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THE INFLUENCE OF OIL'S EXPLOITATION TIME ON LOAD CARRYING CAPACITY IN A SLIDER BEARING

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Abstract

In this paper, authors are presenting conclusions of the numerical calculations of pressure distribution and capacity in a slider bearing with taking changes of oil viscosity in exploitation time into account. Changes of the engine oil's viscosity, which depend on the exploitation time, were determined on Haake Mars III rheometer and the conclusions were published in Solid State Phenomena and Logistyka in 2015. Numerical calculations were performed by solving of Reynolds equation, using finite difference method and own calculation procedures in Mathcad 15. Reynolds equation was developed by solving the continuity equation and the momentum conservation equation from the fundamentals. For the considerations, the laminar and stationary lubricating of the slider bearing of finite length and full angle of wrap were taken. Assumption of the stationary flow concerns lack of changes in flow parameters in short period of considered phenomena, f. ex. in one hour. Smooth and non-porous bushing were assumed. The aim of this paper was preliminary estimation of influence of viscosity changes in the exploitation time on the load carrying capacities of the cross slider bearing. Wherefore, the viscosity changes dependence on the pressure, temperature and also shear rate, were not taken into account. The basic equations were developed to the non-dimensional form and estimated according to the thin layer theory. In the calculations, the Reynolds boundary conditions concerning pressure distribution were taken into account. Preliminary calculations were performed for different models of viscosity changes in time and circumstances, where the viscosity increases and decreases in exploitation time.

Keywords: *oil ageing, numerical calculations, Reynolds equation, pressure distribution, load carrying capacities, viscosity changes in time*

1. Introduction

Exploitation of the slide friction nodes in different types of engines, which are working on land, sea and air as well, causes that lubricator is under ageing process. The most often case of degradation is change of the viscosity and lubricity. Change of the viscosity causes change in carrying capacity load, friction force, and friction coefficient. Temperature distribution changes also and in end effect lubrication gap changes as well. In case of viscosity decrease, may cause lowering of the lubrication gap below the allowed value. This case may cause the border or mixed friction, and as its effect, sooner wear of the friction node's surface. From the results of the viscosity's changes analysis comes out, that the highest viscosity changes are observed in the first period of exploitation time. These changes have an exponential form [5, 6]. Viscosity changes in the vessel's engines cannot be higher than 30 %. In the engines of the land vehicles, which are working under heavy agricultural conditions or equipped with DPF and exploited mostly in urban mode, the viscosity change can be higher than 50 % [5]. This level of changes is dangerous for the slider friction nodes and what influences on proper work of the engine.

The aim of this work is value estimation of the carrying capacity load for the different cases of viscosity changes in exploitation time. For the exploitation time assumed are kilometers, working hours or hours (in dependence of application).

In this work, viscosity dependence of changes of temperature, pressure or shear rate on carrying capacity load are not taken into account.

2. Analytical model

Numerical calculations were performed using Finite Difference Method, own calculation procedures and Mathcad 15 software. Reynolds-type equation has been received from the fundamental equations: conservation of momentum and continuity equation. For the calculations, the laminar and stationary lubrication of the slider bearing of finite length, full angle of wrap and smooth non-porous bushing were taken. Laminar and stationary lubricating flow was assumed. Stationarity of the flow concerned short period of time, at the level of few thousands of seconds. Time elements in the momentum conservation equation are in this case small higher order, whereas, value changes in higher periods of time f. ex. several million of seconds are taken into account. In the thin layer of oil film, constancy of the dependence of density on temperature and independence of the heat conduction coefficient from the thermal changes was assumed. This is because in the range of oil film, the influence of temperature on density and thermal conductivity is negligible.

Deformation of the bushing surface, caused by the pressure or temperature is in this work not taken into account.

Fundamental equations were presented in the author's earlier papers [3, 4]. For the numerical calculations, the classical, dimensionless height of the oil gap, without taking skewing into account was taken into account [3, 7, 8]:

$$h_{p1} = 1 + \lambda \cos\phi . \tag{1}$$

Reynolds-type equation with taking viscosity changes in exploitation time into account has a form [3, 7, 8]:

$$\frac{\partial}{\partial\phi}\left(\frac{h_{p1}^3}{\eta_1(t)}\frac{\partial p_1}{\partial\phi}\right) + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left(\frac{h_{p1}^3}{\eta_1(t)}\frac{\partial p_1}{\partial z_1}\right) = 6\frac{\partial h_{p1}}{\partial\phi}, \tag{2}$$

where:

L_1 – dimensionless length of the bearing,
p_1 – dimensionless pressure,
z_1 – dimensionless axial coordinate,
ϕ – circumference coordinate,
$\eta_1(t)$ – dimensionless dynamic viscosity.

