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THE STUDY OF SLIDE BEARING PROPERTIES INCLUDING THE SURFACE TEXTURE GEOMETRY OF A JOURNAL

BADANIA WŁAŚCIWOŚCI ŁOŻYSK ŚLIZGOWYCH Z UWZGLĘDNIENIEM STRUKTURY GEOMETRYCZNEJ POWIERZCHNI CZOPA

Key words:

static friction factor, molecular-mechanical theory of friction, slide bearing, real area of a contact, bush bearing, journal bearing

Słowa kluczowe:

współczynnik tarcia spoczynkowego, molekularno-mechaniczna teoria tarcia, łożyska ślizgowe, rzeczywista powierzchnia kontaktu, panewka, czop

Summary:

A static friction during a slide bearing's start-up appears to be a complicated process. To deal with the phenomenon, the author uses a molecular-mechanical theory of friction. This paper reports the results that show the influence of a

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surface geometric structure of a journal on a static friction coefficient, a frictional moment and tangential stresses on a bushing surface. In analysis of results, it can be noticed that machining selection and the accuracy of a journal surface have the significant impact on bearing system during its start-up.

REGISTER OF SYMBOLS:

Relating to properties of a bearing bushing material: E – Young's modulus [N/m^2], μ – Poisson's modulus, τ_0 – shear resistance of adhesive bonds [N/m^2], β – molecular component of friction resistance.
Relating to values of a bearing geometry: B – bearing bushing width [m], $C_R = R_{B1}–R_J$ – radial clearance [m], $g_B = R_{B2}–R_{B1}$ – bearing bushing thickness [m], R_J – journal radius [m], R_{B1} – inner radius of bushing [m], R_{B2} – outer radius of bushing [m], $2\varphi_0$ – contact angle on journal and bushing surfaces [rad].
Relating to values describing the geometric structure of a journal surface: A_c – contour area (total) for a journal – bearing bushing contact, A_r **–** real friction surface, A_{ri} – friction surface relating to single micro roughness [m], b – parameter of capacity profile curve, h – penetration depth of roughness peaks [m], k_1 – constant dependent on capacity curve parameters, R – curvature radius of surface roughness peak [m], R_{max} – maximum roughness profile height [μm], R_z – surface roughness height parameter [μm], t_p – capacity profile function, α – coefficient which describes stress states in a static balance position in friction area: $\alpha = 0.5$ – concerning elastic deformation, $\alpha = 1.0$ – concerning elastic–plastic deformation, ν – parameter of capacity profile curve, Δ – roughness dimensionless coefficient, $\varepsilon_h = h / R_{max}$ – relative depth.
Other values: f – friction factor on bearing bushing surface, F – load [N], M_T – frictional moment [Nm], α_{ef} – dissipation factor resulting from hysteresis deformation of micro roughness on bushing surface, σ – stresses on bushing surface [N/m^2], τ_T – resultant tangential stresses on bushing surface [N/m^2], τ_{Tm} – tangential stresses arising from molecular interaction on solids boundary [N/m^2], τ_{Td} – tangential stresses in deformation area of micro roughness [N/m^2].

INTRODUCTION

A static friction during a slide bearing's start-up appears **(Fig. 1)**, to be a complicated process. To deal with the phenomenon, the author uses a molecular-mechanical theory of friction **[L. 4, 5, 8]**. This theory describes the friction force as the sum of forces of a molecular interaction between journal and bushing materials and forces dependent on the deformation of a body's surface coating whose material hardness is low, for example, a bushing. The resultant friction force is estimated by the following equation:

$$F_T = F_{Tm} + F_{Td} \tag{1}$$

A frictional moment on a journal surface is equal to the product of the friction coefficient, the bushing radius, and the force applied to a bearing:

$$M_T = f \cdot F \cdot R_{B1} \tag{2}$$

A frictional moment can be calculated from the following relation:

$$M_T = 2 \cdot R_{B1}^2 \cdot B \cdot \int_{-\varphi_0}^{\varphi_0} \tau_T \cdot d\varphi_0 \tag{3}$$

Therefore, for a given geometry and bearing load, the friction coefficient (Fig.1) is equal to the following:

$$f = \frac{2 \cdot R_{B1}^2 \cdot B \cdot \int_{-\varphi_{-}}^{\varphi_0} \tau_T \cdot d\varphi_0}{F \cdot R_{B1}} = \frac{2 \cdot R_{B1} \cdot B \cdot \int_{-\varphi_0}^{\varphi_0} \tau_T \cdot d\varphi_0}{F} \tag{4}$$

