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Journal bearing with an intensive axial oil flow – computer simulations

Key words

Journal bearing, surface modification, three-dimensional oil flow, finite element method.

Słowa kluczowe

Łożysko ślizgowe, modyfikacja powierzchni, trójwymiarowy przepływ, metoda elementów skończonych.

Summary

The paper presents some selected calculation results whose purpose is to find the dimensions of the spiral groove on the journal bearing that the load capacity should not be impaired. A three-dimensional oil flow was considered which was described with the Navier-Stokes equations and the equation of energy. The system of equations was solved by applying the finite element method. It was proved that a 0.1mm deep and 0.4mm wide triangular groove does not cause a reduction in the load capacity of the bearing and ensures an intensification of the axial oil flow.

1. Introduction

Hard particles of impurities occurring in oil are one of the main causes of premature journal bearing damage. The problem has been the subject of

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scientific research for many years, but it still has not lost its relevance, which is evidenced by papers [1-5]. Experimental research [6] demonstrated that making a spiral groove in the journal can significantly reduce the susceptibility of the bearing system to the damaging effect of impurities contained in oil. This is due to such impurities placing themselves in the grooves and an intensification of axial oil flow. The positive result is the removal of impurities from the oil clearance and the reduction of wear.

The difficulty which may render a practical application of the solution impossible is the fact that, in the case of lubricating with clean oil, a groove in the journal worsens the conditions necessary for oil film formation and may consequently reduce the load capacity of the bearing. The triangular groove made on the journal used for the research described in paper [6] was 0.15 mm wide and 0.6 mm deep. It brought about an almost threefold decrease in wear in the case of lubrication with oil contaminated with aluminium oxide particles. However, when clean oil was used, wear was two times faster compared to a bearing with a plain journal.

It is therefore necessary to carry on further research in order to select the shape and dimensions of a grove on the journal which would collect impurities and intensify the axial oil flow without affecting the load capacity of the bearing. Paper [7] presents preliminary work in this respect based on a simplified bearing model. The investigation described in the paper is a continuation and includes a computer simulation of the flow. Later on, an experimental verification of the simulation results is planned.

2. A model of a bearing and oil film

A model of a bearing with a spiral groove is shown in Fig.1. Based on the preliminary research, a triangular groove with rounded edges was presumed. The shape is analogous to that assumed for the investigation described in paper [6]. The dimensions characterising the groove are its width s and depth g. The model of the oil clearance obtained by the development of the working surfaces of the journal and the sleeve is shown in Fig. 2. While modelling the bearing, the journal and sleeve surfaces were represented by mathematical functions. The sleeve surface (S1) outside the oil groove area was described with the dependence:

$$\mathbf{x} \in \langle \mathbf{0 : 2\Pi r} \rangle,\ \ \mathbf{y = R + m \cdot \cos\left(\frac{x}{r}\right)},\ \ \mathbf{z} \in \langle \mathbf{0 : b} \rangle \tag{1}$$

The journal surface was presented as:

$$\mathbf{x} \in \langle \mathbf{0 : 2\Pi r} \rangle,\ \ \mathbf{y = r + A \cdot e^{-B[(z-C_1)+(D \cdot x)]^2} + A \cdot e^{-B[(z-C_2)+(D \cdot x)]^2}},\ \mathbf{z} \in \langle \mathbf{0 : b} \rangle \tag{2}$$

