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RESEARCH

Methodology of Calculation of Metal-polymer Sliding Bearings for Contact Strength, Durability and Wear

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A B S T R A C T

The methodology of calculation of hybrid (metal-polymer) sliding bearings by load capacity, durability and wear of the bushing made of reinforced polymer composites, in particular polyamide filled with glass or carbon dispersed fibers, is presented in the article. The technique of model triboexperimental studies is presented. Based on the results of these studies, the wear resistance diagrams of the investigated tribocouples of bearing materials were constructed and the characteristics of their wear resistance were determined. The specified contact and tribotechnical parameters have been calculated. Quantitative and qualitative regularities of their change from the main studied factors of influence are established: load, bushing diameter, radial clearance. Empirical relationships between maximum contact pressures and durability are determined depending on the factors specified.

1. INTRODUCTION

In modern mechanical engineering and other areas of human activity, various types of sliding bearings are widely used. This is because there are many applications where they have advantages over rolling bearings. Their diameters vary from 0.02 mm (sub-micromachines [1]) to 1100 mm (marine engines [2]), and widths vary from 0.01 mm to 1000 mm. They operate at loads much smaller than Newton and up to 20 million Newton at rotational speeds from rpm to hundreds of thousands of rpm. They are used in various operating conditions as in liquid (gas), boundary lubrication, and in dry friction. In their design, a different combination of materials of mating elements is used. The shaft is usually made of steel (cast iron), and the bushing can be made of various metal materials that work well with the material of the shaft (babbits, Cu alloys, Cu – Pb alloys, Sn-Pb-Cd alloys, Al alloys, Zn alloys). However, various non-metallic materials (carbon materials, layered composites, polymers, filled polymer-based composites) have become widespread in the manufacture of bushings. Such hybrid bearings work well in dry friction without the need for lubrication and in

conditions where the technological requirements do not allow the use of lubricant or when it is impossible to change it; have increased damping ability; low noise and other positive qualities [2]. Important among all non-metallic bushing materials are polymer composites based on polyamides, polycarbonates, polyimides, polyacetals, polyamidoimides, polyether ketones, polytetrafluoroethylene and others with various fillers that increase their durability. It should be noted that the number of new types of composite materials for metal polymer bearings is growing rapidly. In addition, the properties of the base polymers can be significantly improved by modifying various inclusions that have a positive effect on the processes of friction and wear [3,4].

Metal-polymer bearings consisting of a steel shaft and composite bushings of sufficient thickness usually operate in conditions of dry friction or insufficient lubrication at high coefficients of friction, which causes increase of bushings wear. In many cases, they operate in dry friction conditions where lubrication cannot be used or provided. However, the choice in the engineering practice of the optimal compositions of metal-polymer bearings is practically not based on scientific methods of design calculations. At present, there are no adequate standards for the calculation of metal-polymer bearings, and the known in the literature calculation [2, 5-13] and numerical methods for the study of metal [14-17] and metal-polymer [18-20] sliding bearings have not yet found practical application due to the use of various simplifying conditions of wear-contact interaction of moving parts. Obviously primarily because they are based on the well-known Archard law of abrasive wear, which assumes a linear dependence of wear on contact pressure and sliding speed. And this kind of wear with the micro-cutting mechanism is practically not allowed even in dry friction bearings. Also significantly different are the mechanical properties (6-8 times) and the Young's modulus (50-100 times) of shaft and bushing materials, which fundamentally affects the contact and wear characteristics of this tribotechnical slip system.

Therefore, the development of adequate to the actual operating conditions, efficient and sufficiently simple methods of calculating the load capacity, wear and durability (hours, rpm) of metal-polymer bearings at the design stage remains a crucial task. This research is dedicated to solve this pressing problem. For this purpose, the correct analytical methods [21,22] for calculating the wear of sliding bearings with metallic contacting elements, including the small non-circularity of shaft and bushing contours [22-26], were used. They are based on the methodology of the wear kinetics study at sliding friction as a process of friction - fatigue destruction of the surface layers of materials of tribosystem elements [21,27].

2. MATHEMATICAL MODEL OF THE WEAR KINETICS AT SLIDING FRICTION

It is assumed that there is a functional dependence of the wear rate (wear rate) and the specific friction force arising in the region of the tribocontact. Therefore, to describe the kinetics of wear by the fatigue mechanism in dry sliding friction according to [21,27], a system of linear differential equations is used:

$$\frac{1}{v}\frac{\mathrm{d}h_1}{\mathrm{d}t}\Phi_1(\tau)=1,\ \frac{1}{v}\frac{\mathrm{d}h_2}{\mathrm{d}t}\Phi_2(\tau)=1, \tag{1}$$

where v is the sliding speed; h_1, h_2 are the linear wear of tribosystem elements; t is the wear time; $\Phi(\tau)$ is the basic parameter of the model as the characteristic function of wear resistance of tribocouple materials for the accepted conditions of wear; $L=vt$ is the friction path.

As a parameter of the load of tribocontact at sliding friction is taken the specific friction force τ, the level of which determines the wear rate of the tribosystem elements. The Amonton-Coulomb formula is widely used for its definition

$$\tau = fp, \tag{2}$$

where f is the coefficient of friction; p is the contact pressure calculated by the methods of elasticity theory [5-9, etc.], in particular in this model by the methods given in [5, 22, 25-28].