For the analysis of influence viscosity changes in exploitation time, the following models of changes were taken into account [4-6]:

$$\eta_1(t) = b_1 \cdot e^{-a_1 \cdot t},\ \eta_1(t) = b_1 \cdot e^{+a_1 \cdot t},\ \eta_1(t) = a\cdot t^6 - b\cdot t^5 + c\cdot t^4 - d\cdot t^3 + e\cdot t^2 + f\cdot t + 1, \tag{3}$$

where:

a, a_1, b, c, d, e, f – dimensional coefficients,
b_1 – dimensionless correction coefficient of the characteristic, dimensional viscosity value η_o.

Dimensional value of the carrying capacity load C_Σ in cross slider bearing is being determined from the known equation [2, 3, 7, 8]:

$$C_\Sigma = C_1^{(0)} \cdot b\, R\, \eta_o \omega / \psi^2 , \tag{4}$$

where:

b – half of the bearing length,
R – journal radius,
η_o – dimensional characteristic value of the viscosity,
ω – radial velocity of the journal,
ψ – radial relative gap.

Dimensionless value of the carrying capacity load C_1 in the cross slider bearing is calculated from the following dependence [3, 8]:

$$C_1 = \sqrt{\left(\int_{-1}^{+1}\left(\int_{0}^{\phi_k} p_1 \sin\phi \,\mathrm{d}\phi\right)\mathrm{d}z_1\right)^2 + \left(\int_{-1}^{+1}\left(\int_{0}^{\phi_k} p_1 \cos\phi \,\mathrm{d}\phi\right)\mathrm{d}z_1\right)^2}\,. \tag{5}$$

3. Numerical calculations

Numerical calculations of the pressure distributions and load carrying capacities are performed in Mathcad 15 Professional Program by virtue of the equation (2), (5), by means of the finite difference method (see Fig. 1-4). On the ground of pressure, distributions are calculated the load carrying capacities (see Fig. 3 and 4).

