating parameters, also by using the CFD software.

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