The characteristic function $\Phi_i(\tau_i)$ of the wear resistance of materials at the accepted discrete values of the specific friction forces τ_i is calculated according to the results of tribo-experimental studies [21,27]

$$\Phi_i(\tau_i) = L_i/h_i, \tag{3}$$

where the linear wear h_i of samples is established as a result of model tribo-experimental studies, i is the loading levels.
The approximation of the discrete values of functions $\Phi_i(\tau_i)$ calculated according to expression (3) is performed by the following function:

$$\Phi_k(\tau) = B_k \frac{\tau_{k0}^{m_k}}{(\tau - \tau_{k0})^{m_k}}, \quad (4)$$

where B_k, m_k, τ_{k0} are wear resistance characteristics of materials in tribocouple, determined using the method of least squares, in particular τ_0 - the minimum value τ at which wear stops, k = 1; 2 is the numbering of tribosystem elements.

3. METHOD OF CALCULATING THE INITIAL CONTACT PRESSURES

The plane contact problem of elasticity theory about the internal contact of cylindrical bodies of close radii is considered, as is the case in the sliding bearing. Its design scheme is shown in Fig. 1.

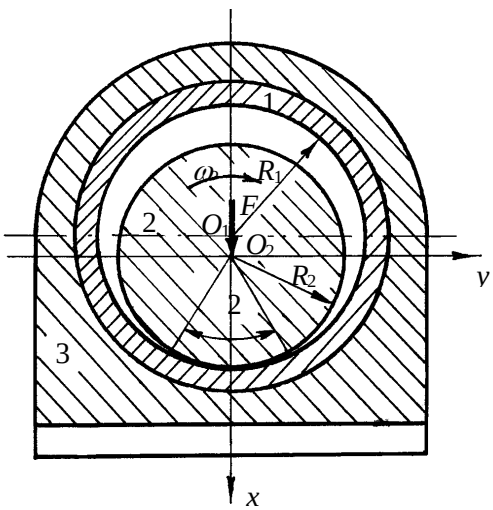


Fig. 1. The scheme of the sliding bearing.

The bearing journal is subjected to a radial load N, which in the plane problem is assigned to the length l of the journal, and then the concentrated load $F=N/l$ will be applied in the center of the shaft (disk). Shaft 2 (Fig. 1) rotates at a constant angular velocity ω_2. Between the journal of the shaft 2 and the bushing 1 is a radial clearance $\varepsilon=R_1-R_2\geq 0<<R$. The shaft is usually made of steel, and the bushing is made of other materials, so their wear resistance and elastic properties will be different. When the bearing force N is applied in the contact area $2R_2\alpha_0$, an initial contact pressure p_α arises, whose distribution law is unknown and which will decrease as the contact angle $2\alpha_0$ increases in the process of element wear.

To solve this plane contact problem of elasticity theory, the Kolosov-Muskhelishvili complex potentials for displacements and stresses are used, which are obtained on the basis of the theory of functions of a complex variable. To establish the original equation of the problem, the condition of equality of curvatures of compressed bodies in the region of their contact is used [22]. The following integer-differential equation is used to calculate the initial static contact pressures p_α arising in the contact area of the bearing elements [27]:

$$c_1 \int_{-\alpha_0}^{\alpha_0} \cot\frac{\alpha-\theta}{2} p'_\theta \, \mathrm{d}\theta = c_2 p_\alpha + c_3 \int_{-\alpha_0}^{\alpha_0} p_\alpha \, \mathrm{d}\alpha + c_4 \cos\alpha \int_{-\alpha_0}^{\alpha_0} p_\alpha \cos\alpha \, \mathrm{d}\alpha + \frac{\varepsilon}{R^2}, \quad (5)$$

where $p'_\theta = \mathrm{d}p/\mathrm{d}\theta$; α is the polar angle; $0\leq\alpha\leq\theta$, $0\leq\theta\leq\alpha_0$; $c_1 = \frac{1}{8\pi}\left(\frac{1+\kappa_1}{G_1R_1} + \frac{1+\kappa_2}{G_2R_2}\right)$; $c_2 = \frac{1}{4}\left(\frac{1-\kappa_1}{G_1R_1} - \frac{1-\kappa_2}{G_2R_2}\right)$; $c_3 = \frac{1+\kappa_1}{8\pi G_1R_1}$; $c_4 = \frac{1}{2\pi}\left(\frac{\kappa_1}{G_1R_1} + \frac{1}{G_2R_2}\right)$; $R_1\approx R_2=R$; G_1, G_2 are the shear modulus of materials; ν_1, ν_2 are the Poisson coefficients; $\kappa=3-4\nu$ is the Muskhelishvili constant, $E=2G/(1+\nu)$.

The approximate solution of equation (5) for determining p_α is performed by the collocation method for one collocation point $\alpha = \pm 0.5\alpha_0$. The contact pressure function is determined as [27]:

$$p_\alpha \approx E_0\varepsilon\sqrt{\tan^2\frac{\alpha_0}{2} - \tan^2\frac{\alpha}{2}}, \quad (6)$$

where $E_0 = (e_4/R_2)\cos^2(\alpha_0/4)$, $e_4 = 16G_1G_2(1+\nu_1)(1+\nu_2)/Z$, $Z = 2G_2(1+\kappa_1)(1+\nu_1)(1+\nu_2) + 2G_1(1+\kappa_2)(1+\nu_2)(1+\nu_1)$.

