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OPERATING PARAMETERS OF SLIDE MICRO-BEARINGS WITH CONSIDERATION OF OIL TEMPERATURE CHANGES AND MICRO-GROOVES ON SLEEVE SURFACE

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Abstract

In this paper author presents results of numerical calculations of hydrodynamic pressure distribution in bearing gap, load-carrying capacity, friction force and friction coefficient of slide micro-bearing considering the influence of lubricating oil temperature changes and also taking into account the influence of micro-grooves, which occur on sleeve internal surface. The micro-grooves on that surface are in longitudinal direction. The equation, which describes a bearing gap with micro-grooves on sleeve surface, was adopted from prof. K Wierzcholski's investigations.

In very thin gap height of cylindrical micro-bearings, large gradients of temperature can be observed. This causes significant changes of oil dynamic viscosity in the gap height direction. According to this, oil flow velocity, friction forces, and a hydrodynamic pressure during the micro-bearing operation are changing. Up to now the influence of temperature on oil viscosity changes and due to this, on hydrodynamic pressure and on load carrying capacity in cylindrical micro-bearing gap in numerical way were not considered yet.

The numerical calculations were performed with the use of Mathcad 14. The finite differences method and own computational procedures were implemented. The calculations were begun by solving the Reynolds' equation, assuming, that the dynamic viscosity is constant. After calculating the hydrodynamic pressure distribution, the temperature distribution in lubricating oil was determined. The obtained function of temperature was used to describe the viscosity changes with temperature. Next step involved determining the hydrodynamic pressure distribution taking into account the viscosity dependence on temperature, and then new distribution of temperature and again new values of viscosity were calculated. Calculations were repeated until assumed convergence and accuracy were reached.

The friction force depends on pressure gradient and rotational motion of bearing journal. Part of friction force, which resulting from the pressure gradient, is determined for the area, where the oil film occurs, i.e. from φ_p to φ_k. Part of friction force, which is related to journal motion, is determined for full wrap angle, i.e. from 0 to 2π. The results were presented in the form of graphs, for eccentricity ratio λ from 0.1 to 0.9, for dimensionless length of the bearing L_1=1/4. In numerical calculations were used the theoretical considerations and solutions presented in papers of K. Wierzcholski and A. Miszczak.

***Keywords**: HDD micro-bearing, micro-grooved surfaces, oil viscosity changes in gap height, hydrodynamic pressure, friction forces, load carrying capacity*

1. Introduction

The issue of hydrodynamic lubrication in case of classical slide bearings lubricated with classical Newtonian oil has already been researched thoroughly [1, 2, 7, 9, 10]. The analytic and numerical calculations are made without consideration of the viscosity change and the pressure on the lubricating layer thickness. In case of lubricating the slide bearings with the non-Newtonian fluids such like Rivlin-Ericksen fluids, the calculations have to be made with consideration of the pressure on the lubricating layer thickness. The newest research on the slide micro-bearings lubrication are made with consideration of the change of viscosity on the lubricating layer thickness that results from the influence of the adhesive force and the changes of the lubricating factor temperature, [12]. Additionally, in case of slide micro-bearings the slide surfaces with micro-grooves are used. The micro-grooves used on the surface of the bearing sleeves are usually in herringbone shape [8, 14]. The use of micro-grooves in circumferential or longitudinal directions is also possible. The surface topography of the slide micro bearing with the micro-grooves in herringbone shape is showed on the Fig. 1.

The aim of this work is to determine the numerical distribution of the hydrodynamic pressure, capacity force, friction force and friction factor with consideration of the viscosity changes on the lubricating layer thickness and of the micro-grooves on the internal surface of the sleeve.