where: A, B, C_1, C_2, D – are coefficients

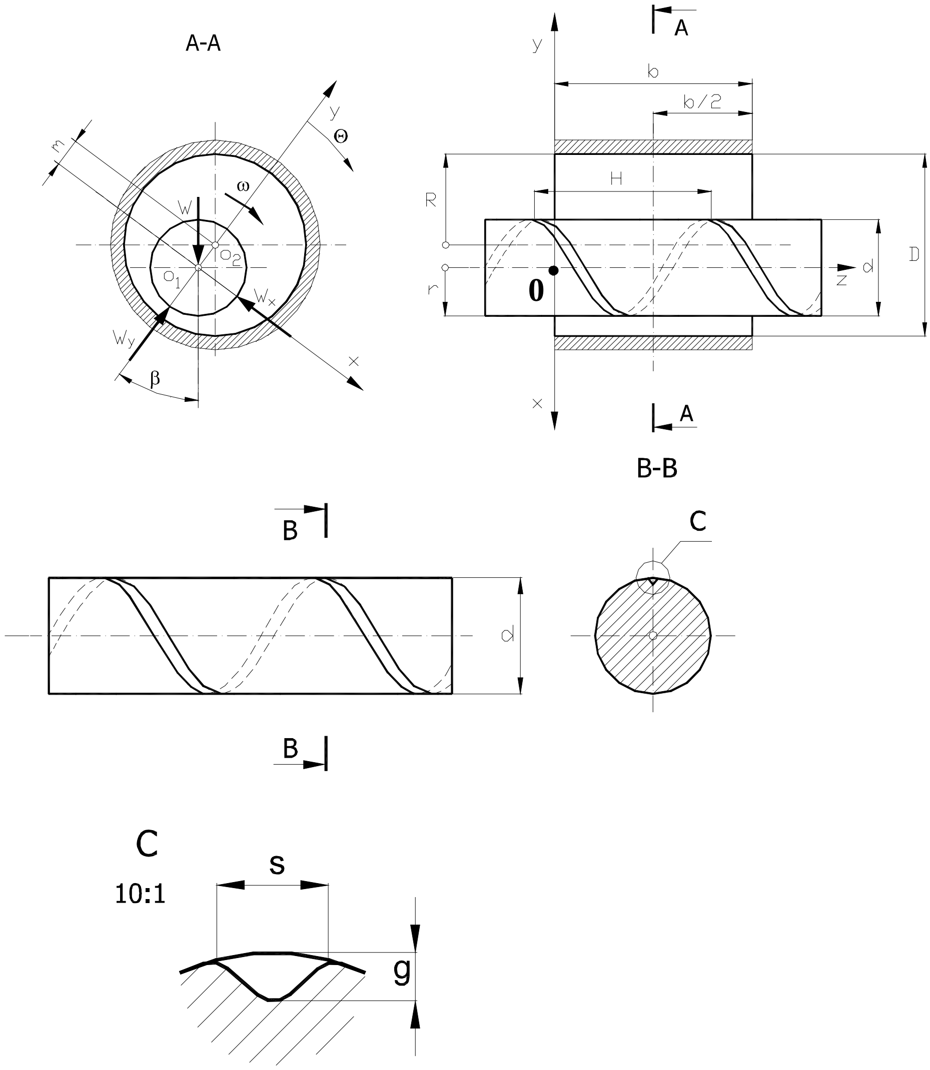

Fig. 1. Bearing with a spiral-grooved journal
Rys. 1. Łożysko z czopem ze śrubowym rowkiem

In formula (2), coefficient A defines the depth of the groove, coefficient B defines its width, coefficients C_1 and C_2 define its location, and coefficient D defines the inclination angle of the spiral lead of the groove. The oil groove was modelled as a cubical of length b_r, width s_r and height h_r.

