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A NON-CONVENTIONAL METHOD FOR THE IMPROVEMENT OF THE FUNCTIONAL PROPERTIES OF SLIDING PAIRS

One of the ways to improve the functional properties of sliding friction pairs is to apply heterogeneous rubbing surfaces. Although this approach is still under investigation, heterogeneous surfaces are commonly employed in friction pairs under extreme lubrication conditions. The paper is concerned with the modeling of the friction process of flat textured sliding surfaces. The results can be used for the design of sliding friction pairs operating under extremely high loads.

***Keywords**: sliding friction pairs, heterogeneous surfaces, geometrical texture.*

1.Introduction

Numerous studies have been undertaken to establish the effect of various physical, chemical and mechanical phenomena on the friction and wear processes. Advanced experimental methods make it possible to determine precisely the real loads that machine parts will be exposed to and select the right material for a friction pair to eliminate its damage or failure. Since the most frequent cause of the product malfunction is wear due to friction, a lot of attention is paid to the development of methods of friction reduction.

The number of materials that can be used for this purpose and technologies to harden these materials is limited. Designers carefully select the geometrical features of friction pair elements to reduce to a minimum the negative effects of friction. The selection is made both at the micro and macro levels, which is possible thanks to rapid advances in technology.

As was observed long ago, in certain cases of lubrication, pores or other specially generated (honed) cavities help trapping lubricant films and reducing the probability of the occurrence of loads leading to seizure of a friction pair. The paper discusses a nonconventional method of friction reduction in a sliding pair that involves applying heterogeneous surfaces. In conventional methods friction is reduced by using special techniques, lubricants, materials and self-lubricating films.

2. Heterogeneous surfaces

Surfaces are called heterogeneous when they possess regularly distributed areas characterized by different geometrical, physical-mechanical and physical-chemical properties. The areas constituting surface heterogeneities are generated by applying a technology different from that used to produce the rest of the surface. Thus, surface heterogeneities can include:

- cavities around the surface of the sliding rings, e.g. grooves, channels, form cavities produced by milling, erosion, etching, laser treatment, etc.,
- areas with different physical-chemical and mechanical properties, e.g. surfaces locally differing in hardness and mechanical strength due to surface hardening (laser, electron or thermal-chemical treatment),
- areas with different surface microgeometry, e.g. areas obtained by point erosion (laser treatment) or ones with shaped surface microgeometry, e.g. specially designed orientation of microirregularities or surface load capacity (laser treatment and electrospark deposition).

The regular cavities around the sliding surface produced with different techniques are reported to be of benefit to the friction process. The increased amount and better circulation of lubricant, which is trapped in the cavities, result in a lower temperature and more favourable distribution of pressure in the clearance, which improves the load capacity of the sliding pair [1, 2, 3].

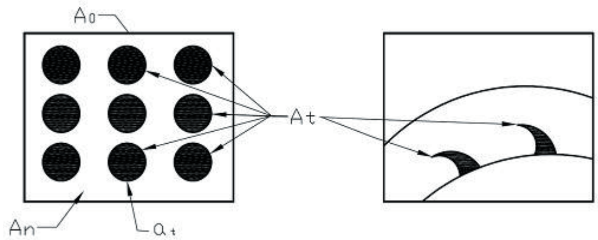

Fig. 1. Examples of heterogeneous surfaces: A_t – total area of heterogeneity (cavities, hardened areas), a_t – area of a single heterogeneity, A_n – flat surface, A_o- nominal surface

It should be noticed that the system of regular heterogeneities constitutes the geometrical surface texture, which may cover the whole area of the sliding friction pair or only selected areas regularly distributed around this surface. Figure 1 shows examples of heterogeneous surfaces.

2.1. Analysis of the geometry of contact and load

The analysis of the geometry of surfaces whose depth of heterogeneity is comparable to that of the height of roughness involves determining standard roughness parameters and the load curve. However, in a number of cases, the depth of surface heterogeneity is considerably higher than the height of surface roughness. It may turn out that the geometry of cavities is of importance in the formation of real contact. Surface heterogeneity should be considered in three aspects:

- cavities at the friction interface constitute the lubricant traps,
- cavities reduce the contact area and increase the load; in this way, they affect the value of the coefficient of friction,
- cavities at the friction interface have influence on the distribution of pressure in the clearance, which may cause changes in the load capacity of the sliding pair.