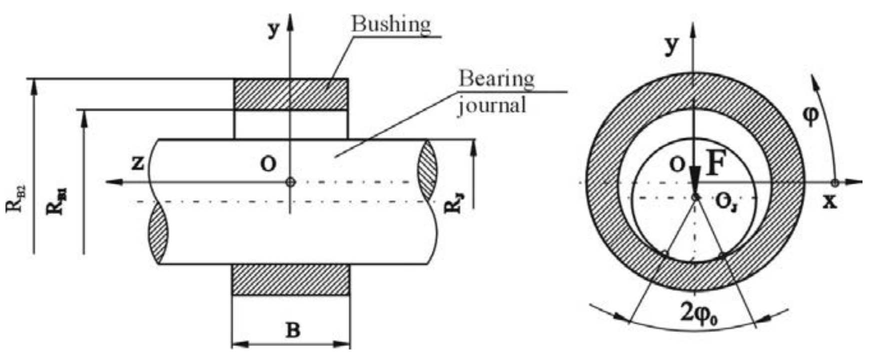

Fig. 1. Slide bearing geometry
Rys. 1. Geometria łożyska ślizgowego

To design a model, the following assumptions are accepted: the bushing surface is perfectly smooth while the journal surface is rough. The material hardness of a journal is HRC>40; whereas, the bushing is made of a bearing alloy with a hardness of HB<100. It is assumed has that the deformations of the journal outside surface are essentially small in comparison with deformations of the outside surface of the bushing. Moreover, the deformations of the outside surface will be elastic. Other assumptions concerning the use of a physical model and a mathematical model equation for slide bearings are presented in

[L. 6]. This paper reports the results that show the influence of the surface geometric structure of a journal on the static friction coefficient, the frictional moment and tangential stresses on a bushing's surface.

SURFACE TEXTURE GEOMETRY OF JOURNAL

A micro-geometric model of the journal surface **[L. 1–3, 7]** is taken into consideration and shown in **Fig. 2**. It is described by the following values:

- A real friction surface which refers to a single micro roughness can be written in the following terms: $A_r = \alpha \cdot 2 \cdot \pi \cdot R \cdot h$ (5)
- Profile capacity function: $t_p = b \cdot \varepsilon_h^{\nu}$ (6)
- Roughness dimensionless coefficient: $\Delta = \dfrac{R_{\max}}{R \cdot b^{\frac{1}{\nu}}}$ (7)
- Penetration depth of roughness peaks of a journal in a bushing surface:

$$h = \left[\frac{5 \cdot \sigma_n \cdot R^{0,5} \cdot \left(1 - \mu^2\right) \cdot R_{\max}^{\nu}}{b \cdot \nu \cdot (\nu - 1) \cdot k_1 \cdot E} \right]^{\frac{2}{2\nu + 1}} \tag{8}$$

There is a relationship between the real friction surface and the contour surface that can be presented by the following equation:

$A_r = \alpha \cdot A_c \cdot t_p = \alpha \cdot A_c \cdot b \cdot \varepsilon_h^{\nu}$, where:

$$A' = \frac{A_r}{A_c} = \alpha \cdot \left[\frac{5 \cdot \sigma_n \cdot R^{0,5} \cdot \left(1 - \mu^2\right)}{\nu \cdot (\nu - 1) \cdot k_1 \cdot E \cdot \Delta^{\frac{1}{2}}} \right]^{\frac{2\nu}{2\nu + 1}} \tag{9}$$

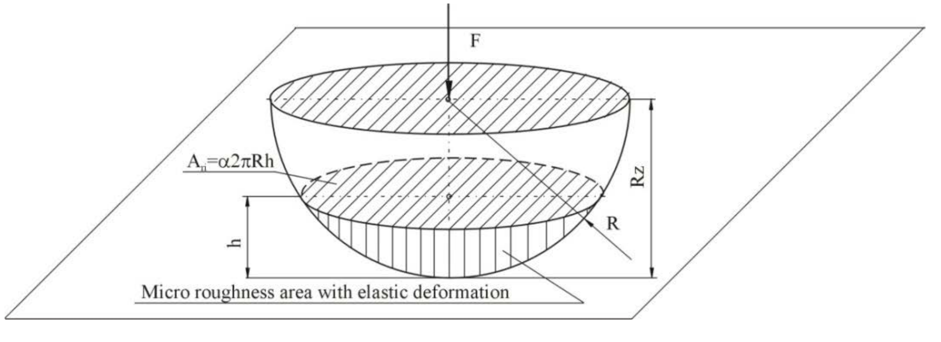

Fig. 2. Micro roughness of a journal surface model
Rys. 2. Model mikronierówności powierzchni czopa

