Aleksander MAZURKOW*

METHOD FOR DETERMINATION OF THE STATIC FRICTION FACTOR IN SLIDE BEARINGS

METODA WYZNACZANIA WSPÓŁCZYNNIKA TARCIA SPOCZYNKOWEGO W ŁOŻYSKACH ŚLIZGOWYCH

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

Abstract

This research work describes a method for the determination of the static friction factor in slide bearings. To determine tangential stresses distribution in a real area of a contact the author uses a molecular-mechanical theory of friction. In this theory, the total frictional force is equal to the sum of system of forces: molecular and material resistance to the deformation of bodies' surface coating. To design a model the following assumptions are accepted: a journal surface is rough, and a bushing surface is perfectly smooth. The results are represented in the form of graph which

* Rzeszow University of Technology, Faculty of Mechanical Engineering and Aeronautics, al. Powstańców Warszawy 8, 35-959 Rzeszów, phone: (017) 8651640, fax: (017) 8651150, e-mail: almaz@prz.edu.pl.

shows an influence of the contact angle of a journal surface and a bushing on a load, a frictional moment, tangential stresses, and a friction coefficient.

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 bush [m], R_{B2} – outer radius of bushing and [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].

ASSUMPTIONS FOR A FRICTION MODEL CONSTRUCTION

When a slide bearing starts operating **(Fig. 1)** the essential structural parameter that has an influence on this operation is a static friction coefficient **[L. 1]**, **[L. 5**, **6]**, and **[L. 9]**.

This research suggests a method that determines a static friction coefficient with the use of a molecular-mechanical theory of friction.

The theory was developed by Kragelsky and discussed in these works **[L. 3, 4]**. The molecular–mechanical theory of friction accepts that friction dissipation results from tangential stresses. Tangential stresses are examined on a real area of a contact where its geometrical structure is taken into consideration. The total frictional force is equal to the sum of system of forces: molecular and material resistance to deformation of bodies' surface coating.

To characterize frictional forces in slide bearings at rest, the author accepts the following assumptions:

– A deformation of a shaft surface coating is essentially small in comparison with a deformation of a bushing.
– A frictional force and pressure are examined on the contact area between a journal and a bushing. (This area is called contour one.)
– A real contact between a journal and a bushing is studied by means of micro roughness, assuming that the journal surface is rough and the bushing surface is perfectly smooth.
– A deformation of a bearing bushing is examined as an elastic deformation.
– Micro roughness of a surface is described by a profile capacity function.
– A bearing journal is made of a very hard material while a bushing is made of an elastic-plastic material.
– A bearing journal temperature does not have an influence on the material-physical properties of a journal and a bearing bushing.

EQUATIONS OF A MATHEMATICAL MODEL TO DETERMINE A FRICTIONAL MOMENT AND A STATIC FRICTION COEFFICIENT

The above assumptions suggest that a friction force can be described as the sum of forces of a molecular interaction between the 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.

A resultant friction force can be written as an equation as follows:

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

The friction force molecular component (F_{Tm}) is equal to a product of the real friction surface and tangential stresses that appear on this surface:

$$F_{Tm} = A_r \cdot \tau_{Tm} \tag{2}$$

The friction force component (F_{Td}) depends on a material surface coating deformation in the bushing. This relationship is expressed by the following equation **[L. 2, 5]**:

$$F_{Td} = \frac{0{,}25 \cdot \alpha_{ef} \cdot h_i^2 \cdot E}{1-\mu^2} \tag{3}$$

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

$$M_T = f \cdot F \cdot R_{B1} \tag{4}$$

The frictional moment can be calculated from the following relation:

$$M_T = 2 \cdot R_{B1}^2 \cdot B \cdot \int_{-\varphi_0}^{\varphi_0} \tau_T \cdot d\varphi_0 \tag{5}$$

The friction coefficient on a journal surface is equal to:

$$f = \frac{2 \cdot R_{B1}^2 \cdot B \cdot \int_{-\varphi_-}^{\varphi_0} \tau_T \cdot d\varphi_0}{F \cdot R_{B1}} = \frac{2 \cdot R_{B1} \cdot B \cdot \int_{-\varphi_0}^{\varphi_0} \tau_T \cdot d\varphi_0}{F} \tag{6}$$