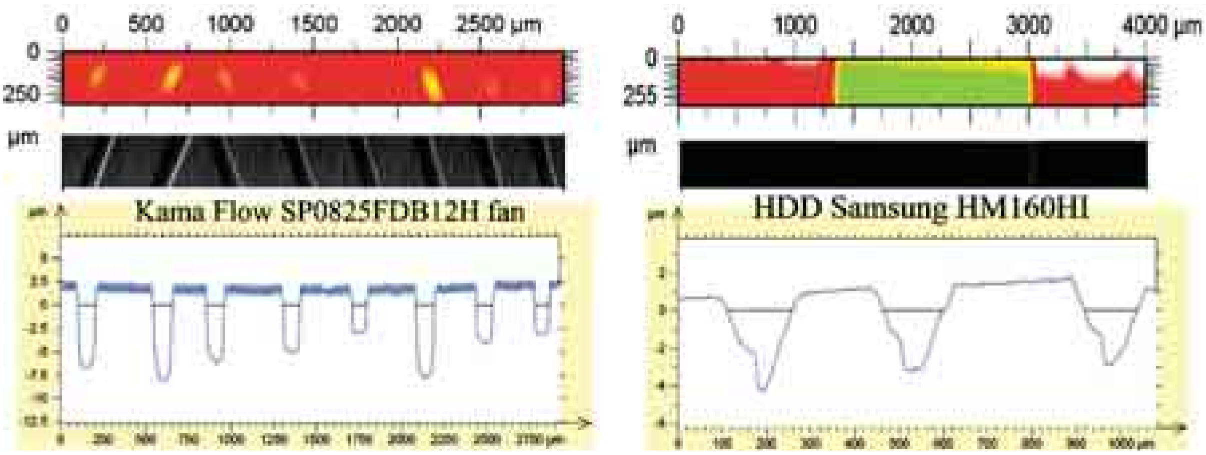


Fig. 1. The topography of micro-bearing sleeve section with the micro-grooves on surface

In numerical calculations, the author assumed the oil clearance height with the consideration of the micro-grooves in circumferential or longitudinal directions on the internal surface of the sleeve in the following equation [14]:

$$\varepsilon_\Sigma(\varphi) = \varepsilon\left\{1+\lambda\cos\varphi - \frac{2\varepsilon_{g1}}{\pi}\left[\frac{1}{2}+\frac{\pi}{4}\sin\left(\frac{2\pi\varphi}{\varphi_T}\right) - \frac{1}{1\cdot 3}\cos\left(\frac{4\pi\varphi}{\varphi_T}\right) - \frac{1}{3\cdot 5}\cos\left(\frac{8\pi\varphi}{\varphi_T}\right) - \frac{1}{5\cdot 7}\cos\left(\frac{12\pi\varphi}{\varphi_T}\right) - ......\right]\right\}, \quad (1)$$

$$\varepsilon_\Sigma(\varphi,z) = \varepsilon\left\{1+\lambda\cos\varphi - \frac{2\varepsilon_{g1}}{\pi}\left[\frac{1}{2}+\frac{\pi}{4}\cos\left(\frac{2\pi z}{z_T}\right) + \frac{1}{1\cdot 3}\cos\left(\frac{4\pi z}{z_T}\right) - \frac{1}{3\cdot 5}\cos\left(\frac{8\pi z}{z_T}\right) + \frac{1}{5\cdot 7}\cos\left(\frac{12\pi z}{z_T}\right) - ...\right]\right\}, \quad (2)$$

for $0 \le \varphi < 2\pi$, $-b \le z \le b$ where λ – eccentricity ratio in cylindrical micro-bearing, ε – radial clearance in cylindrical micro-bearing, $\varepsilon_{g1} \equiv \varepsilon_g/\varepsilon$, ε_g – ridge height. Symbols φ_T, z_T denote periods of grooves sequence in φ and z – directions respectively.

The view of the cylindrical crosswise slide bearing oil clearance with the micro-grooves determined on the basis of the (1) and (2) equations are showed on the Fig. 2.