To determine the unknown half contact angle α_0 by the method of successive approximations, the condition of equilibrium of forces applied to the shaft is used:

$$F = R_2 \int_{-\alpha_0}^{\alpha_0} p_\alpha \cos\alpha \, \mathrm{d}\alpha = 4\pi R_2 E_0 \varepsilon \sin^2(\alpha_0/4). \quad (7)$$

4. METHOD OF CALCULATING TRIBOTECHNICAL CHARACTERISTICS OF THE BEARING

To determine the half tribocontact angle α_{0h} characterizing the contact zone during wear, a condition similar to condition (7) is used.

$$F = 4\pi R_2 E_0 (\varepsilon + c_{\alpha h}\varepsilon_h)\sin^2(\alpha_{0h}/4), \quad (8)$$

where $\varepsilon_h = h_{k\max}\left(-K_t^{(k)} + h_k'\right)$; $h_1' = h_2/h_1$, $h_2' = h_1/h_2$ are relative wears in the tribosystem; $K_t^{(1)}, K_t^{(2)}$ are the mutual overlapping factors bearing elements in the moving contact; $c_{\alpha h}$ is the rate in the tribocontact angle increase.

$$h_1' = \frac{h_2}{h_1} = \frac{\Phi_1(\tau)}{\Phi_2(\tau)} = \frac{B_1\tau_{10}^{m_1}(\tau-\tau_{20})^{m_2}}{B_2\tau_{20}^{m_2}(\tau-\tau_{10})^{m_1}}K_t^{(2)},$$

$$h_2' = \frac{h_1}{h_2} = \frac{\Phi_2(\tau)}{\Phi_1(\tau)} = \frac{B_2\tau_{20}^{m_2}(\tau-\tau_{10})^{m_1}}{B_1\tau_{10}^{m_1}(\tau-\tau_{20})^{m_2}}K_t^{(1)};$$

$$\tau = fp_0. \quad (9)$$

The tribocontact pressures acting in the bearing during the elements wear are determined as follows [21,25]:

$$p_{\alpha h} = p_\alpha + p_h, \quad (10)$$

where p_h is the change in initial pressures due to wear.

$$p_h = E_h\varepsilon_h\sqrt{\tan^2\frac{\alpha_{0h}}{2} - \tan^2\frac{\alpha}{2}}, \quad (11)$$

where $E_h = c_h(e_4/R_2)\cos^2(\alpha_{0h}/4)$, $c_h>0$ is the wear rate indicator.

The maximum contact pressures p_0 occur at α = 0. Then

$$p_0 \approx E_0\varepsilon\tan(\alpha_0/2). \quad (12)$$

After integrating the system of tribokinetic equations (1) taking into account the dependences (2), (4), (6), (10), (11), (12), (14) it is obtained the calculated ratio for the predictive estimate of the bearing durability at the given bushing wear $h_1=h_{k\max}$

$$t = \frac{-B_k\tau_{ko}^{m_k}\left\{\left[\tau-\tau_{k0}\right]^{1-m_k} - \left[(\tau-\tau_{k0}) + c_h h_{k\max}\Sigma_k\tau_h\right]^{1-m_k}\right\}}{vc_h\tau_h\Sigma_k(1-m_k)K_t^{(k)}}, \quad (13)$$

where $\Sigma_1 = (-K_t^{(1)} + h_1'), \Sigma_2 = \left(K_t^{(2)} - h_2'\right); K_t^{(1)} = 1, K_t^{(2)} = \alpha_0/\pi$.

Accordingly, the specific friction force

$$\tau_h = fp_{0h} = fE_h\tan(\alpha_{0h}/2). \quad (14)$$

After the transformations of expression (13) the formula for calculating the linear wear of the bearing bushing 1 and the shaft 2 at a given durability t_* is:

$$h_k = \left|\frac{1}{c_h S_h K_t^{(k)}\Sigma_k}\left[H_k - \sqrt[1-m_k]{\frac{L_k H_k^{1-m_k} - t_*}{L_k}}\right]\right|, \quad (15)$$

where $L_k = B_k\tau_{k0}^{m_k}\big/c_h vS_h(1-m_k)\Sigma_k K_t^{(k)}, H_k = \tau - \tau_{k0}$.

The linear wears h_k of the bearing elements are interconnected, i.e.

$$h_1 = h_2 h_2'/K_t^{(2)}, \quad h_2 = h_1 h_1'/K_t^{(1)}. \quad (16)$$

5. EXPERIMENTAL STUDY OF MATERIAL WEAR RESISTANCE

To determine the linear wear of material samples of bushings made of polymer-filled composites (glass and carbon-filled polyamide PA6), triboexperimental studies were performed according to the force scheme of end friction (two composite rods - steel disk) (Fig. 2) on a modernized Amstler friction machine. Research program: contact pressure p = 1, 2, 5, 10, 20 MPa, sliding speed v = 0.4 m / sec, friction path L = 2000 m, diameter of the pin specimen d = 4 mm. The wear mass loss of the samples ΔM was determined, and the average linear wear h of their contact surface was calculated by the formula:

$$h = \frac{\Delta M}{\rho S}, \quad (17)$$

where ρ is the density of the polyamide composite; S is nominal contact area.