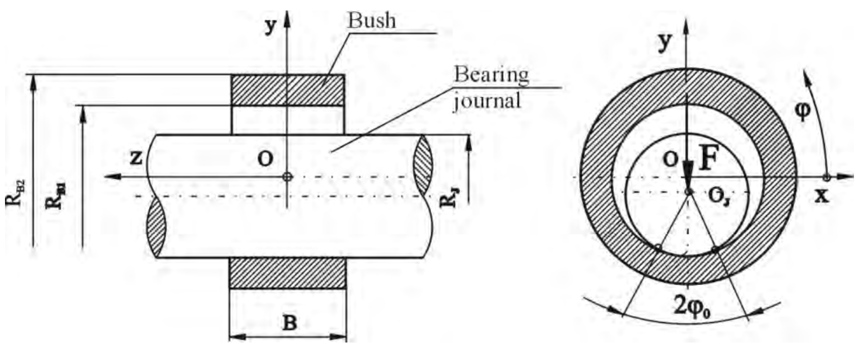

Fig. 1. Slide bearing geometry
Rys. 1. Geometria łożyska ślizgowego

Equations describing a journal micro geometry

A micro roughness model **[L. 7]** is taken into consideration and is shown in **Figure 2**. The micro geometry of a journal surface is described by the following:

- A real friction surface which refers to a single micro roughness can be written in the following terms: $$A_r = \alpha \cdot 2 \cdot \pi \cdot R \cdot h \tag{7}$$
- The profile capacity [7] function is: $$t_p = b \cdot \varepsilon_h^{\nu} \tag{8}$$

- The roughness dimensionless [7] coefficient is: $\Delta = \dfrac{R_{\max}}{R \cdot b^{\frac{1}{\nu}}}$ (9)
- The penetration depth of roughness peaks of a journal in a bushing surface is: $h = \left[\dfrac{5 \cdot \sigma_n \cdot R^{0.5} \cdot \left(1 - \mu^2\right) \cdot R_{\max}^{\nu}}{b \cdot \nu \cdot (\nu - 1) \cdot k_1 \cdot E} \right]^{\frac{2}{2\nu+1}}$ (10)

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

$$A_r = \alpha \cdot A_c \cdot t_p = \alpha \cdot A_c \cdot b \cdot \varepsilon^{\nu},$$

where: $A' = \dfrac{A_r}{A_c} = \alpha \cdot \left[\dfrac{5 \cdot \sigma_n \cdot R^{0.5} \cdot \left(1 - \mu^2\right)}{\nu \cdot (\nu - 1) \cdot k_1 \cdot E \cdot \Delta^{\frac{1}{2}}} \right]^{\frac{2\nu}{2\nu+1}}$ (11)

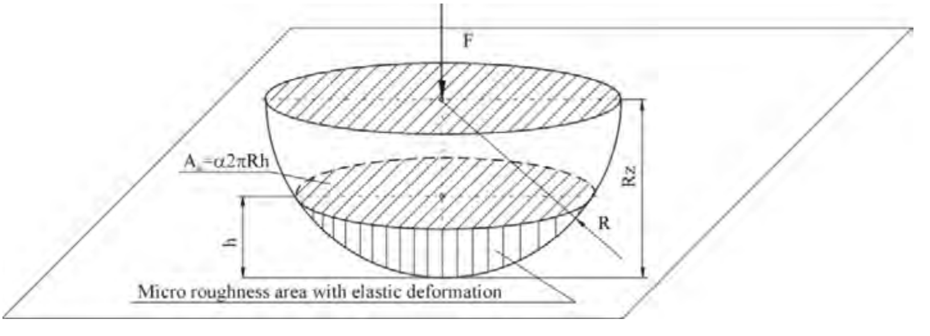

Fig. 2. Micro roughness of a journal surface model
Rys. 2. Model mikronierówności powierzchni czopa

Tangential stresses referring to a real contact area of journal and bushing surfaces

Tangential stresses in a contact area resulting from a molecular interaction and a stress in deformation areas can be written in as the following equation:

$$\tau_T = \tau_{Tm} + \tau_{Td} \tag{12}$$

Stresses arising from a molecular interaction on the solids' boundary can be described by this function **(Fig. 3)**:

$$\tau_{Tm} = \tau_{To} + \beta \cdot \sigma_n \tag{13}$$

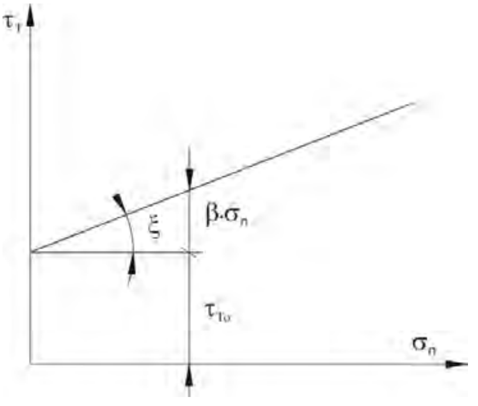

Fig. 3. Tangential stresses as a function of a normal stresses
Rys. 3. Zależność naprężeń stycznych w funkcji naprężeń normalnych