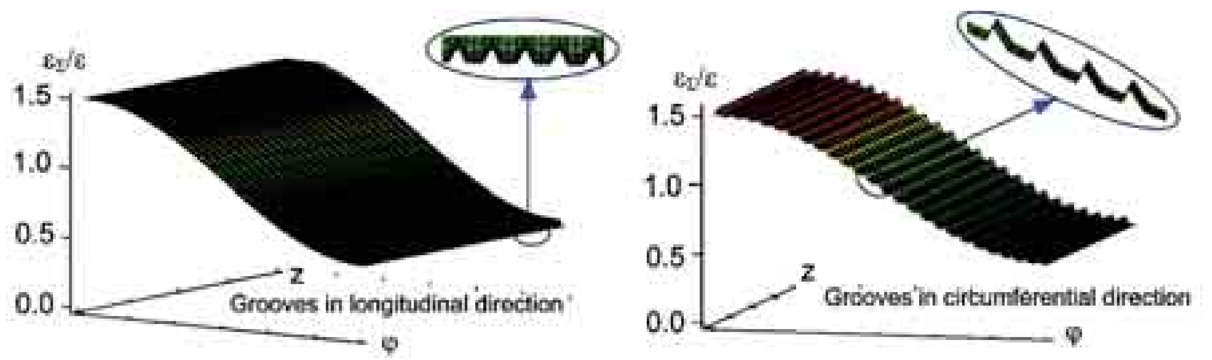


Fig. 2. The oil clearance in the crosswise slide bearing with micro-grooves

2. Basic equations

For the cylindrical micro bearing, we assume following cylindrical co-ordinates in φ, r, z directions. We consider classical boundary conditions [11] and take the time independent system of conservation of momentum, continuity and energy equation after thin boundary layer simplifications neglecting terms of greatness $\psi=\varepsilon/R$. After calculations, we obtain finally:

$$\frac{\partial}{\partial r}\left(\eta_\Sigma \frac{\partial v_\varphi}{\partial r}\right)=\frac{\partial p_\Sigma}{\partial \varphi},\quad \frac{\partial p_\Sigma}{\partial r}=0,\quad \frac{\partial}{\partial r}\left(\eta_\Sigma \frac{\partial v_z}{\partial r}\right)=\frac{\partial p_\Sigma}{\partial z},\quad \frac{1}{R}\frac{\partial v_\varphi}{\partial \varphi}+\frac{\partial v_r}{\partial r}+\frac{\partial v_z}{\partial z}=0, \tag{3}$$

$$\kappa_s\frac{\partial^2 T}{\partial r^2}+\eta_\Sigma\left[\left(\frac{\partial v_\varphi}{\partial r}\right)^2+\left(\frac{\partial v_z}{\partial r}\right)^2\right]=0, \tag{4}$$

where:
R – the radius of the cylindrical shaft,
2b – the cylindrical bearing length.
Integrating (3) under boundary conditions, we obtain following oil velocity components:

$$v_\varphi(\varphi,r,z)=\frac{1}{R}\frac{\partial p_\Sigma}{\partial \varphi}A_\eta+(1-A_s)\omega R,\quad v_z(\varphi,r,z)=\frac{\partial p_\Sigma}{\partial z}A_\eta,\quad v_r(\varphi,r,z)=-\frac{1}{R}\int_0^r\frac{\partial v_\varphi}{\partial \varphi}dr-\int_0^r\frac{\partial v_z}{\partial z}dr, \tag{5}$$

where:

$$A_s(\varphi,r,z)\equiv\frac{\int_0^r(\eta_\Sigma)^{-1}dr}{\int_0^{\varepsilon_\Sigma}(\eta_\Sigma)^{-1}dr},\quad A_\eta(\varphi,r,z)\equiv\int_0^r r(\eta_\Sigma)^{-1}dr-A_s(\varphi,r,z)\int_0^{\varepsilon_\Sigma}r(\eta_\Sigma)^{-1}dr, \tag{6}$$

and total hydrodynamic pressure:

$$p_\Sigma=p(\varphi,z)+p_T(\varphi,z)+p_{adh}(\varphi,z), \tag{7}$$

whereas:
$0\le r\le\varepsilon_\Sigma$, $0\le\varphi<2\pi\theta$, $0\le\theta_1<1$, $-b\le z\le b$,
$\varepsilon_\Sigma(\varphi,z)$ – gap height,
$p(\varphi,z)$ – hydrodynamic pressure,
$p_{adh}(\varphi,z)$ – average changes of hydrodynamic pressure caused by the adhesion forces,
$p_T(\varphi,z)$ – hydrodynamic pressure changes caused by the temperature.