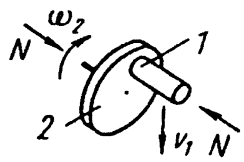


Fig. 2. Friction unit pin-on-disk.

The function value $\Phi_i(\tau_i)$ of material wear resistance at the accepted discrete values of the specific friction forces τ_i is calculated by formula (3), and their approximation is carried out by expression (4). As a result, according to the

above method, the wear resistance characteristics of the studied polyamide composites paired with steel were determined:
- glass fiber-reinforced polyamide PA6+30GF: B_{1GF} = 6.67·10^{10}, m_{1GF} = 1.9, τ_{10}= 0.05 MPa;
- carbon fiber-reinforced polyamide PA6+30CF: B_{1CF} = 24·10^{10}, m_{1CF} = 1.9, τ_{10}= 0.05 MPa.

Accordingly, in Fig. 3 points show the values of wear resistance functions $\Phi_i(\tau_i)$ for both composites, and the lines show of their wear resistance diagrams $\Phi(\tau)$ ~τ as indicators of absolute and relative wear resistance of these materials in the studied range of specific friction forces.

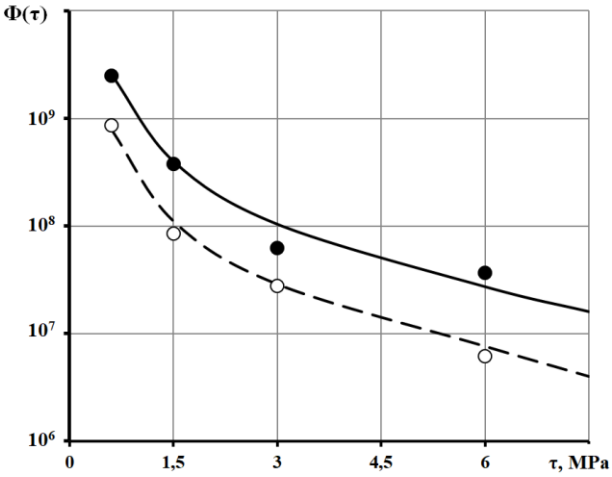


Fig. 3. Wear resistance diagram of polyamide composites: solid line – carbon-filled composite, dashed line – glass-filled composite; experimental values of wear resistance function: carbon-filled composite – dark circles, glass-filled composite – light circles.

The constructed wear resistance diagram of materials makes it possible to visually assess the material wear resistance, reduced to a unit of friction path (3), over the entire range of changes in specific friction forces. It can also be used by extrapolation to establish the expected material wear resistance in the range of greater specific friction forces than they occur in triboexperimental studies, while maintaining the same wear mechanism. This is very useful, because in triboexperimental studies it is sometimes difficult, if not impossible, to provide higher values of specific friction forces that actually occur in tribotechnical sliding systems. In addition, when studying the wear of several materials, as in this case there were two, it is easy to compare their wear resistance at different values of specific friction forces.

Based on the obtained results of triboexperimental studies of these composites on their wear resistance diagrams, their comparative wear resistance was evaluated and it was found that the carbon-filled composite will be 3.6 times more wear-resistant than the glass-filled one.

If we take into account that wear resistance is an inverse characteristic of wear rate, then, accordingly, the experimental inverse function of linear wear resistance $1/\Phi_i(\tau_i)$ will be the experimental function of linear wear rate $I_{hi}(\tau_i)$ at discrete values of specific friction force τ_i.

$$\frac{1}{\Phi_i(\tau_i)} = \frac{h_i}{L_i} = I_{hi}(\tau_i)\cdot \tag{18}$$

Also wear resistance function $\Phi_i(\tau_i)$ is related to the linear wear rate $\gamma_{hi}(\tau_i)$ by the following relation:

$$\Phi_i(\tau_i) = v/\gamma_{hi}(\tau_i), \tag{19}$$

inverse dependence

$$\gamma_{hi}(\tau_i) = v/\Phi_i(\tau_i) = vI_{hi}(\tau_i), \tag{20}$$

which allows to compare the results of experimental evaluation of material wear resistance according to the developed method with the results available in the literature under similar wear conditions, first of all at the same specific friction forces and sliding speeds.

6. EXAMPLE OF WEAR CONTACT PROBLEM SOLUTION, RESULTS

Metal-polymer bearings with the following materials of elements are investigated:
Shaft - steel 0,45%C normalized, grinding; modulus of elasticity and Poisson's ratio E_2= 210 GPa, μ_2 = 0.3; B_2= 10^{13}, m_2 = 2, τ_{20}= 0.1 MPa.

Bushing:
1) composite: glass-filled polyamide PA6 + 30GF, E_{GF} = 3.90 GPa, μ_{GF} = 0.42, volume content of filler - 30%.
2) composite: carbon-filled polyamide PA6 + 30CF, E_{CF} = 5.20 GPa, μ_{CF} = 0.42, volume content of filler - 30%.

Data for calculation: N_{max} =1500 N, N_s = 1000 N, N_{min} = 500 N; $F=N/l$; D_2 = 20, 25, 30 mm; $l=D_2$; at D_2 = 20 mm - F_{max}=75 N, F_s = 50 N, F_{min} = 25 N; at

D_2 = 25 mm - F_{max} =60 N/mm, F_s = 40 N/mm, F_{min} = 20 N/mm; at D_2 = 30 mm - F_{max}=50 N/mm, F_s = 33.3 N/mm, F_{min} = 16.67 N/mm; ε=0.2, 0.3 mm; n_1 = 60 rpm; f_{GF} = f_{CF} = 0.3 - dry friction; h_{1*} = 1.0 mm – allowable wear of the bushing.