The changes in the load profile may be due to load and wear. Figure 2 analyzes changes in the load capacity for a profile form with spherical cavities. The strain due to load or wear causes a decrease in the depth of cavities by the value h, so the cavity radius decreases to the value R_h:

$$R_h = A_1B_1 = \sqrt{R^2 - (R - h_0 + h} = \sqrt{(h_0 - h)(2R - h_0 + h)} \quad (1)$$

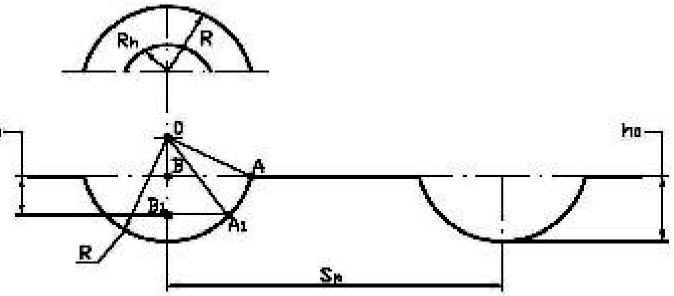

Fig. 2. Schematic diagram of the geometrical surface structure with spherical cavities

For 2R>>h $R_h = A_1B_2 = (R - h_0)^{\frac{1}{2}}(h_0 - h)^{\frac{1}{2}}$ (2)

The load capacity α (ratio of the surface without cavities to the nominal surface) will change into the value α_h:

$$\alpha = \frac{A_n}{A_0} \qquad \alpha_h = \frac{A_h}{A_0} \tag{3}$$

where : $A_n = A_0 - k\pi R^2 \qquad A_h = A_0 - k\pi R_h^2$

Applying force F, we obtain the following relationship for the load against the surface:

$$\sigma_h = \frac{F}{A_h}$$

After substituting relationships (2) and (3), we get:

$$\sigma_h = \left(\frac{1}{A_0 - k\pi (R - h_0)(h_0 - h)} \right) \sigma_0 \tag{4}$$

2.2. Analysis of friction resistances

Assuming the regularity of the cavities around the rubbing surfaces, we can develop a model of friction which takes into account both the area of flat surfaces and the area with cavities, here called the area of wavy surfaces. In the model, the flat surfaces are ideally flat, and the porous ones have roughness with equivalent parameters [4, 5].

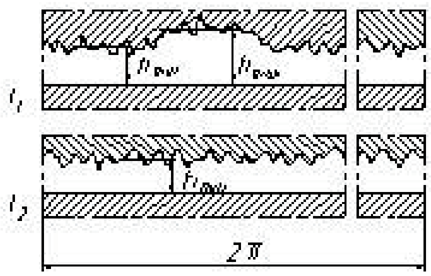

Fig. 3. Model of the clearance: fragment of surface with the cavitation area (upper part) and a fragment of surface without cavities (lower part)

In the general case, the total load force of a flat sliding pair with regular cavities around the opposing surfaces is expressed by the relationship:

$$W = W_m + W_h \tag{5}$$

where : W_m - mechanical component of the load force, W_h- hydraulic component of the load force.

The mechanical component of the load force resulting from the contact of the peaks of the microirregularities after taking into consideration the cavities [6] has the following form:

$$W_m = \alpha \cdot A \cdot P(H > h) \cdot p_g \tag{6}$$

where : A – area of microirregularities taking part in the transmission of loads, p_g - shearing strength of the softer material, $P(H>h)$ – probability of the occurrence of the assumed clearance height.

Because of the surface heterogeneity, the load force of the fluid film has two components:

$$W_h = W_{h1} + W_{h2} \tag{7}$$

where : W_{h1} – load force of the fluid film in the areas with no cavities, W_{h2} – load force of the fluid film in the cavitation areas.

For areas without cavities, the relationship is:

$$W_{h1} = \frac{\mu \cdot U}{h^2} \alpha \cdot A \cdot L \cdot f(L, \Delta r) \tag{8}$$

where: $f(L,\Delta r)$ – function determining the probability of occurrence of the fluid film [7], μ - viscosity of the fluid in the clearance, U – sliding speed, $L,\Delta r$ – length and width of the friction area.

For the cavitation areas, we adapt the relationship given by Lubeck [8]:

$$W_{h2} = (1 - \alpha) A \left[\frac{k \cdot \mu \cdot \omega}{8\pi} \cdot \frac{(\Delta r)^2}{(h_{min})^2} \cdot f_1\left(\frac{h_a}{h_{min}}\right) + \frac{p_0}{4} \right] \tag{9}$$

where, additionally:

$$f_1\left(\frac{h_a}{h_{min}}\right) = 1 - \frac{1}{\left(1 + 2\frac{h_a}{h_{min}}\right)^2}$$

h_a - waviness amplitude, h_{min} – minimum clearance height, ω - rotational speed, p_0 - fluid pressure in the clearance.