The total and dimensionless dynamic viscosity coefficient of the lubricant depends on temperature and for $T\ge T_0$ has the following form [11, 13]:

$$\eta_\Sigma(\varphi,r,z)=\eta(\varphi,r,z)+\eta_{adh}(\varphi,r,z),\ \eta(\varphi,r,z)=\eta_0\eta_1(\varphi,z)\eta_{T1}(\varphi,r,z),\ \eta_{T1}(\varphi,r,z)\equiv e^{[-\delta_T(T-T_0)]}. \tag{8}$$

We denote: η_Σ – total oil dynamic viscosity, η – oil dynamic viscosity depended on temperature, η_0 – characteristic value of oil dynamic viscosity, η_{T1} – dimensionless oil dynamic viscosity changes caused by the temperature, η_1 – dimensionless oil dynamic viscosity independent on temperature, η_{adh} – oil dynamic viscosity changes caused by the adhesion forces, δ_T – temperature coefficient [K^{-1}] describing dynamic viscosity decreases during the oil flow, T_0 – characteristic average temperature and $T=T(r,\varphi,z)$.

If oil dynamic viscosity is constant in the gap height direction i.e. adhesion influences on the dynamic viscosity are neglected $\eta_{adh}=0$, temperature changes in gap height direction are neglected, $T=T_0$, hence formula (11) tends to unity i.e. $\eta_{T1}=1$.

Imposing the boundary condition $v_r=0$ for $r=\varepsilon_\Sigma$ on the radial velocity component $(5)_3$, then unknown pressure function $p_\Sigma(\varphi,z)$ satisfies Reynolds equations in the following forms [11, 13]:

$$\frac{1}{R^2}\frac{\partial}{\partial\varphi}\left[\left(\frac{\partial p_\Sigma}{\partial\varphi}\right)\int_0^{\varepsilon_\Sigma} A_\eta dr\right]+\frac{\partial}{\partial z}\left[\left(\frac{\partial p_\Sigma}{\partial z}\right)\int_0^{\varepsilon_\Sigma} A_\eta dr\right]=\omega\frac{\partial}{\partial\varphi}\left[\int_0^{\varepsilon_\Sigma} A_s dr-\varepsilon_\Sigma\right]. \tag{9}$$

The calculations were begun by numerical solving the Reynolds' equation (9), assuming, that the dynamic viscosity does not depend on the temperature After that, the distribution of temperature has to be determined assuming, that the dynamic viscosity is constant in the following dependence [11, 13]:

$$T_1(r,\varphi,z)=f_{c1}+\frac{1}{2}\eta_1(1-2s)-q_{c1}\varepsilon_{\Sigma 1}s-\frac{1}{6}\varepsilon_{\Sigma 1}^2\frac{\partial p_1}{\partial\varphi}s\,(3-3s+s^2)-\frac{1}{2}\eta_1\left[\left(v_{\varphi 1}\right)^2+\frac{1}{L^2}\left(v_{z1}\right)^2\right]+$$
$$+\frac{1}{24\eta_1}\varepsilon_{\Sigma 1}^4\left[\left(\frac{\partial p_1}{\partial\varphi}\right)^2+\frac{1}{L^2}\left(\frac{\partial p_1}{\partial z_1}\right)^2\right]s^3(s-2), \tag{10}$$

where: $f_{c1}=f_c/T_0$, $s=r/\varepsilon_\Sigma$, $0\le s\le 1$, $p_1=p/p_0$, $p_0=R^2\omega\eta_0/\varepsilon^2$, $v_{\varphi 1}= v_\varphi/U$, $v_{z1}= Lv_z/U$, $v_{r1}= v_r/U\psi$, $L=b/R$, $q_{c1}\equiv \upsilon\varepsilon/\kappa_s\Delta f$, $T=T_0+T_0BrT_1$, $Br=U^2\eta_0/\kappa_sT_0$, $T(r,\varphi,z)=f_c$ for $r=0$, $T(r,\varphi,z)= f_p(\varphi,z)$ for $r=\varepsilon_\Sigma$, T_1 – dimensionless temperature changes, Br – Brinkman number, q_c - heat flux, q_{c1} – dimensionless heat flux, υ – heat transfer, κ_s - dimensional thermal conductivity coefficient, Δf – dimensionless difference between micro-bearing journal temperature and ambient lubricant temperature, ω – angular velocity.