CALCULATION EXAMPLE

To carry out the calculation, the author has chosen a journal slide bearing and described its geometry and material properties in **Table 1**. The following values are examined for a given load (F): $2\varphi_0$ – contact angle of a journal and a bushing surface, M_T – frictional moment on a bushing surface [Nm], $\tau(\varphi)$ – tangential stresses on a bushing surface [N/m^2], and f – friction coefficient on a bushing surface. To study the bearing's properties, the author has accepted three types of geometric forms of the journal surface that can be created in turning, grinding, and honing processes. These operations have been examined according to accuracy classes, namely, rough and precise. Parameter values that describe a surface geometric structure are accepted based on studies published in **[L. 4]**. Results are given in the form of a function: $F = F(\varphi_0)$, $M_T = M_T(\varphi_0)$, $\tau_T = \tau_T(\varphi_0)$, $f = f(\varphi_0)$ and presented in the **Figures 3–4.**

Table 1. Calculation example
Tabela 1. Przykład obliczeniowy

Given values	
LOAD AND VALUES DESCRIBING BEARING GEOMETRY	
F – load [N]	200.0 – 1.0 x 10^6
R_J – journal radius [m]	209.745 x 10^{-3}
R_{B1} – inner radius of bearing bushing [m]	210.0 x 10^{-3}
R_{B2} – outer radius of bearing bushing [m]	214.0 x 10^{-3}
B – bushing width [m]	315.0 x 10^{-3}
Material properties of bushing	
E – Young's modulus [N/m^2]	0.38 x 10^{11}
μ – Poisson's modulus	0.38
$\overline{p}_{lim}$ – acceptable mean forces [Pa]	7.0 x 10^6
τ_{T0} – shear resistance of adhesive bonds [N/m^2]	8.0 x 10^6
β – molecular component of friction resistance	0.065
α_{ef} – losses coefficient caused by hysteresis deformations of micro roughness on bushing surface	0.1
k_1 – constant dependent on capacity curve parameters,	0.4
Values describing surface texture geometry of journal	
rough turning	Δ= 7.9 x 10^{-1}; ν= 1.9
precise turning	Δ= 6.3 x 10^{-2}; ν= 1.6
rough grinding	Δ= 1.6 x 10^{-1}; ν= 2.0
precise grinding	Δ= 2.8 x 10^{-2}; ν= 1.5
rough honing	Δ= 1.2 x 10^{-1}; ν= 1.7
precise honing	Δ= 4.65 x 10^{-3}; ν= 1.6

It is assumed that an acceptable value of mean forces is $\overline{p}_{\lim} = 7{,}0\,x10^{6}[Pa]$, acceptable bearing load is F_{lim} = 926 x 10^3[N]. The acceptable load is equal to this angle, $2\varphi_{0lim} = 19^0$. For the established given values presented in **Table 1** regions are defined by this inequality $F \leq F_{\lim}\ i\ 2\varphi_0 \leq (2\varphi_0)_{\lim}$ and shown in the **Fig. 3**.

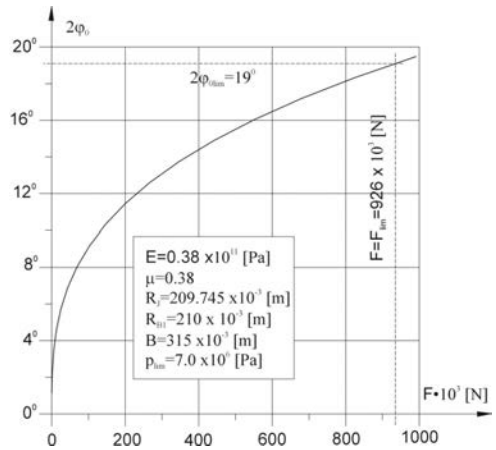

Fig. 3. The influence of a bearing load (F) on the contact angle of a journal and a bushing surface ($2\varphi_0$)