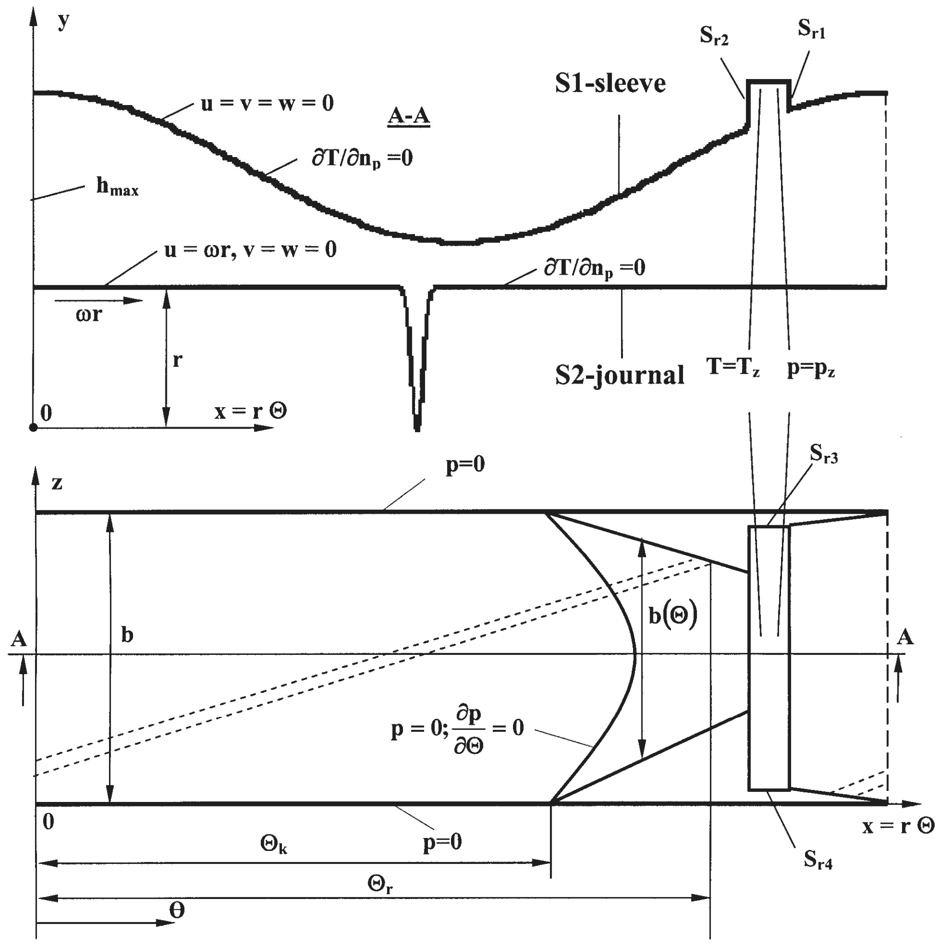

Fig. 2. Development of the working surfaces of the journal and the sleeve together with boundary conditions
Rys. 2. Rozwinięcie powierzchni roboczych czopa i panewki wraz z warunkami brzegowymi

3. Formulation of flow problem

The groove in the journal surface has a depth that may exceed 100 μm, so it is comparable to the amount of oil clearance. It necessitates a modification of the classical, hydrodynamic lubrication theory. Typical simplifying assumptions related to, e.g. a constant pressure value along the height of the bearing interspace (along axis y) and lack of effect of a number of constituent gradients of oil velocity, are no longer relevant to reality. This makes it necessary to consider a three-dimensional flow described with the Navier-Stokes energy and

continuity equations. While analysing the oil flow in the bearing under consideration, the following assumptions were made:
- Oil is a Newtonian liquid,
- Flow is laminar,
- Flow at a fixed moment of time t is considered (quasi-static model) – turn was modelled assuming successive positions of the groove,
- Body force field does not occur,
- Oil is incompressible,
- Oil density is independent of temperature,
- Oil viscosity depends only on temperature,
- Heat conditions are fixed, and
- The coefficients of oil specific heat and oil heat conductivity are constant values.

For the above assumptions the Navier-Stokes equations are as follows:

$$\rho\left(\mathbf{u}\frac{\partial \mathbf{u}}{\partial \mathbf{x}}+\mathbf{v}\frac{\partial \mathbf{u}}{\partial \mathbf{y}}+\mathbf{w}\frac{\partial \mathbf{u}}{\partial \mathbf{z}}\right)=-\frac{\partial \mathbf{p}}{\partial \mathbf{x}}+\eta\left(\frac{\partial^2 \mathbf{u}}{\partial \mathbf{x}^2}+\frac{\partial^2 \mathbf{u}}{\partial \mathbf{y}^2}+\frac{\partial^2 \mathbf{u}}{\partial \mathbf{z}^2}\right), \tag{3}$$