In next step the equation (9) has to be solved numerically with the use of the (6), (8) and (10) equations. With the values of hydrodynamic pressure determined for the viscosity variable of the temperature and the viscosity changes with the temperature, the component of the velocity vector (5) can be determined and the new value of the temperature can be obtained with the numerical equation of the energy (4). In next steps, the calculations can be repeated to obtain the convergence results in assumed accuracy.

With determined hydrodynamic pressure, we can determine the capacity force with the following dependence:

$$C_\Sigma=bR\eta_o\omega/\psi^2\cdot\sqrt{\left(\int_{-1}^{+1}\left(\int_0^{\phi_k}p_1\cos\gamma\sin\phi\,d\phi\right)dz_1\right)^2+\left(\int_{-1}^{+1}\left(\int_0^{\phi_k}p_1\cos\gamma\cos\phi\,d\phi\right)dz_1\right)^2}\;. \tag{11}$$

The components of friction forces in circumferential and longitudinal directions and friction coefficient μ occurring in cylindrical micro-bearing gaps have the following forms [11, 13]:

$$F_{R\varphi}=\iint_\Omega\left[\eta_\Sigma\frac{\partial v_\varphi}{\partial r}\right]_{r=\varepsilon_\Sigma}Rd\varphi dz,\quad F_{Rz}=\iint_\Omega\left[\eta_\Sigma\frac{\partial v_z}{\partial r}\right]_{r=\varepsilon_\Sigma}Rd\varphi dz,\ \mu=\frac{\sqrt{F_{R\varphi}^2+F_{Rz}^2}}{C_\Sigma}, \tag{12}$$

where: $0\le r\le\varepsilon_\Sigma$, $0\le\varphi<2\pi\theta_1$, $0\le\theta_1<1$, $-b\le z\le b$, and η_Σ – total dynamic viscosity.

3. Numerical calculations

We determine the pressure distributions and load carrying capacity values in micro bearing for cylindrical journal in the lubrication region Ω, which is defined by the following inequalities: $0\le\varphi\le\varphi_k$, $z_1=z/b$, $-b\le z\le b$ where 2b – micro-bearing length. Numerical calculations are performed in Mathcad 14 Program by virtue of the procedure calculation based on equations from (5) to (9) by means of the finite difference method.

We show in Fig. 3 the numerical calculations results of hydrodynamic pressure by virtue of (9) taking into account grooves lying in circumferential directions (Fig. 3b, 3c) and longitudinal directions (Fig. 3e, 3f) on the bearing sleeve. In presented calculations, we take following data for the grooves: $\varepsilon_{g1}= -0.05$, $\varphi_T=\pi/25$, $z_T=2/25$. We assume the radius of the journal R=0.0015 m, length/radius ratio L=b/R=1/2, dynamic viscosity of the oil η_0=0.050 Pas, angular velocity $\omega=754\ s^{-1}$, characteristic dimensional value of hydrodynamic pressure $p_o=\omega\eta_0/\psi^2$=9.425 MPa, relative radial clearance $\psi=\varepsilon/R$=0.002, eccentricity ratio λ from 0.1 to 0.9. By virtue of good known [11] boundary Gümbel conditions the angular coordinate of the film end has the values φ_k=3.1415 rad. Moreover we assume average temperature T_0=323 K and thermal conductivity coefficient κ_s=0.15 W/mK and coefficient describing influence of temperature on viscosity equal to δ_T=0.047 1/K. We denote $Q_{Br}=BrT_0\delta_T$.

The values of load carrying capacity, friction force and friction coefficient are showed on the Fig. 4-6.