It has been assumed that normal stresses distribution in a contact area of a journal and a bushing can be written as the following function **[L. 8]**:

$$\sigma_n(\varphi) = \frac{E \cdot C_R}{g_B} \cdot \left(\frac{\cos\varphi}{\cos\varphi_0} - 1 \right) \cdot \frac{(1-\mu)}{(1+\mu)\cdot(1-2\cdot\mu)} \tag{14}$$

On the basis of the assumption that a molecular interaction occurs in the real contact area of the journal and bushing, a molecular component of tangential stresses can be written in the following terms **[L. 2]**:

$$\tau_{Tm} = \frac{\tau_{T0}}{2} \cdot \left[\frac{5 \cdot \sigma_n \cdot (1-\mu^2)}{\nu \cdot (\nu - 1) \cdot k_1 \cdot E \cdot \Delta^{\frac{1}{2}}} \right]^{\frac{2}{2\cdot\nu+1}} + \beta \cdot \sigma_n \tag{15}$$

Whereas, tangential stresses in the micro roughness deformation area of solid surfaces **[L. 2]** will be the following:

$$\tau_{Td} = \frac{T_d}{A_{ri}} \cdot A' \tag{16}$$

Taking into account these relationships (7), (10), (11) Equation (15), a tangential stress component in the deformation zone will equal to the following:

$$\tau_{Td} = 0,0796 \cdot 0.5 \cdot \alpha_{ef} \cdot E^{\frac{-1}{2\nu+1}} \cdot \left(1-\mu^2\right)^{\frac{1}{2\nu+1}} \cdot \Delta^{\frac{\nu}{2\nu+1}} \cdot \left[\frac{5 \cdot \sigma_n}{\nu \cdot (\nu-1) \cdot k_1}\right]^{\frac{2\nu+2}{2\nu+1}} \quad (17)$$

CALCULATION EXAMPLE

To carry out the calculation, the author has accepted a journal slide bearing and described its geometry and material properties in the **Table 1**. The following values are examined for given load (F): $2\varphi_0$ is the contact angle of a journal and bushing surface, M_T is the frictional moment on the bushing surface [Nm], τ_T is the tangential stress on the bushing surface [N/m^2], and f is the frictional coefficient on a bushing surface.

Results are given in the form of a functions: $F = F(2\varphi_0)$, $M_T = M_T(2\varphi_0)$, $\tau_T = \tau_T(2\varphi_0)$, $f = f(2\varphi_0)$ and presented in the **Figure 4**.

Table 1. Structural task, calculation example
Tabela 1. Zadanie konstrukcyjne, przykład obliczeniowy

Given values	
LOAD AND VALUES DESCRIBING A BEARING GEOMETRY	
F – load [N]	200 – $1.62 \cdot 10^6$
R_J – journal radius [m]	$209.745 \cdot 10^{-3}$
R_{B1} – inner radius of bearing bushing [m]	$210.0 \cdot 10^{-3}$
R_{B2} – outer radius of bearing bushing [m]	$214.0 \cdot 10^{-3}$
B – bushing width [m]	$315.0 \cdot 10^{-3}$
Materials properties for a bearing bushing	
E – Young's modulus for bushing material [N/m^2]	$0.38 \cdot 10^{11}$
μ – Poisson's modulus for bushing material	0.38
τ_{T0} – shear resistance of adhesive bonds [N/m^2]	$8.0 \cdot 10^6$
β – molecular component of friction resistance	0.065
α_{ef} – losses coefficient caused by hysteresis deformation of micro roughness on bushing surface	0.1
k_1 – constant dependent on capacity curve parameters	0.4
Values describing a surface coating structure	
Δ – micro roughness dimensionless coefficient	0.18
ν – parameters of capacity profile curve	2.3