$$\rho\left(\mathbf{u}\frac{\partial \mathbf{v}}{\partial \mathbf{x}}+\mathbf{v}\frac{\partial \mathbf{v}}{\partial \mathbf{y}}+\mathbf{w}\frac{\partial \mathbf{v}}{\partial \mathbf{z}}\right)=-\frac{\partial \mathbf{p}}{\partial \mathbf{y}}+\eta\left(\frac{\partial^2 \mathbf{v}}{\partial \mathbf{x}^2}+\frac{\partial^2 \mathbf{v}}{\partial \mathbf{y}^2}+\frac{\partial^2 \mathbf{v}}{\partial \mathbf{z}^2}\right), \tag{4}$$

$$\rho\left(\mathbf{u}\frac{\partial \mathbf{w}}{\partial \mathbf{x}}+\mathbf{v}\frac{\partial \mathbf{w}}{\partial \mathbf{y}}+\mathbf{w}\frac{\partial \mathbf{w}}{\partial \mathbf{z}}\right)=-\frac{\partial \mathbf{p}}{\partial \mathbf{z}}+\eta\left(\frac{\partial^2 \mathbf{w}}{\partial \mathbf{x}^2}+\frac{\partial^2 \mathbf{w}}{\partial \mathbf{y}^2}+\frac{\partial^2 \mathbf{w}}{\partial \mathbf{z}^2}\right). \tag{5}$$

The equation of flow continuity is in the form:

$$\frac{\partial \mathbf{u}}{\partial \mathbf{x}}+\frac{\partial \mathbf{v}}{\partial \mathbf{y}}+\frac{\partial \mathbf{w}}{\partial \mathbf{z}}=\mathbf{0}, \tag{6}$$

however, the energy equation is

$$\rho c\left(\mathbf{u}\frac{\partial \mathbf{T}}{\partial \mathbf{x}}+\mathbf{v}\frac{\partial \mathbf{T}}{\partial \mathbf{y}}+\mathbf{w}\frac{\partial \mathbf{T}}{\partial \mathbf{z}}\right)=\mathbf{k}\left(\frac{\partial^2 \mathbf{T}}{\partial \mathbf{x}^2}+\frac{\partial^2 \mathbf{T}}{\partial \mathbf{y}^2}+\frac{\partial^2 \mathbf{T}}{\partial \mathbf{z}^2}\right)+\eta\boldsymbol{\Phi}, \tag{7}$$

where the energy dissipation function is

$$\boldsymbol{\Phi}=\left\{2\left[\left(\frac{\partial \mathbf{u}}{\partial \mathbf{x}}\right)^2+\left(\frac{\partial \mathbf{v}}{\partial \mathbf{y}}\right)^2+\left(\frac{\partial \mathbf{w}}{\partial \mathbf{z}}\right)^2\right]+\left(\frac{\partial \mathbf{u}}{\partial \mathbf{y}}+\frac{\partial \mathbf{v}}{\partial \mathbf{x}}\right)^2+\left(\frac{\partial \mathbf{v}}{\partial \mathbf{z}}+\frac{\partial \mathbf{w}}{\partial \mathbf{y}}\right)^2+\left(\frac{\partial \mathbf{w}}{\partial \mathbf{x}}+\frac{\partial \mathbf{u}}{\partial \mathbf{z}}\right)^2\right\}. \tag{8}$$

The dependence of oil dynamic viscosity on temperature was described by function

$$\eta(T) = 0{,}0625 \cdot e^{\left(-7{,}973 + \frac{661246}{T^2}\right)} \tag{9}$$

To determine the pressure zone and the flow rate, the following boundary conditions were assumed (Fig. 3):

– In the supply groove, the pressure equals the supply pressure p_z;

$$p = p_z\,, \tag{10}$$

– Oil pressure in the axial direction that occurs on the outlet edges of the bushing equals the ambient pressure that was adopted as the reference level = 0;

$$p(\Theta, z = 0) = p(\Theta, z = b) = 0\,, \tag{11}$$

– Pressure in the non-working zone = 0;

$$p(\Theta, z)_n = 0\,, \tag{12}$$

– The end of the oil film zone is the locus of points where pressure equals ambient pressure and pressure gradient equals zero;

$$p_{\Theta=\Theta_{x,z}} = 0 \;\text{ and }\; \left(\frac{\partial p}{\partial \Theta}\right)_{\Theta=\Theta_{x,z}} = 0\,, \tag{13}$$