The results of the calculations are presented in Tables C1, C2, B1, B2 and Figs. 4 - 9.

Table C1. Maximum tribocontact pressures (glass-filled composite).

h_1, mm	ε, mm	D_2, mm		
		20	25	30
0	0.2	15.0/12.2/8.6	10.7/8.7/6.2	8.1/6.6/4.7
0.5	0.2	11.6/10.4/7.6	8.6/7.4/5.5	6.7/5.7/4.2
1.0	0.2	5.6/6.05/5.6	4.8/4.8/4.2	4.1/3.95/3.35
0	0.3	18.3/14.9/10.5	13.1/10.7/7.5	9.95/8.1/5.7
0.5	0.3	16.2/13.6/9.9	11.8/9.8/7.1	9.1/7.55/5.45
1.0	0.3	13.4/11.8/9.0	10.1/8.7/6.6	7.9/6.76/5.1

Note: p_{0h}=15.0/12.2/8.6 (MPa) at N_{max}=1500/1000/500 (N)

In particular, for the bearing with a glass-filled polymer bushing table C1 shows the tribocontact pressures during wear of the bushing, and in table C2 - its calculated durability at the data selected for calculation. The steel shaft journal wears about three orders of magnitude less.

Table C2. Calculated bearing durability (glass-filled composite).

D_2, mm	ε, mm	N, N		
		500	1000	1500
30	0.3	252.3/124.1	126.2/61.0	83.8/39.8
	0.2	377.3/186.6	188.8/92.4	125.6/60.7
25	0.3	176.4/86.1	88.0/41.9	58.0/27.0
	0.2	263.5/129.7	131.8/63.9	87.4/41.7
20	0.3	113.8/54.9	56.2/26.0	36.7/16.3
	0.2	170.2/83.0	84.8/40.3	55.8/25.9

Note: t = 252.3/124.1 (hours) at h_1=1.0/0.5 (mm)

More detailed results for the bearing with the glass-polymer bushing are shown graphically in Figs. 4 - 6.

Accordingly, Fig. 4a, b shows the influence of load N, radial clearance ε and shaft diameter D_2 on the maximum initial contact pressures p_0. As the diameter of the shaft journal increases, a nonlinear decrease in the value of p_0 is observed (Fig. 4a). In the studied range of loads there is a linear increase of p_0 with different intensity (Fig. 4b).

Figure 5 shows the change in the tribocontact pressures p_{0h} during wear of the bushing.

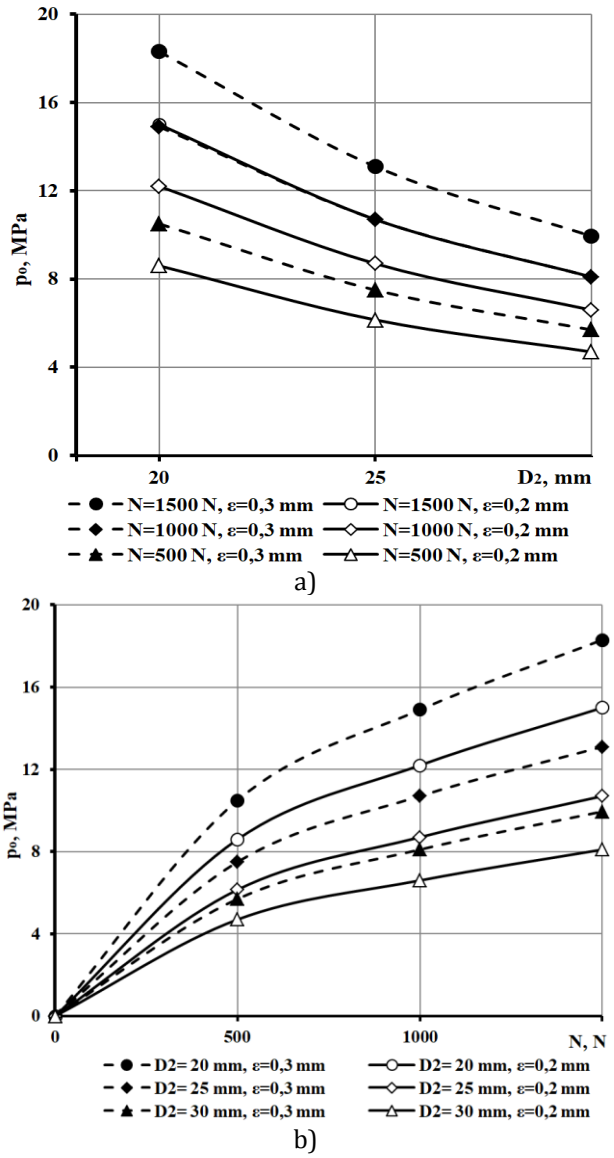


Fig. 4. Regularities of change in the initial maximum contact pressures from various factors of influence.