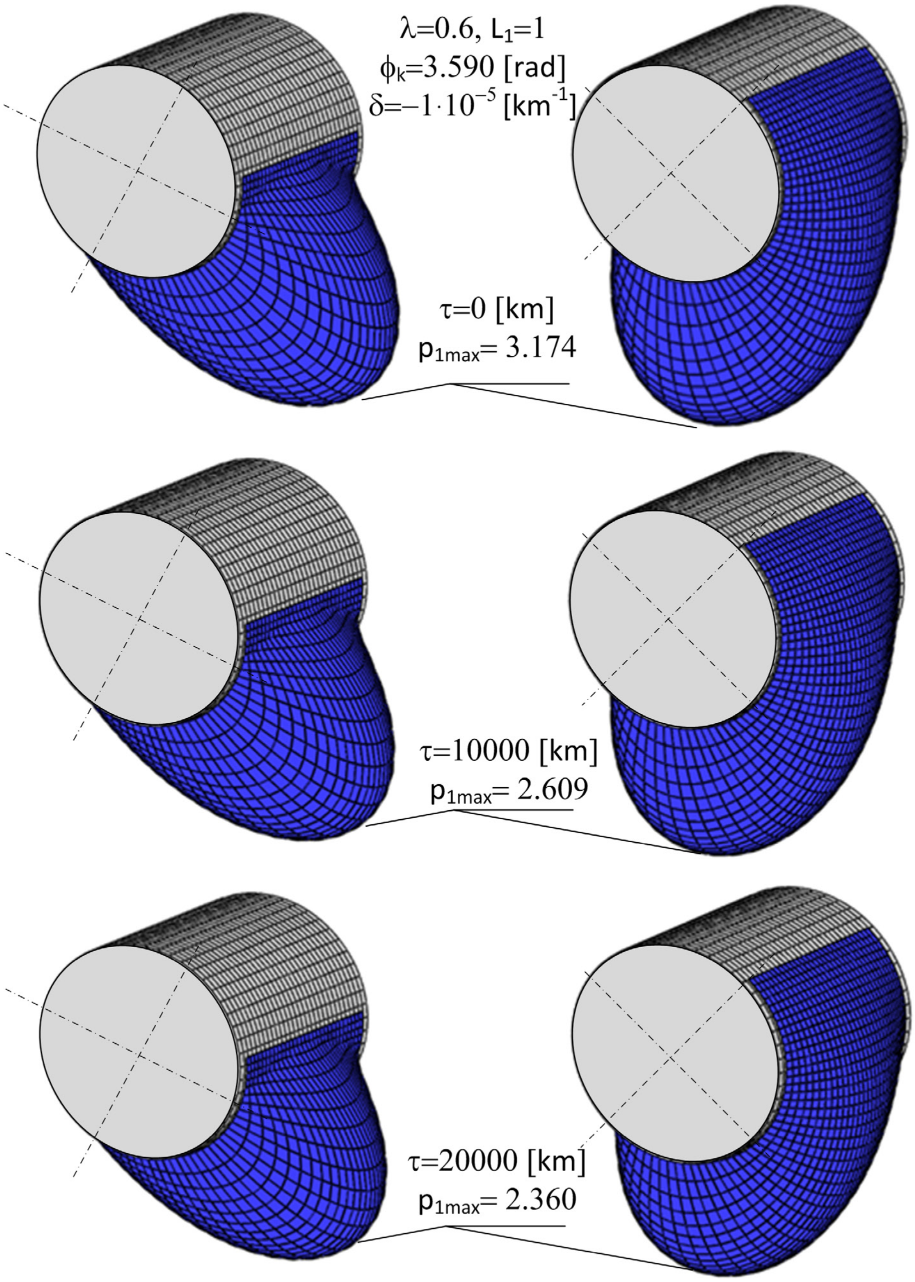

Fig. 1. The dimensionless pressure distributions p_1 in cylindrical sliding journal bearings for $\eta_1(t) = b_1 \cdot e^{-a_1 \cdot t}$

Numerical calculations were performed for the relative eccentricities $\lambda = 0.6$; and dimensionless bearing's length $L_1 = 1$ by three models of viscosity changes in time (3).

For the calculations, the following values of the coefficients were assumed:

$$a_1 = \pm 1 \cdot 10^{-5}\ [(\text{e.t.})^{-1}],\quad a = 2.41474 \cdot 10^{-24}\ [(\text{e.t.})^{-6}],\quad b = 1.6710280093400003 \cdot 10^{-20}\ [(\text{e.t.})^{-5}],$$

$$b_1 = 0.891,\quad c = 4.510846724853931 \cdot 10^{-16}\ [(\text{e.t.})^{-4}],\quad d = 6.03991 \cdot 10^{-12}\ [(\text{e.t.})^{-3}],$$

$$e = 4.224328585 \cdot 10^{-8}\ [(\text{e.t.})^{-2}],\quad f = 0.00015232132236633\ [(\text{e.t.})^{-1}]$$

where (e.t.) – exploitation time unit f. ex. kilometer, working hour, hour.

Values of the coefficients are determined with the high accuracy, because of the risk of big error in case of rounding. Coefficients were determined in Excel (trendline), basing on the experimental research [4-6].