– Journal velocity = ωr, the journal only rotates, therefore on surface S2:

$$u = \omega r,\; v = w = 0\Big|_{S2}\,, \tag{14}$$

– The sleeve is stationary, so on surface S1 and the surrounding surfaces the supply groove

$$u = v = w = 0\Big|_{S1,Sr1,Sr2,Sr3,Sr4}\,, \tag{15}$$

– Providing that, in the bearing interspace, the following dependencies are satisfied:

$$p(\Theta = 0) = p(\Theta = 2\Pi)\,, \tag{16}$$

$$u(\Theta = 0) = u(\Theta = 2\Pi)\,, \tag{17}$$

$$v(\Theta = 0) = v(\Theta = 2\Pi)\,, \tag{18}$$

$$w(\Theta = 0) = w(\Theta = 2\Pi)\,. \tag{19}$$

While modelling the flow in the non-working zone, it was assumed that oil flows just like one stream and does not get outside the bearing. It was also presumed that oil fills the whole oil clearance height and the spiral groove is flooded with oil when covered by the stream. Another groove filling occurs in the supply groove. The width of the oil film (axial direction) in the non-working zone in the area where the model oil stream covers the groove (between Θ_k and Θ_r – Fig. 3) was derived from the dependence:

$$\mathbf{b}(\Theta) = \mathbf{b}\frac{\mathbf{h}(\Theta_\mathbf{k})}{\mathbf{h}(\Theta)}. \tag{20}$$

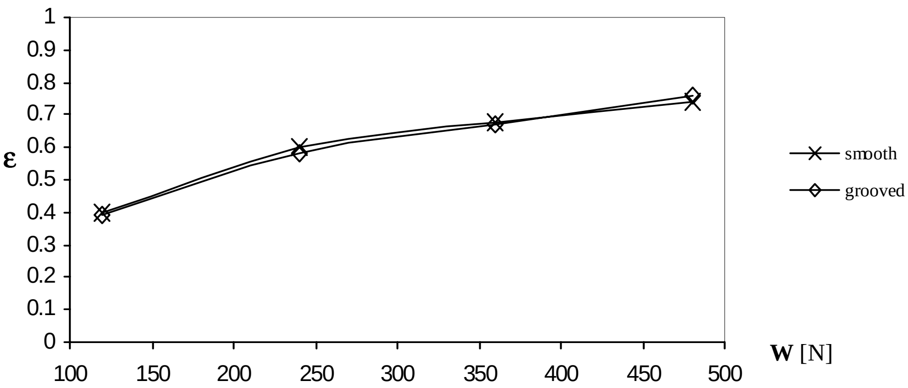

Fig. 3. Effect of bearing load on eccentricity ratio
Rys. 3. Wpływ obciążenia na mimośrodowość względną

It was assumed that the oil streams out at the point where the groove is not covered. After approximately representing the shape of the groove section as a triangle, the width of the modal stream in the zone was derived from dependence 21, where the other component of the sum allows for the oil in the groove.

$$\mathbf{b}(\Theta) = \mathbf{b}\frac{\mathbf{h}(\Theta_\mathbf{k})}{\mathbf{h}(\Theta)} + \frac{\mathbf{0.5} \cdot \mathbf{s} \cdot \mathbf{g}}{\mathbf{h}(\Theta)} \tag{21}$$

As a consequence of adopting an adiabatic model of the flow, the following boundary conditions were accepted to find the temperature zone:

- The groove is totally filled with oil of temperature denoted as T_z (the effect of the streams mixing with each other in the supply groove was ignored assuming they mix in the oil film),

– The heat stream penetrating the journal and the sleeve surfaces equals 0; therefore,

$$\frac{\partial T}{\partial n}\Big|_{S1} = \frac{\partial T}{\partial n}\Big|_{S2} = 0 \tag{22}$$

where n denotes a normal to the surface.