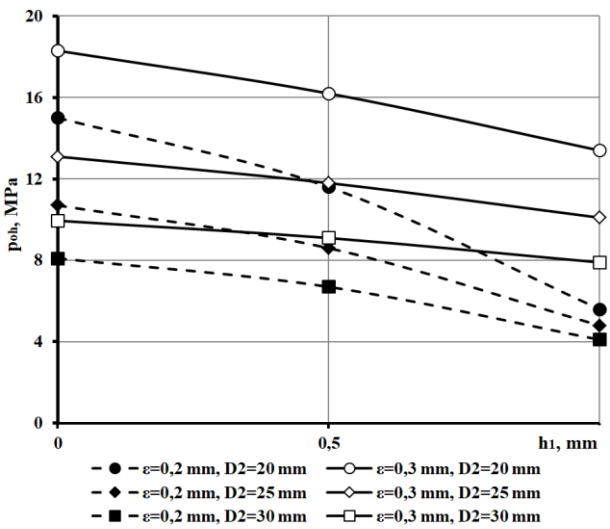
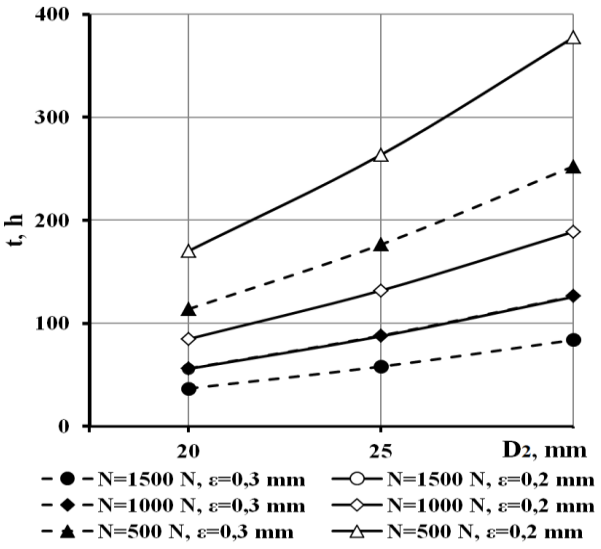


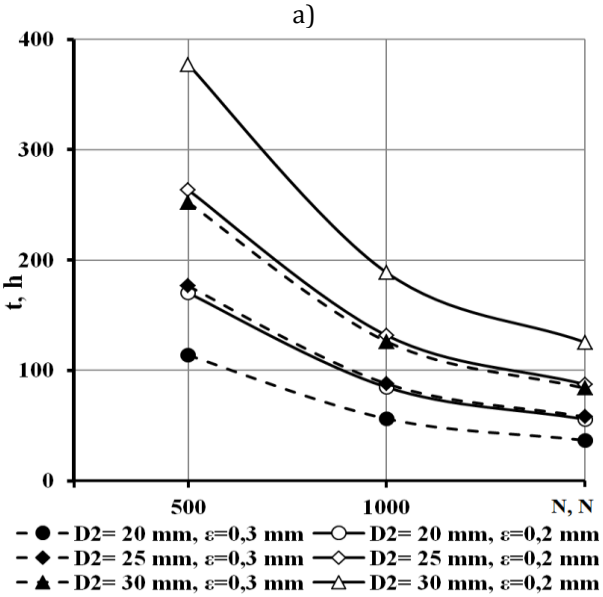
Fig. 5. Regularities of change in the maximum tribocontact pressures during wear at N_{max} = 1500 N.

At a radial clearance ε= 0.3 mm, the dependence of the initial maximum contact pressures p_0 on the wear of the bushing is almost linear. At a smaller radial clearance ε = 0.2 mm, the wear of the bushing causes a more significant nonlinear decrease in p_0, especially at h_1> 0.5 mm.

Regularities of change of the bearing durability t from various factors of influence are given in fig. 6. With increasing shaft journal diameter, the increase in the bearing durability with different intensities is almost linear (Fig. 6a). With a proportional increase in load, the durability of the bearing decreases nonlinearly with different intensities (Fig. 6b).



a)



b)

Fig. 6. Durability of the bearing.

Table B1 shows the tribocontact pressures during wear of the carbon polymer bushing of the bearing, and table B2 shows the calculated durability of such a bearing.

Figures 7a and 7b shows the effect of load N, radial clearance ε and shaft diameter D_2 on the maximum initial contact pressures p_0 for a bearing with a carbon-filled polymer bushing.

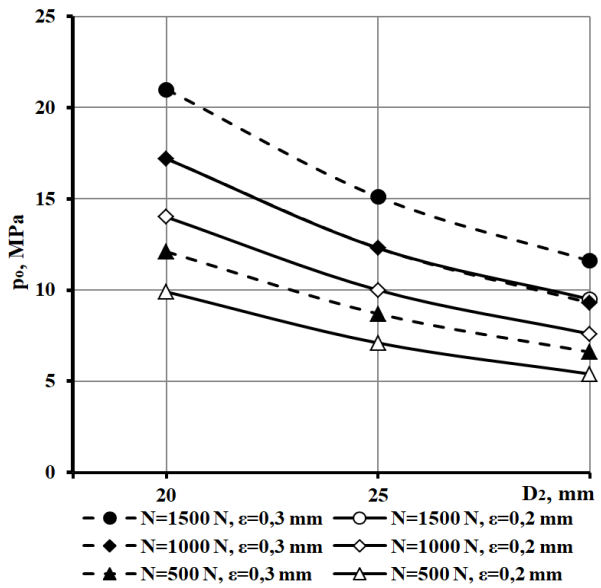
Table. B1. Maximum tribocontact pressures (carbon-filled composite).