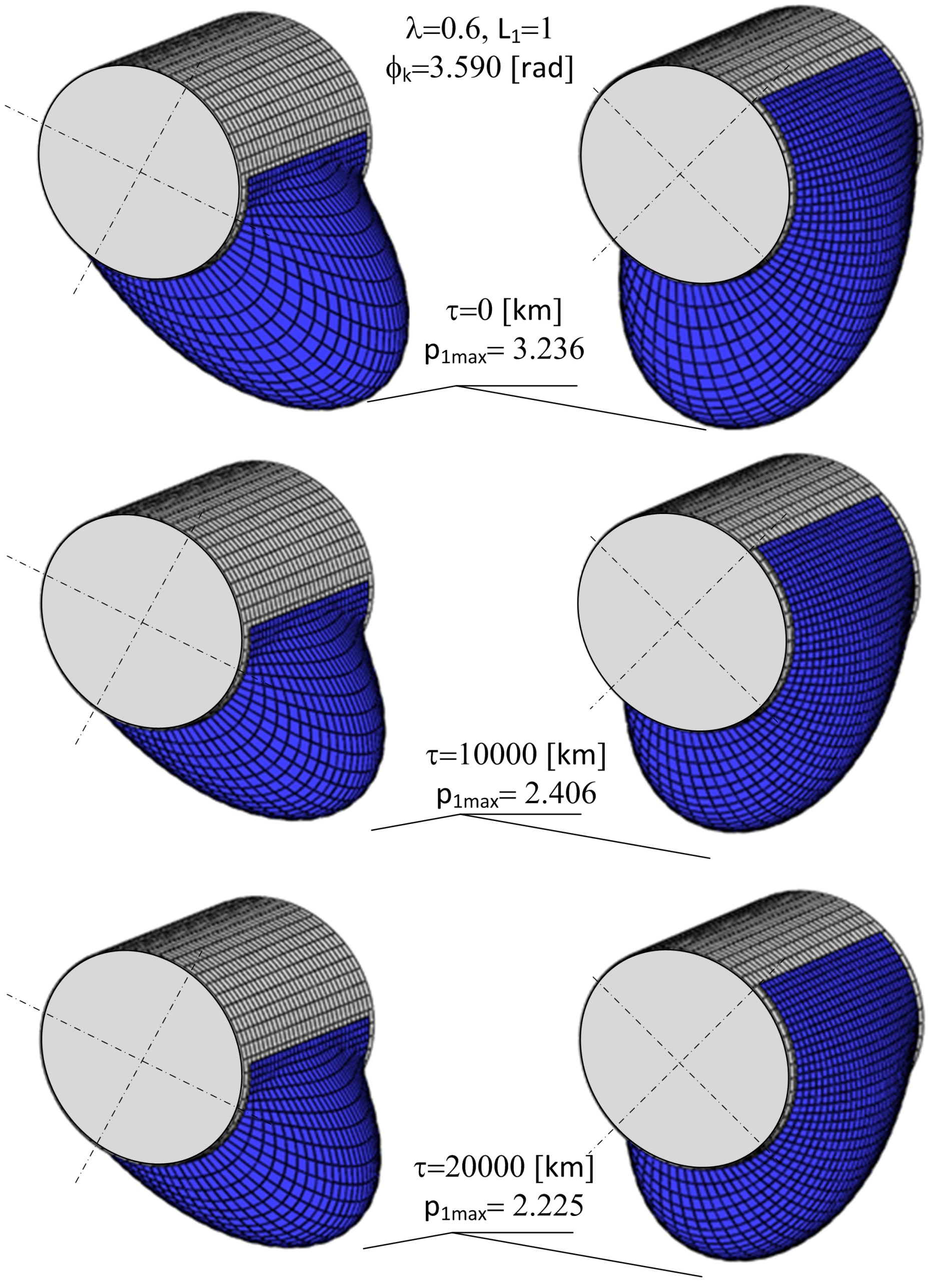

Fig. 2. The dimensionless pressure distributions p_1 in cylindrical sliding journal bearings for $\eta_1(t) = a \cdot t^6 - b \cdot t^5 + c \cdot t^4 - d \cdot t^3 + e \cdot t^2 - f \cdot t + 1$

Hydrodynamic pressure distribution for the first model of viscosity changes in time (3_1) are presented in Fig. 1, while for the polynomial (3_3) in Fig. 2. By determination of the hydrodynamic pressure distribution, the Reynolds, boundary conditions were assumed.

Dimensionless carrying capacity load in the cross slider bearing are calculated basing on the formula (5) are shown in Fig. 3. In the figure, both models are compared for the same lubricating oil. This oil showed viscosity decrease in the exploitation time.

In Fig. 4, authors presented changes of the load carrying capacity for the two oils, where viscosity increases or decreases in exploitation time. For these calculations, the power series model (3_1) and (3_2) has been used, by the usage of the same values of the material coefficients.