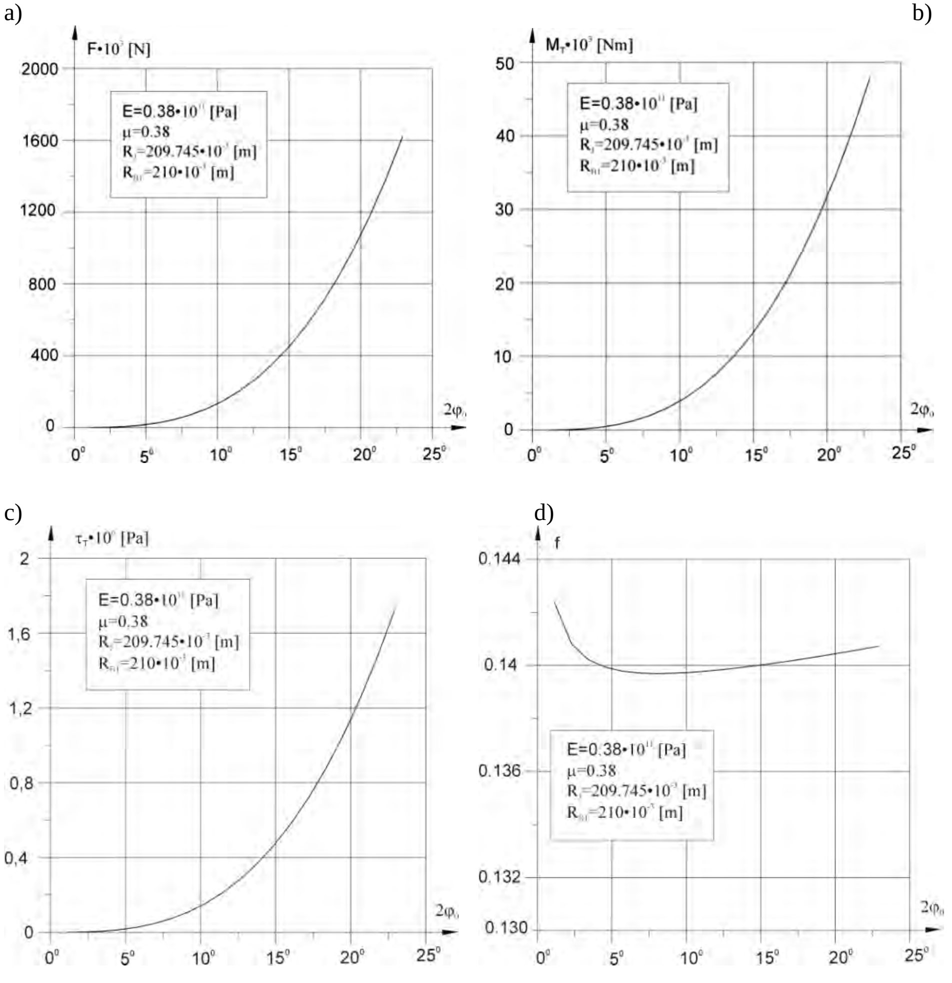

Fig. 4. Functions: load (F), frictional moment on a journal surface (M_T), tangential stresses on a journal surface (τ_T), f – static friction coefficient in dependence on a contact angle of a journal and a bushing surfaces ($2\varphi_0$)

Rys. 4. Funkcje: obciążenie (F), moment tarcia na powierzchni czopa (MT), naprężenia styczne na powierzchni czopa (tT), f – współczynnik tarcia spoczynkowego w zależności od kąta kontaktu powierzchni czopa i panewki (2j0)

Studying the functions in **Figure 4**, it can be noticed that an increase of contact angle ($2\varphi_0$) promotes an increase of load (F), frictional moment (M_T) and tangential stress (τ_T) while, friction coefficient (f) decreases. Furthermore, there can be observed the values of load, frictional moment, and tangential stress change slightly for given bearing and angles $2\varphi_0<5^0$.

CONCLUSION

The developed method helps to calculate journal slide bearings with taking into consideration motion resistance on a boundary of a journal and bearing bushing. The discussed model describes materials properties, including the macro and micro geometry of a bearing.

The developed model can be used for the analysis of slide bearings properties at the beginning of motion.

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Streszczenie

W pracy przedstawiono metodę wyznaczania współczynnika tarcia spoczynkowego w łożyskach ślizgowych. Do określenia rozkładu naprężeń stycznych w rzeczywistym obszarze styku przyjęto molekularno-mechaniczną teorię tarcia. W teorii tej całkowita siła tarcia jest równa sumie sił

składowych: molekularnej i odporności materiałów na deformowanie warstwy powierzchniowej ciał. W budowie modelu przyjęto m.in. założenia: powierzchnia czopa jest chropowata, natomiast powierzchnia panewki jest idealnie gładka. Wyniki obliczeń, wpływ kąta kontaktu powierzchni czopa i panewki na obciążenie, moment tarcia, naprężenia styczne, współczynnik tarcia przedstawiono w formie wykresów.