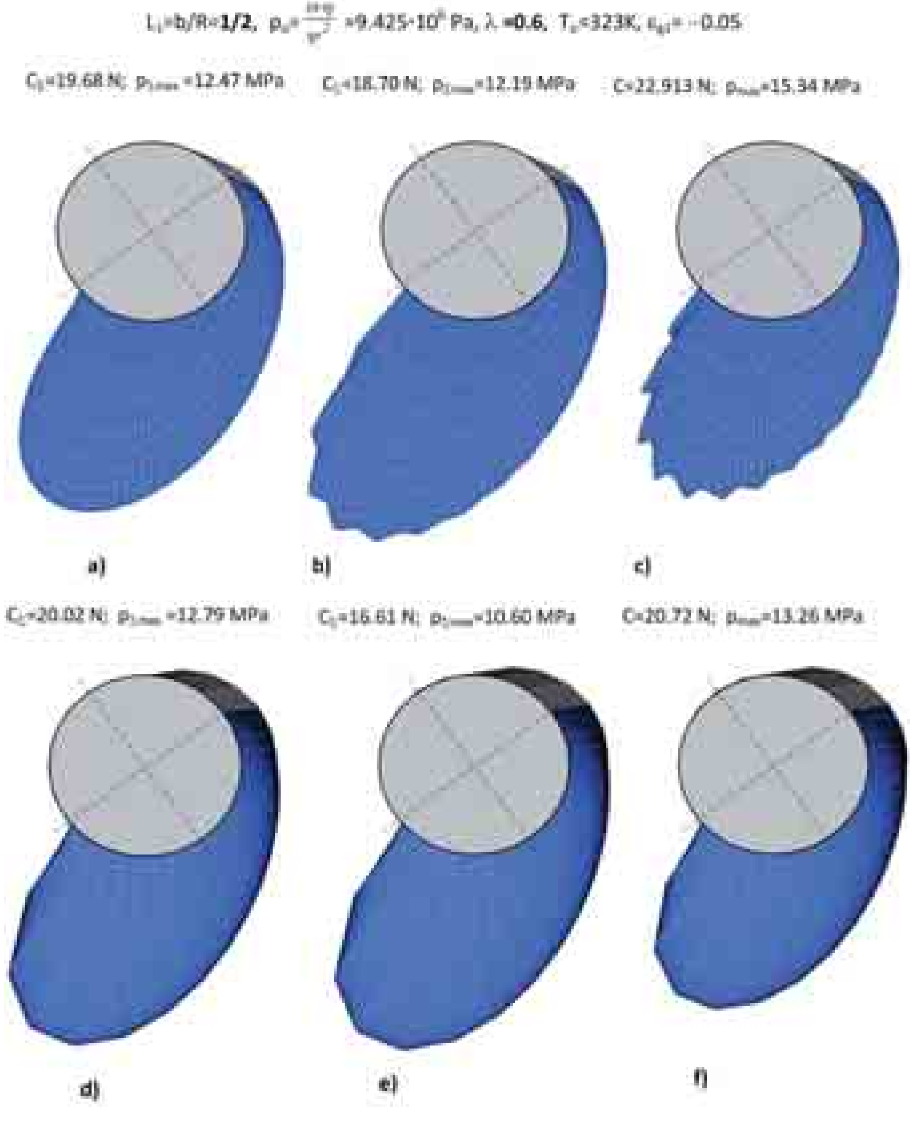


Fig. 3. Hydrodynamic pressure distribution for cylindrical micro-bearing: a), d) without grooves and without temperature influences, b), e) without temperature influences and for grooves on the sleeve surface, c), f) for grooves and viscosity changes with temperature in gap high direction; b), c) grooves lying in circumferential directions, e), f) grooves lying in longitudinal directions