Finally, taking into account the above, we obtain the relationship for the load force of the flat sliding pair:

$$W = W_m + \frac{\mu \cdot U}{h^2} \alpha \cdot A \cdot L \cdot f(L, \Delta r) + \\ +(1 - \alpha) A \left[\frac{k \cdot \mu \cdot \omega}{8\pi} \cdot \frac{(\Delta r)^2}{(h_{min})^2} \cdot f_1\left(\frac{h_a}{h_{min}}\right) + \frac{p_0}{4} \right] \tag{10}$$

For simplicity, we shall consider a real case of a flat sliding pair, i.e. a pair of rings of a face seal, which is shown in Fig. 4.

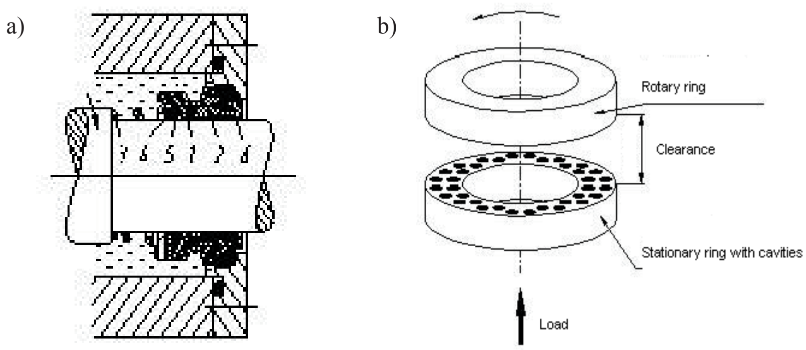

Fig. 4. a) Schematic diagram of the face seal: 1- axially shifted sliding ring, 2- anti-ring, 3- spring, 4- clamping ring, 5,6 – secondary seals, b) model sliding pair with textured surface

Case 1

Let us consider the case when $h_{min} > h_c$, where h_c is the clearance height for which the peaks of the microirregularities are in contact. The clearance height is big enough so there is no contact of the microirregularities.

Since $W_m = 0$, relationship (10) can be written as:

$$W = \frac{\mu \cdot U}{h^2}\alpha \cdot A \cdot L \cdot f(L,\Delta r) + \\ +(1-\alpha)A\left[\frac{k \cdot \mu \cdot \omega}{8\pi} \cdot \frac{(\Delta r)^2}{(h_{min})^2} \cdot f_1(\frac{h_a}{h_{min}}) + \frac{p_0}{4}\right] \tag{11}$$

In the general case, the friction force between the rubbing surfaces will be a sum of two components:

$$F = F_h + F_m \tag{12}$$

where: F_h – friction force of the fluid film, F_m – friction force of the peaks of the microirregularities in contact.
Thus, when $h_{min} > h_c$, $F_m = 0$.

The friction force of the viscous fluid, F_h, can be described by a known relationship:

$$F_h = A\frac{\mu \cdot U}{h} \tag{13}$$

Let us consider the case of complete fluid film with a variable thickness. If we base the considerations on the works by Stanghan-Batch and Ina [6] and Lebeck [8], we can calculate the coefficient of friction as follows:

$$f = \frac{\pi \cdot G \cdot r_m}{f_2 \cdot h_{min}} \tag{14}$$

where:

$$f_2 = \sqrt{1 + 2\left(\frac{h_a}{h_{min}}\right)} \tag{15}$$

In relationship (14), h_{min} is an unknown quantity, so it needs to be determined from other conditions. As the axial forces are in equilibrium, the load force in the clearance of the flat sliding pair of a face seal is counterbalanced by the external axial forces acting on the axially flexible element of the sliding pair. Thus, the condition of the equilibrium of the axial forces:

$$W = A \cdot p_0(b + \frac{p_s}{p_0}) = A \cdot p_0 \cdot b^{'} \tag{16}$$

where: p_s – spring tension, b – coefficient of load, b' – corrected coefficient of load:

$$b^{'} = b + \frac{p_s}{p_0} \tag{17}$$

Comparing relationship (11) with (16) and including the size of the friction area, we can calculate the clearance height:

$$h_{min} = \sqrt{\frac{\mu \cdot U \cdot \alpha \cdot L \cdot f(L,\Delta r) + (1-\alpha)\frac{k \cdot \mu \cdot U}{8\pi \cdot r_m}\Delta r^2 f_1(\frac{h_a}{h_{min}})}{p_0 \cdot b' - (1-\alpha)\frac{p_0}{4}}} \tag{18}$$