What is more, an assumption was made that

$$T(\Theta = 0) = T(\Theta = 2\Pi) \tag{23}$$

The equation system (3) – (7) was solved by means of the finite element method making use of an ADINA 8.1.W software package. In the volume of the oil clearance 113340 three-dimensional elements were generated.

The quantities chosen to compare the calculation results at this stage were

– oil film maximum pressure p_{max},
– oil film load capacity W:

$$W = \sqrt{(W_x)^2 + (W_y)^2} \tag{24}$$

where:

$$W_x = r\int_0^b \int_0^{\Theta_k} p \sin\Theta \, d\Theta \, dz\,, \tag{25}$$

$$W_y = -r\int_0^b \int_0^{\Theta_k} p \cos\Theta \, d\Theta \, dz\,, \tag{26}$$

– attitude angle β:

$$\beta = \operatorname{arctg}\left|\frac{W_x}{W_y}\right|, \tag{27}$$

– oil film maximum temperature T_{max},
– oil flow rate:

$$q_o = 2\int_0^h \int_0^{\Theta_k} w \, r d\Theta dy\,. \tag{28}$$

4. Research results and their analysis

Computer simulations were carried out for various geometrical variants of the bearing and various running conditions. About 100 different variants were examined. The detailed computation results are given in paper [8]. They showed that a spiral groove on a journal with a lead equal 2/3 of the bearing width, of a depth not exceeding 30% of the oil clearance and width equal the threefold depth did not affect the load carrying capacity. The article presents some selected results corroborating the fact that a groove in the journal does not necessarily mean a reduction in the bearing load capacity. They allow us to make a comparison between the characteristics of a plain journal bearing and those of a spiral-grooved journal variant. A description of the geometry of the bearings and the simulated running conditions were presented in Table 1.

Table 1. Geometry of the bearings and simulated running conditions
Tabela 1. Geometria łożysk oraz symulowane warunki pracy

Variant	D [mm]	d [mm]	b [mm]	p_z [MPa]	T_z [K]	ω [rad/s]	s [mm]	g [mm]	H [mm]
Smooth	61,19	60,88	47	0	295	25,12	-	-	-
Grooved	61,19	60,88	47	0	295	25,12	0,4	0,1	30

The results of the calculations are shown in Figs. 4–8. The parameters characterising the bearing were given as a function of load capacity. It will make it easier to compare them later with the results of experimental research.

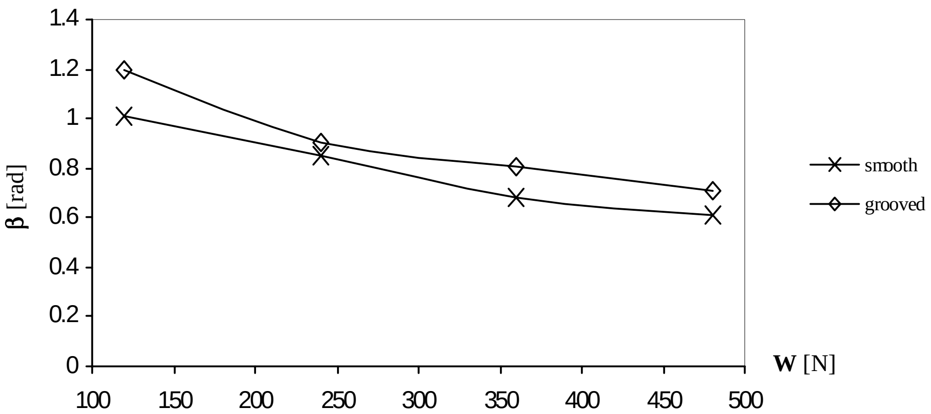

Fig. 4. Effect of bearing load on attitude angle
Rys. 4. Wpływ obciążenia na kąt położenia linii środków