Rys. 3. Wpływ obciążenia łożyska (F) na kąt kontaktu powierzchni czopa i panewki ($2\varphi_0$)

a) b)

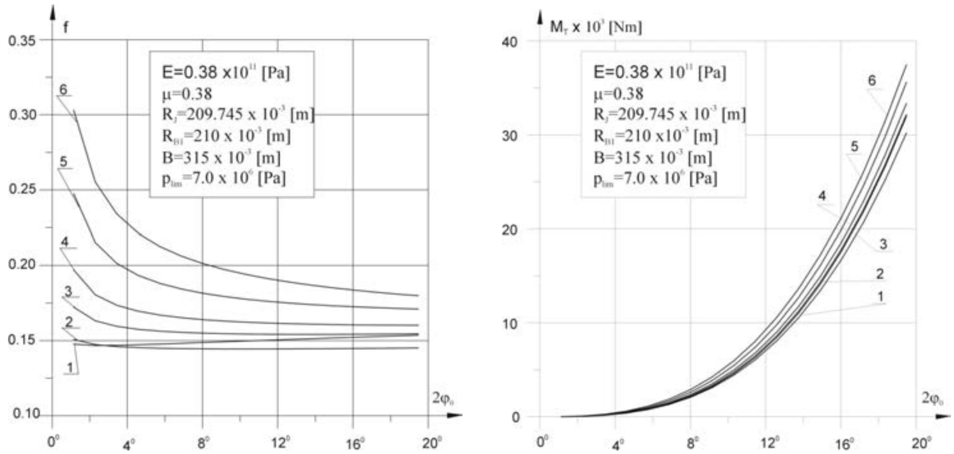

Fig. 4. a) The influence of the contact angle of a journal and bushing surface ($2\varphi_0$) on the coefficient of static friction (f);
b) The influence of the contact angle of a journal and bushing surface ($2\varphi_0$) on the frictional moment on the bearing journal surface (M_T)

Rys. 4. a) Wpływ kąta kontaktu powierzchni czopa i panewki ($2\varphi_0$) na współczynnik tarcia statycznego (f); b) Wpływ kąta kontaktu powierzchni czopa i panewki ($2\varphi_0$) na moment tarcia na powierzchni czopa łożyskowego (M_T)

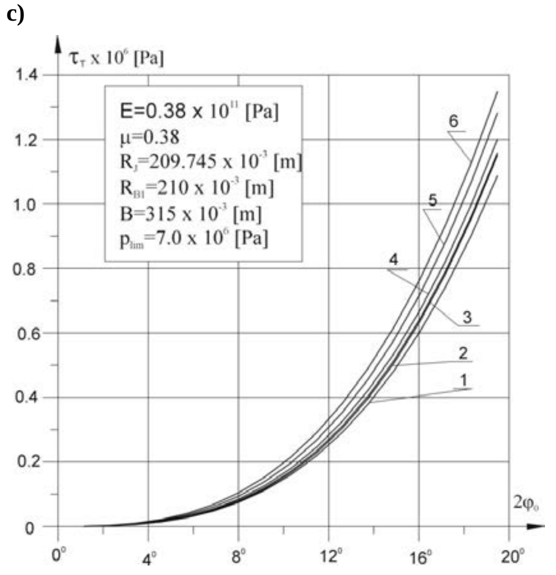

Fig. 5. c) The influence of the contact angle of a journal and bushing surface ($2\varphi_0$) on tangential stresses on bushing surface (τ_T), in a journal slide bearing: 1 – rough turning Δ = 7.9 x 10^{-1}, ν = 1.9, 2 – rough grinding Δ = 1.6 x 10^{-1}, ν= 2.0, 3 – rough honing Δ = 1.2 x 10^{-1}, ν = 1.7, 4 – precise turning Δ = 6.3 x 10^{-2}, ν= 1.6, 5 – precise grinding Δ = 2.8 x 10^{-2}, ν = 1,5, 6 – precise honing Δ = 4.65 x 10^{-3}, ν= 1.6