Using the definition of parameter G, we calculate the minimum clearance height:

$$h_{min} = C_1\sqrt{G} \tag{19}$$

where:

$$C_1 = \sqrt{\frac{\frac{1-\alpha}{4}k \cdot (\Delta r)^2 f_1(\frac{h_a}{h_{min}}) + 4\alpha\pi^2 r_m^{\ 2} f(L,\Delta r)}{1 - \frac{1-\alpha}{4b'}}} \tag{20}$$

$$G = \frac{\mu \cdot U \cdot \Delta r}{W} \tag{21}$$

Substituting the calculated clearance height to relationship (14) and performing certain transformations, we obtain the value of the coefficient of friction:

$$f = \frac{\pi \cdot r_m \cdot \sqrt{G}}{f_2 \cdot C_1} \tag{22}$$

Case 2

We consider a situation when there is a direct contact of the peaks of the microirregularities, i.e. when $h_{min} = h_c$. The clearance height is known, as it results from the height of the microirregularities. Like in Refs. [9, 10], it is assumed that the distribution of the peaks of the microirregularities is the Gaussian distribution and the height the of microirregularities in contact, h_c, is 3σ, with σ being a standard deviation defined by the following relationship:

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \tag{23}$$

where: σ_1, σ_2 - standard deviations of random variables representing the profiles of both surfaces.

Like in the previous case, the load force of the seal resulting from the fluid film is calculated according to formulas (8) and (9). The value of the friction force of the fluid film is given by relationship (13), while the friction force of the microirregularities in contact is defined by the following relationship:

$$F_m = f_b \cdot W_m \tag{24}$$

where f_b - coefficient of boundary friction.

Calculating W_m from relationship (10), we get:

$$W_m = W - \frac{\mu \cdot U}{h^2}\alpha \cdot A \cdot L \cdot f(L,B) - \\ -(1-\alpha)A\left[\frac{n \cdot \mu \cdot \omega}{8\pi} \cdot \frac{(\Delta r)^2}{(h_{min})^2} \cdot f_1(\frac{h_a}{h_{min}}) + \frac{p_0}{4}\right] \tag{25}$$

Using relationships (5), (11) and (25) and performing certain transformations, we obtain the following relationship for the coefficient of friction:

$$f = f_b\left[1 - G \cdot f(L,\Delta r) \cdot \alpha \cdot \frac{4 \cdot \pi^2 \cdot r_m^{\ 2}}{h_{min}^{\ 2}} - (\frac{1-\alpha}{4}) \\ (G \cdot k \cdot f_1(\frac{h_a}{h_{min}})\frac{(\Delta r)^2}{(h_{min})^2} + \frac{1}{b'})\right] + \frac{2\pi \cdot r_m \cdot G}{f_2 \cdot h_{min}} \tag{26}$$

The model of friction assumes that the opposing surfaces are in contact or that there is no contact between them. The model can be used for the analysis of friction of seals with rings characterized by different geometries of surface heterogeneities.

2.3. Examples of the model analysis

The model was analyzed in a numerical example using the MATHCAD 7.0 program for the following data:

μ – 700 10^{-6} Pa s – dynamic viscosity of the medium,
μ h_c – 10^{-6} m – clearance height in the area of the flat surface,
h_a – 10^{-6} m – clearance height in the cavitation area,
k – 3, 6,.... 12 – number of cavities per surface unit,
α – 0.2...0.8 – share of the flat surface,
r_m = 0.0205 m – average ring radius,
Δr = 0.005 m – ring width.

The relationship between the coefficient of friction and parameter G was analyzed for different shares of the flat surface α and different number of surface structures k in the friction area. It was necessary to define the influence of these parameters on the coefficient of friction for the following cases $h > h_c$ and $h = h_c$

Figures 5 and 6 show typical relationships between the coefficient of friction and parameter G both for the regime of mixed friction and that of fluid friction. As can be seen in Fig. 5, in the mixed friction regime, the coefficient of friction decreases when there is a rise in parameter G. In the regime of fluid friction, an inverse relationship is observed (Fig. 6). This general relationship between the coefficient of friction and parameter G is not at all dependent on the number of cavities k or the share of flat surface α. The influence of parameters k and α is not analyzed in this paper.