h_1, mm	ε, mm	D_2, mm		
		20	25	30
0	0.2	17.2/14.25/9.9	12.3/10.0/7.1	9.5/7.6/5.4
0.5	0.2	14.1/12.1/8.9	10.4/8.75/6.45	8.1/6.75/4.95
1.0	0.2	8.3/8.1/6.95	6.6/6.25/5.2	5.4/5.0/4.1
0	0.3	21.0/17.4/12.1	15.02/12.3/8.7	11.6/9.3/6.6
0.5	0.3	18.5/16.1/11.5	13.85/11.5/8.3	10.8/8.8/6.3
1.0	0.3	15.4/14.3/10.6	12.15/10.4/7.7	9.55/8.0/5.9

Table. B2. Calculated bearing durability (carbon-filled composite).

D_2, mm	ε, mm	N, N		
		500	1000	1500
30	0.3	697.5/347.0	329.6/162.7	214.8/102.8
	0.2	1038/517.0	489.5/242.7	319.0/157.4
25	0.3	490.6/243.3	232.0/114.0	155.5/75.7
	0.2	728.6/362.3	344.4/170.1	231.0/113.4
20	0.3	320/157.8	218.0/71.2	97.6/47.6
	0.2	474.3/235.0	152.6/106.8	145.0/73.0

Slightly higher contact pressures occur in a bearing with a carbon-filled polymer bushing than in a bearing with a glass-filled polymer bushing. Instead, the qualitative patterns of change of the initial maximum contact pressures from different influencing factors are similar to those shown in Figs. 4a and 4b.



a)

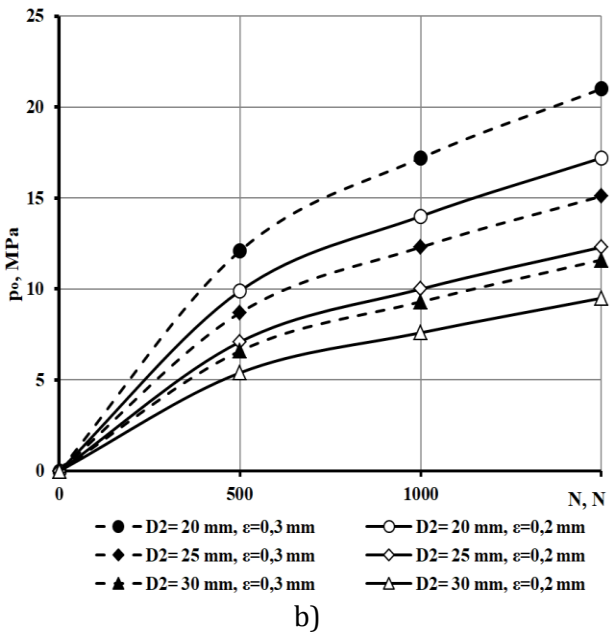


Fig. 7. Regularities of change in the initial maximum contact pressures from various factors of influence.

Figure 8 shows the change in the tribocontact pressures p_{0h} during wear of the carbon-filled composite bushing.

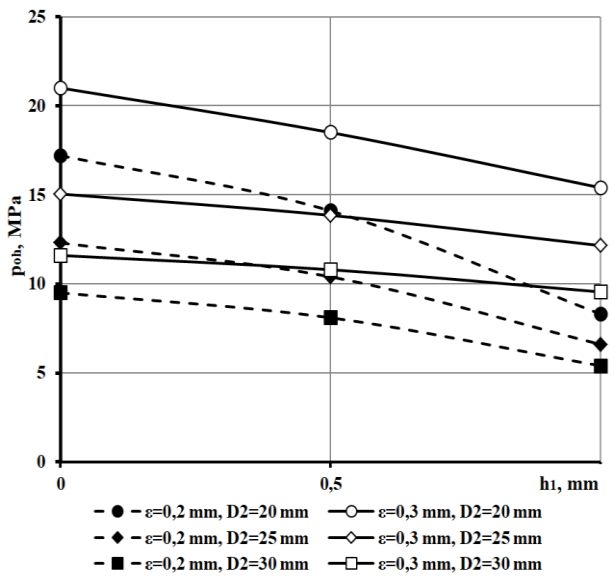


Fig. 8. Regularities of change in the maximum tribocontact pressures during wear at N_{max} = 1500 N.

In Fig. 8 qualitative patterns of change of the maximum tribocontact pressures from various factors of influence are similar to those shown in Fig. 5, and in terms of values they are slightly larger, as in the case of a bearing with a glass-filled polymer bushing.

The influence of various investigated parameters on the durability of the bearing is given in Fig. 9.