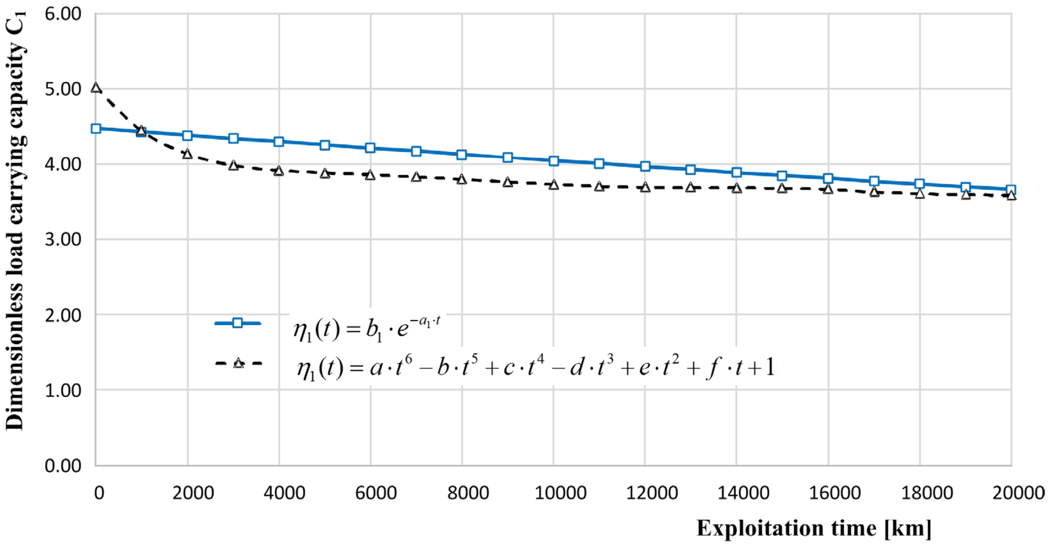

Fig. 3. Dimensionless vaules of the carrying capacity load for two models of viscosity changes in exploitation time: $\eta_1(t) = b_1 \cdot e^{-a_1 \cdot t}$ and $\eta_1(t) = a \cdot t^6 - b \cdot t^5 + c \cdot t^4 - d \cdot t^3 + e \cdot t^2 - f \cdot t + 1$

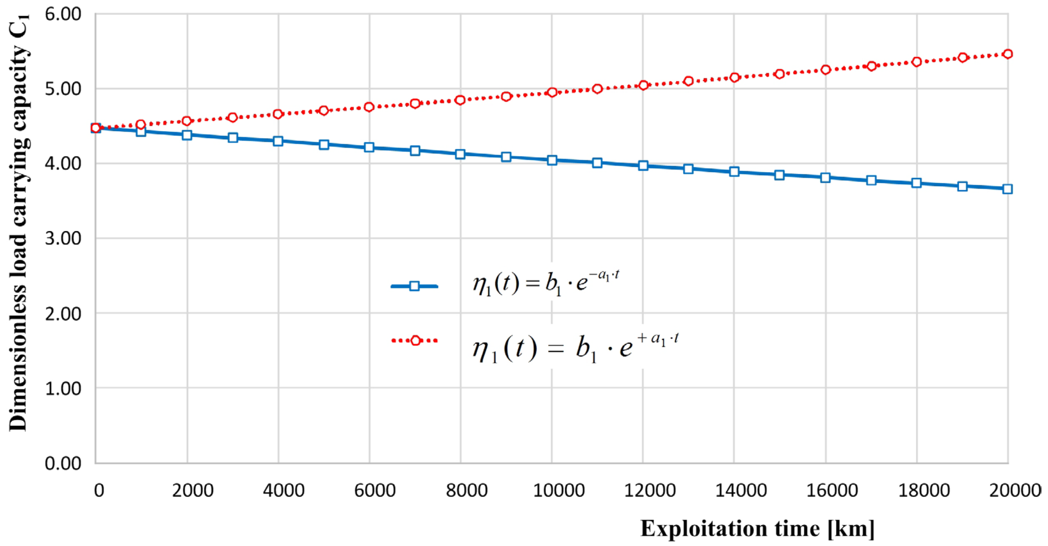

Fig. 4. Dimensionless values of the carrying capacity load for the decreasing and increasing dynamic viscosity in exploitation time: $\eta_1(t) = b_1 \cdot e^{-a_1 \cdot t}$ and $\eta_1(t) = b_1 \cdot e^{+a_1 \cdot t}$

4. Conclusion

In this paper, only influence of viscosity changes in exploitation time were taken into account. Depending on the accuracy of the model, which describes viscosity changes in exploitation time, the differences in carrying capacity load of even 10 % are reached.

Viscosity change, which results from the exploitation time, makes a change in hydrodynamic pressure and carrying capacity load as well. The values of these changes are proportional to the value of changes in in dynamic viscosity and can reach even too few dozens of percent.

In this paper, the influence of the viscosity changes in exploitation time on the temperature, which highly influences on the other flow and exploitation parameters. Such a research is planned to be performed in the future.

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