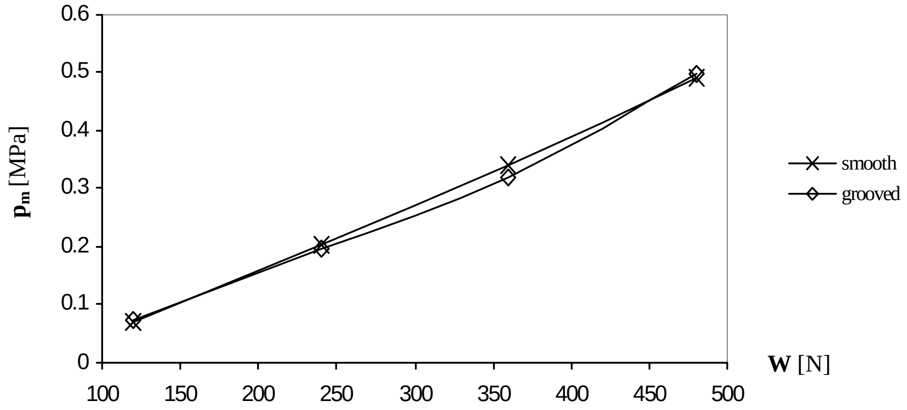

Fig. 5. Effect of bearing load on oil maximum pressure in oil film
Rys. 5. Wpływ obciążenia na maksymalne ciśnienie w filmie olejowym

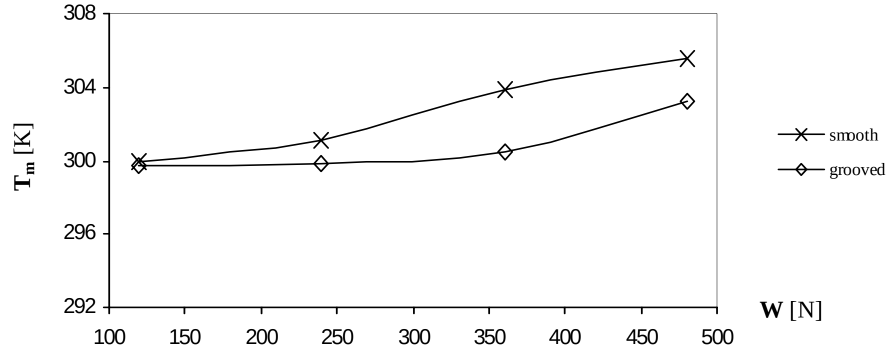

Fig. 6. Effect of bearing load on maximum temperature of oil film
Rys. 6. Wpływ obciążenia na maksymalną temperaturę filmu olejowego

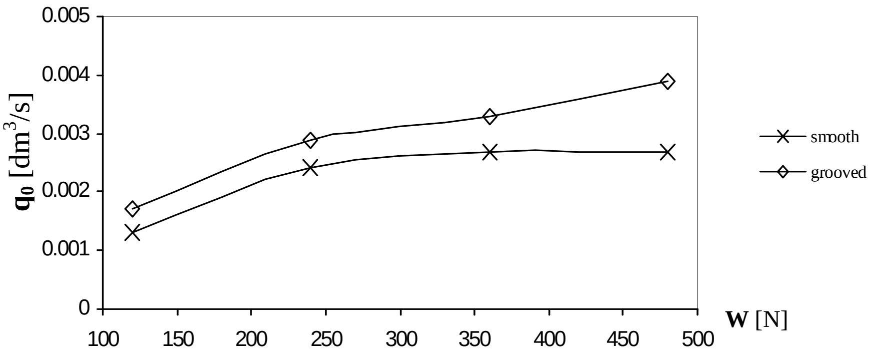

Fig. 7. Effect of bearing load on oil flow rate
Rys 7. Wpływ obciążenia na wydatek oleju

The results shown in Fig. 4 indicate that both a plain journal bearing and that with a modified journal run practically at the same relative eccentricities. Within the medium range of the investigated load capacities, a modified bearing has even slightly lower eccentricities (1.5-3%) than a standard bearing. It can be concluded from the results that it is possible to select such parameters of the spiral groove on the journal of a slide bearing that they do not impair its load capacity. On the contrary, the computations seem to indicate that the groove can slightly improve load capacity. A groove on the journal is also responsible for an almost 20% increase in the value of the attitude angle (Fig.5). The maximum pressure of the oil film values for the two variants are similar (Fig. 6). At the assumed running conditions of the bearing, no significant temperature increase in the oil film occurred (Fig. 7). However, throughout the investigations, it was found that the maximum oil film temperatures are lower for a bearing with a spiral-grooved journal than for a standard one with a plain bearing. The groove also causes a higher oil flow rate (Fig. 8). It is from 20% (at 240N) to 50% (at 480N) greater than in the case of a plain journal.