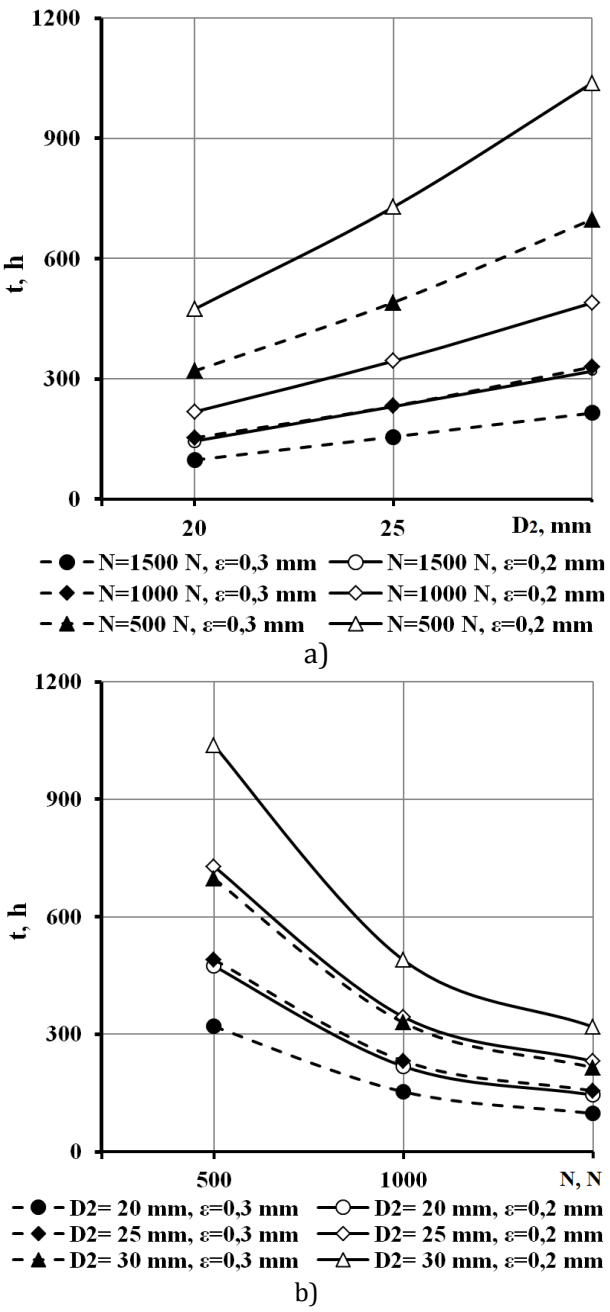


Fig. 9. Durability of the bearing.

The calculated durability of this bearing is 2.54…2.78 times greater than the durability of the bearing with a glass-filled polymer bushing. This is due to the fact that the wear resistance of these materials according to the triboexperimental studies differs by 3.6 times. Qualitative patterns of change in durability are the same as in Fig. 6.

7. VERIFICATION OF RESULTS

The calculation of the metal-polymer bearing presented in [17] was performed by the

computational method. The following data were used for calculation: N = 100 N, D_2 = 19 mm, ε=0.25 mm, l = 10 mm, F = 10 N/mm, E = 0.97 GPa (at 20 ^{0}C), μ = 0.35. The material of the bushing - unstrengthened polyamide PA. According to the calculation results p_0 = 3.48 MPa, and according to [17] - 2.8 MPa. Thus, the discrepancy between the results of the exact solution according to the presented method and the numerical method is 19.5 %.

8. CONCLUSION

1. The developed methodology for calculating hybrid (metal-polymer) sliding bearings provides at the design stage the possibility of predictive assessment of load capacity, durability and radial wear of polyamide bushing, dispersed - reinforced with glass or carbon fibers.
2. According to the model triboexperimental researches carried out according to the developed technique, the characteristics of wear resistance of the investigated polyamide composites in pair with steel in the conditions of dry friction are determined.
3. As a result of numerical solution, the maximum contact pressures and their transformation at a given wear of the bushing are determined. Quantitative and qualitative patterns of their change with the change of the main studied factors of influence are established.
4. In particular, when the load is increased 3 times, the maximum contact pressures increase ≈ $\sqrt{3}$ times regardless of the radial clearance in the bearing and the bushing material. When the journal diameter is increased 1.5 times, regardless of the radial clearance in the bearing, the maximum contact pressures are reduced by ≈1.5 $\sqrt{1.5}$ times. When the radial clearance is increased 1.5 times, the maximum contact pressures increase approximately 1.5 times.
5. It is established that the maximum contact pressures in bearings with a carbon-filled composite bushing will be greater than in bearings with a glass-filled composite bushing ≈ $\sqrt{E_{CF} / E_{GF}}$ times.
6. The forecast calculation of bearings durability at the set maximum allowable wear taking into account change of the bushing diameter, radial clearance and radial loading is realized. Quantitative and qualitative patterns of its change depending on the change of the specified factors of influence are established.
7. In particular, when the load is increased 3 times, the durability of bearings with a glass-filled polymer bushing, regardless of the radial clearance is reduced almost 3 times, and in a bearing with a carbon-filled polymer bushing - about 3.08 times. When the journal diameter is reduced 1.5 times, the durability of the bearings decreases ≈2.25 times, i.e. 1.5^2 times. When the radial clearance increases 1.5 times, the durability of the bearings decreases in direct proportion, i.e. ≈1.5 times.
8. The calculated durability of bearings with a carbon-filled polymer bushing is on average $2E_{CF}/E_{GF}$ times greater than the durability of bearings with a glass-filled polymer bushing.
9. The given results of researches confirm efficiency of use at designing of metal-polymer sliding bearings of the presented scientifically sound calculation methodology of their loading capacity, durability or wear.

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