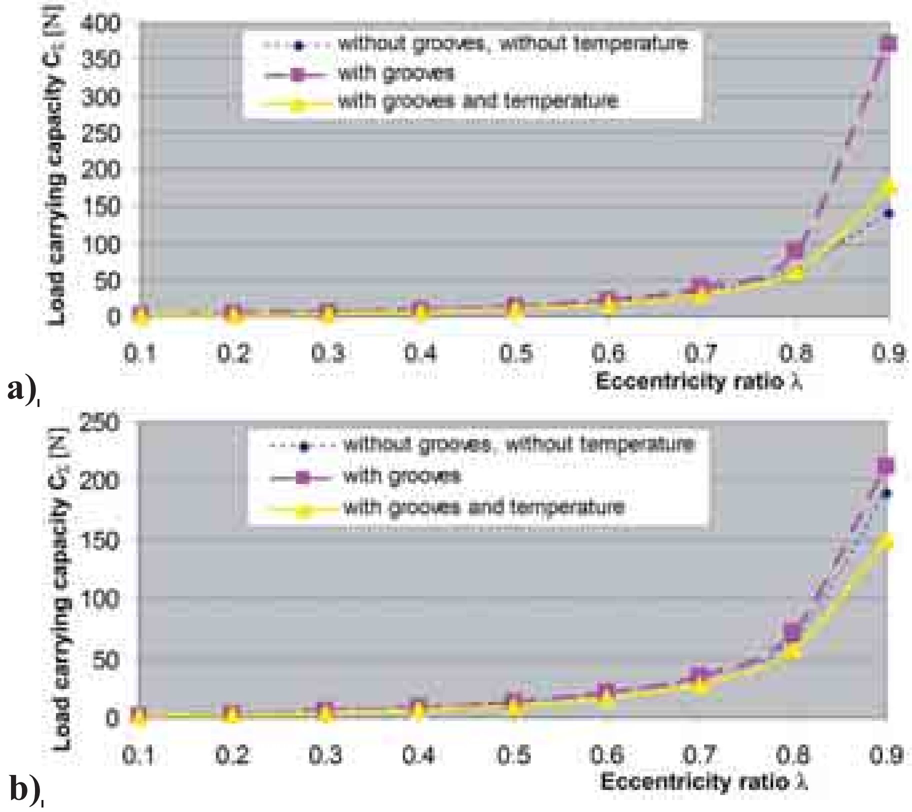


Fig. 4. Load carrying capacity for cylindrical micro bearing: a) grooves lying in circumferential directions, b) grooves lying in longitudinal directions