The obtained results corroborate a possibility of selecting such groove dimensions that the load capacity of the journal bearing does not decrease and, at the same time, the axial oil flow intensifies. A groove on the journal together with an enhanced flow towards the axis can make the bearing more immune to the damaging effect of the impurities contained in the oil. An extra effect of an increased oil flow through a bearing is a decrease in the oil film temperature. The obtained effects are due to a change in the bearing interspace geometry. In the oil film of a bearing with a modified journal there occur flow phenomena that do not take place in a standard bearing [8]. It is due to them that, despite potentially worse conditions for film formation (because of the groove on the journal), achieving the same load capacity as in the case of a bearing with a plain journal is possible. The next stage of the research will be an experimental verification of the results of computer simulations.

5. Conclusions

1. A spiral-grooved journal bearing of appropriate dimensions can have the same load-carrying ability as a standard bearing with a plain journal.
2. A spiral groove on the journal of a slide bearing causes an intensification of axial oil flow and markedly (even up to 50% in the simulations carried out) increases oil use.
3. A groove in the journal and an enhanced axial oil flow can facilitate the removal of impurities from the bearing interspace and contribute to the experimentally proved increased immunity of such a bearing to the damaging effect of hard particles.

Nomenclature

b – bearing length (m),
c – oil specific heat ($c = 2000$ J/(kg·K)),
D – sleeve diameter (m),
d – journal diameter (m),
g – groove depth (m),
H – spiral lead of the groove (m),
k – oil heat conductivity ($k = 0.145$ W/(m·K)),
m – journal eccentricity (m),
R – sleeve radius (m),
r – journal radius (m),
p – oil pressure (Pa),
p_m – oil maximum pressure
p_z – supply pressure (Pa),
q_o – oil flow rate (m^3/s),
s – groove width (m),
S_1 – journal surface,
S_2 – sleeve surface,
T – temperature (K),
T_m – oil film maximum temperature (K),
T_z – supply temperature (K),
u – x-component of oil velocity (circumferential) (m/s),
v – y-component of oil velocity (radial) (m/s),
w – z-component of oil velocity (axial) (m/s),
W – load carrying capacity (N),
x, y, z – axis of coordinate system,
β – attitude angle (rad),
ε – eccentricity ratio ($\varepsilon = m/(R-r)$),
Φ – dissipation function,
η – oil dynamic viscosity (Pa·s),
Θ – coordinate in circumferential direction as measured from the point where oil layer thickness is maximum (rad),
Θ_k – angle at which the oil pressure zone ends (rad),
ρ – oil density ($\rho = 880$ kg/m^3),
ω – journal angular velocity (rad/s).

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Łożysko ślizgowe z intensywnym osiowym przepływem oleju – symulacje komputerowe

S t r e s z c z e n i e

W artykule przedstawiono wyniki komputerowych symulacji przepływu oleju w łożysku ślizgowym, ukierunkowanych na poszukiwanie takich wymiarów śrubowego rowka na czopie, który nie będzie powodował zmniejszenia nośności łożyska. Rozważno trójwymiarowy przepływ oleju, który został opisany równaniem Naviera-Stokesa oraz równaniem energii. Opisujący przepływ oleju układ równań rozwiązano metodą elementów skończonych. Stwierdzono, że trójkątny rowek o głębokości 0,1 mm oraz szerokości 0,4 mm nie powoduje zmniejszenia nośności łożyska, a jednocześnie intensyfikuje osiowy przepływ oleju.