Rys. 5. c) Wpływ kąta kontaktu powierzchni czopa i panewki ($2\varphi_0$) na naprężenia styczne na powierzchni panewki (τ_T): 1 – toczenie zgrubne Δ = 7.9 x 10^{-1}, ν = 1.9, 2 – szlifowanie zgrubne Δ = 1.6 x 10^{-1}, ν= 2.0, 3- dogładzanie zgrubne Δ = 1.2 x 10^{-1}, ν = 1.7, 4 – toczenie dokładne Δ = 6.3 x 10^{-2}, ν = 1.6, 5 – szlifowanie dokładne Δ = 2.8 x 10^{-2}, ν = 1,5, 6 – dogładzanie dokładne Δ = 4.65 x 10^{-3}, ν = 1.6

ANALYSIS OF RESULTS

Studying the curves in **Figure 3**, reveals that an increase of bearing load promotes an increase in the contact angle of the journal and bushing surface. The types and accuracy of the machining operation have an essential influence on the friction coefficient, the frictional moment, and the values of tangential stresses. **Figure 4** shows that, if the contact angle increases ($2\varphi_0$), the friction coefficient value decreases (f), and frictional moment values (M_T) and tangential stresses values (τ_T) increase. The growth of the angle ($2\varphi_0$) reduces the influence of machining type on the friction coefficient value. The examination results for angles $2\varphi_0 = 5^0$ and $2\varphi_{0lim} = 19^0$ are presented in **Table 2**.

Table 2. The influence of machining type and accuracy on the friction coefficient and frictional moment in a lateral slide bearing

Tabela 2. Wpływ rodzaju i dokładności obróbki na współczynnik tarcia i moment tarcia w poprzecznym łożysku ślizgowym

Machining type	$2\varphi_0=5^0$, $F=16.623 \times 10^3$[N]		$2\varphi_0=19^0$, $F_{lim}=926 \times 10^3$[N]	
	f	M_T [Nm]	f	M_T [Nm]
accurate honing $\Delta=4.65 \times 10^{-3}$, $\nu= 1.6$	0.218	7.60×10^2	0.180	3.51×10^4
accurate grinding $\Delta=2.8 \times 10^{-2}$, $\nu= 1.5$	0.191	6.68×10^2	0.171	3.33×10^4
accurate Turing $\Delta=6.3 \times 10^{-2}$, $\nu= 1.6$	0.169	5.88×10^2	0.160	3.12×10^4
rough honing $\Delta=1.2 \times 10^{-1}$, $\nu= 1.7$	0.157	5.48×10^2	0.154	3.00×10^4
rough Turing $\Delta=7.9 \times 10^{-1}$, $\nu= 1.9$	0.147	5.15×10^2	0.153	2.99×10^4
rough grinding $\Delta=1.6 \times 10^{-1}$, $\nu= 2.0$	0.145	5.10×10^2	0.145	2.82×10^4

Functions $M_T = M_T(\varphi_0)$, $\tau_T = \tau_T(\varphi_0)$, described in **Figs. 4** and **5** are increasing. The smallest values of tangential stresses and frictional moments appear in rough grinding, i.e. $\Delta = 1.6 \times 10^{-1}$, $\nu = 2.0$, while the highest values appear in accurate honing: $\Delta = 4.65 \times 10^{-3}$, $\nu = 1.6$. Additionally, machining accuracy has an essential influence on the friction coefficient value during bearing start-up. For angle $2\varphi_0 = 5^0$, a change of grinding accuracy from a rough machining with parameters ($\Delta = 1.6 \times 10^{-1}$, $\nu = 2.0$) into an accurate machining with parameters ($\Delta = 2.8 \times 10^{-2}$, $\nu = 1.5$) causes an increase of the friction coefficient by 31%. While, for the angle $2\varphi_0 = 19^0$, the increase is 18%. Consequently, it can be noticed that the machining type and the accuracy of a journal surface have at significant impact on the bearing system during its start-up. The research work reveals that an introduction of a static friction issue into calculation methods can improve in the essential operations of slide bearings.

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Streszczenie

Zjawisko tarcia spoczynkowego podczas rozruchu łożysk ślizgowych jest procesem złożonym. Do analizy zjawisk przyjęto molekularno-mechaniczną teorię tarcia. W pracy przedstawiono wyniki badań opisujące wpływ struktury geometrycznej powierzchni czopa na współczynnik tarcia spoczynkowego, moment tarcia i naprężenia styczne na powierzchni panewki. W analizie wyników badań wykazano, że rodzaj i dokładność obróbki powierzchni czopa w istotny sposób wpływa na właściwości węzła łożyskowego podczas jego rozruchu.