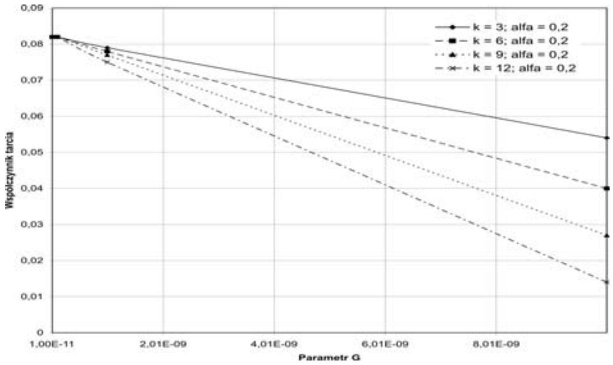

Fig. 5. Coefficient of friction between the face seal rings vs. parameter G in the regime of mixed friction for surfaces with a different number of cavities k and a constant share of flat surface α

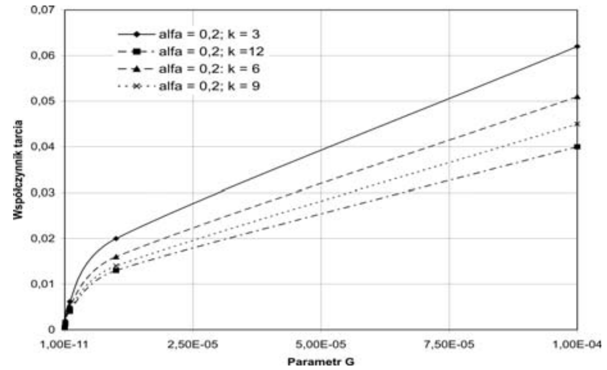

Fig. 6. Coefficient of friction between the face seal rings vs parameter G in the regime of fluid friction for surfaces with a different number of cavities

The proposed model of friction makes it possible to determine the values of the coefficient of friction and parameter G for which there is a transition from mixed friction to fluid friction. Figure 7 shows the values of the coefficient of friction recorded for both regimes. The transition point is in the range of parameters $G = 10^{-7} \div 10^{-8}$. The exact position of the transition point is shown in Fig. 8. The hypothetical values of the coefficient of friction f and parameter G at the transition point were established by comparing the mathematical model for mixed friction with that of fluid friction.

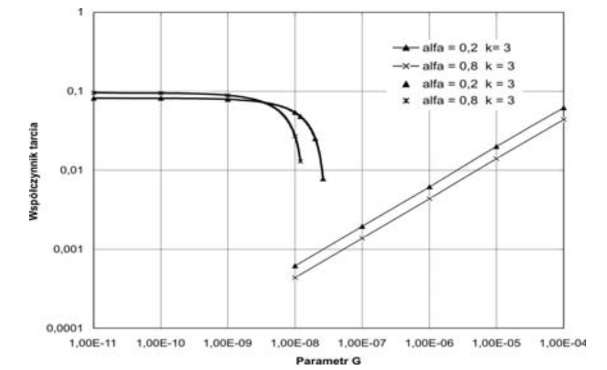

Fig. 7. Coefficient of friction between the face seal rings vs parameter G (logarithmic scale)

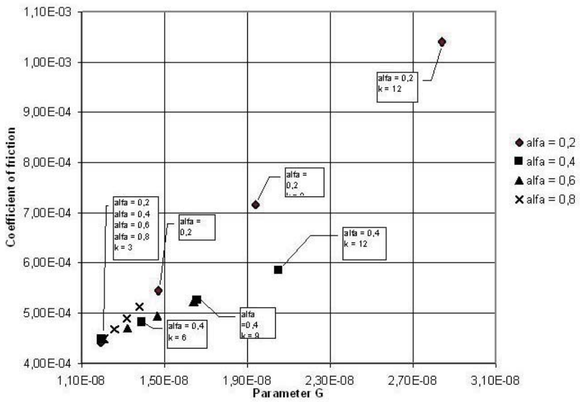

Fig. 8. Positions of the points of transition from mixed to fluid friction for the mating face seal rings

3.Conclusions

Analyzing the above diagrams, one can draw the following conclusions about face seal rings whose surfaces are partly flat α and partly with cavities *k*:

- the greater the number of cavities k on the surface of a face seal ring, the lower the coefficient of friction both in the regime of fluid friction and that of mixed friction,
- the greater the area of the flat surface *α*, i.e. the greater the share of the flat surface in the total area of friction in the regime of mixed friction, the greater the coefficient of friction; the relationship is true for all the considered number of cavities (Fig. 6.15),
- for fluid friction, an increase in the share of the flat surface α causes a decrease in the coefficient of friction, the exception being the case when $k = 12$ for which the coefficient of friction rises slightly with an increase in *α*,
- a rise in the value of parameter *G* causes that the type of friction between the seal rings changes from mixed into fluid; the developed model makes it possible to assess the parameters of the transition point depending on the number of cavities and the share of the flat surface.

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