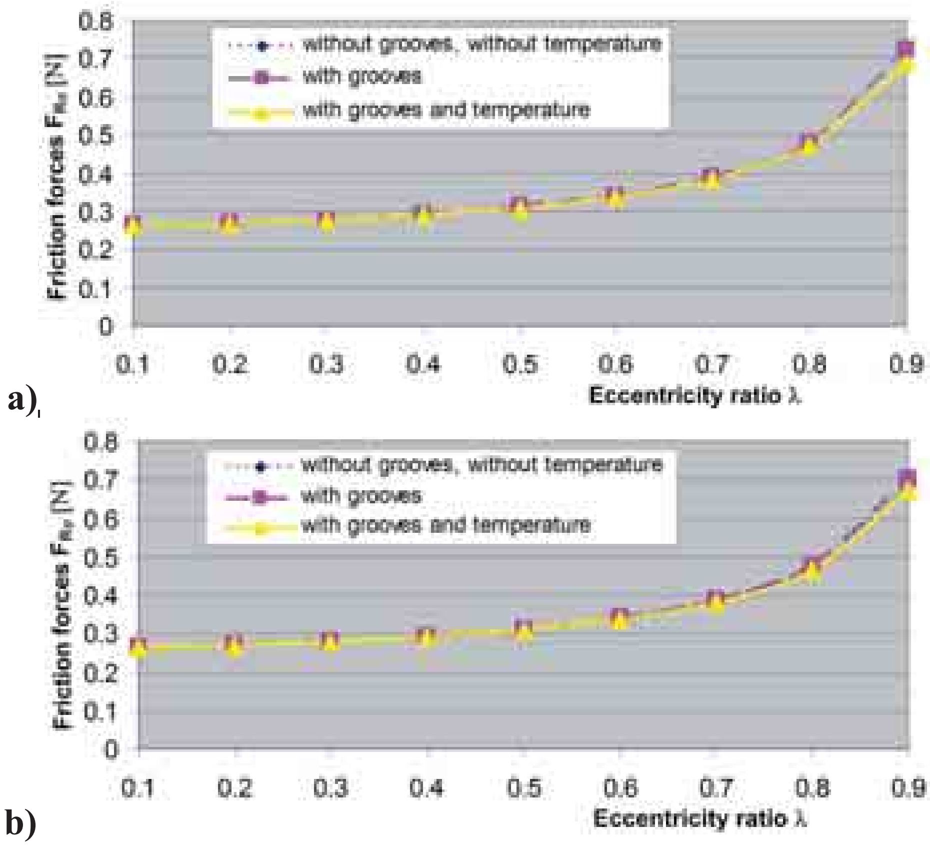
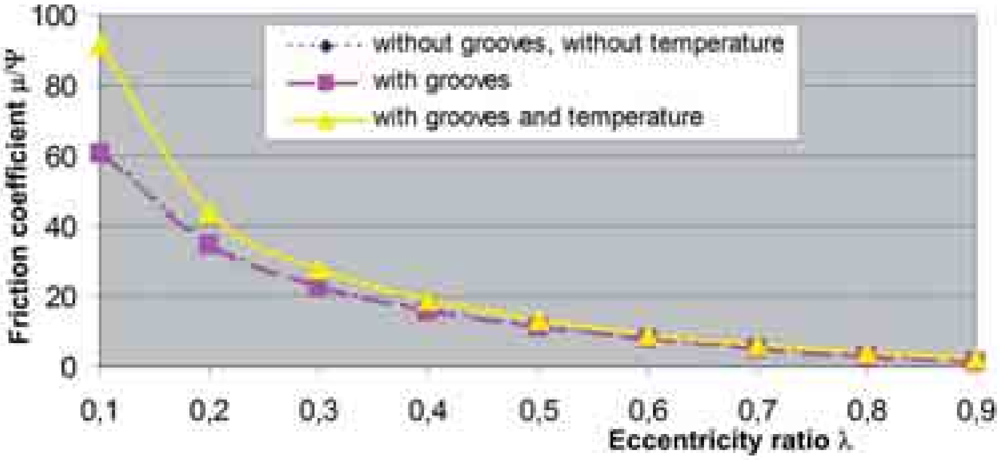
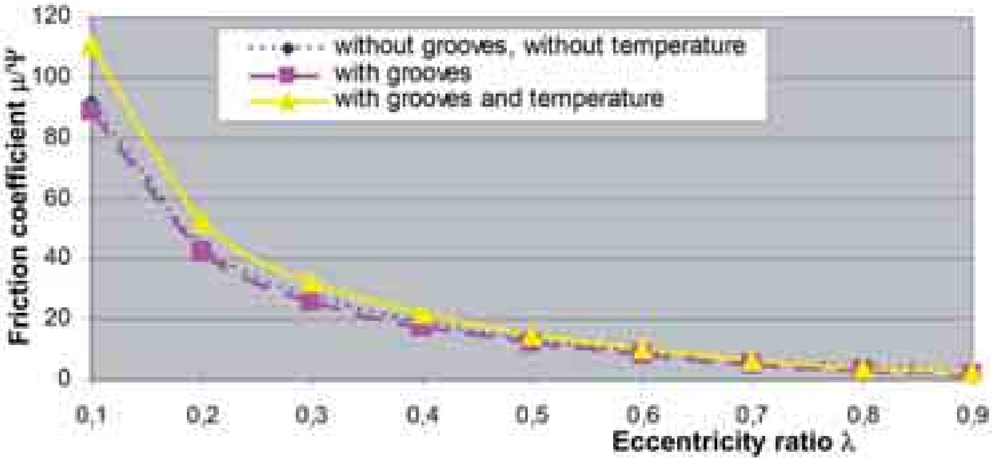


Fig. 5. Friction forces for cylindrical micro bearing: a) grooves lying in circumferential directions, b) grooves lying in longitudinal directions



a)



b)

Fig. 6. Friction coefficient for cylindrical micro bearing: a) grooves lying in circumferential directions, b) grooves lying in longitudinal directions

4. Conclusions

On the ground of numerical calculation, the following results are obtained:

1. In gap height direction, the gradients of temperature are very significant, however the value changes are negligible small. Therefore, the temperature changes in gap height cannot be neglected.
2. In Fig. 3b, 3c) was shown, that micro-grooves have influence on the pressure distribution shapes.
3. The grooves of the micro-bearing surfaces have significantly influence on the hydrodynamic pressure increases (about 25 percent for grooves lying in circumferential directions and 4 percent for grooves lying in longitudinal directions).
4. Non classical assumptions of the viscosity and temperature changes in gap high directions implicate hydrodynamic pressure decreases with temperature (about 25 percent) i.e. this fact denotes that mentioned decreases are many more than in classical cases.
5. The performed calculation is limited by the possibilities of Mathcad Program. During the considerations had been attained the precision possibilities of calculations due to the micro-grooves.

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