

On the Question of Methodology of Hybrid Sliding Bearings Estimated Load Capacity and Durability Evaluation

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ABSTRACT

The computational method of hybrid (metal-polymer) sliding bearings according to the load capacity and durability of the bushing made of polyamide filled with glass or carbon dispersed fibers was presented. The initial contact pressures were calculated and the influence of bushing wear on their relative reduction was investigated. The wear resistance of bearings with bushings made of both polymer composites was also evaluated. Regularities of their change from loading, diameter of the bushing and radial clearance were established.

Keywords: metal-polymer sliding bearings, polyamide reinforced composite, computational method, maximum contact pressures, durability

INTRODUCTION

In various branches of the technosphere (mechanical engineering, aircraft construction, instrument making, automobile construction, food processing, pharmaceutical, textile, pulp and paper, chemical industry, etc.), as well as in other various areas of human activity, sliding bearings with a composite bushing (micro, miniature, small, medium and large diameters) are used increasingly often. This is due to the fact that there are many applications where they have advantages over rolling bearings. They can work well in dry friction without the need for lubrication and under the conditions that make it impossible to use and replace lubricants due to the technological requirements or special operating conditions. They also have increased damping capacity and other positive qualities.

The polymer composite materials used for the manufacturing of sliding bearing bushings have the strength characteristics and modulus

of elasticity 1–2 orders of magnitude lower than typical metallic materials, which fundamentally changes the conditions of contact and tribocontact interaction. The most common are polyamides, polycarbonates, polyimides, polyacetals, polyamidoimides, polytetrafluoroethylene, polyesterketones and others with different fillers that increase their wear resistance. More and more new types of such materials are appearing. However, their choice in engineering practice is practically not based on the scientifically sound methods of design calculation.

The simplified or approximate calculation [3–7, 14–16] methods, including and numerical research methods for metal [11–13] and metal-polymer [8–10] sliding bearings known in the literature have not yet found practical application. This is obviously because they use the well-known Archard law of abrasive wear, which assumes a linear dependence of wear on the contact pressure and the path of sliding friction. However, this type of wear by the micro-cutting

mechanism is practically not allowed even in dry friction bearings. Therefore, estimates are very approximate and with sufficient complexity of their implementation, they are unpromising in the engineering practice.

Therefore, the creation of sound, efficient and fairly simple methods for estimating the load capacity, wear and durability of metal-polymer bearings at the design stage remains an urgent task. This publication is devoted to solving this current problem. For this purpose, the methodology of studying the wear kinetics during sliding friction was used as a process of frictional-fatigue destruction of the near-surface layers of tribosystem elements under the influence of specific friction forces [1]. The methods for calculating the wear of metal contact elements of sliding bearings, including bearings with a small roundness of the shaft and bushing contours, developed on its basis were also employed [1, 2].

METHODS

The calculation of such a sliding tribosystem is performed as a plane contact problem of the theory of elasticity about the internal contact of cylindrical bodies of close radii. The design scheme of the bearing is shown in Figure 1.

The bearing journal is subjected to a radial load N , which in the plane problem is assigned to the length l of the journal. That is, a reduced

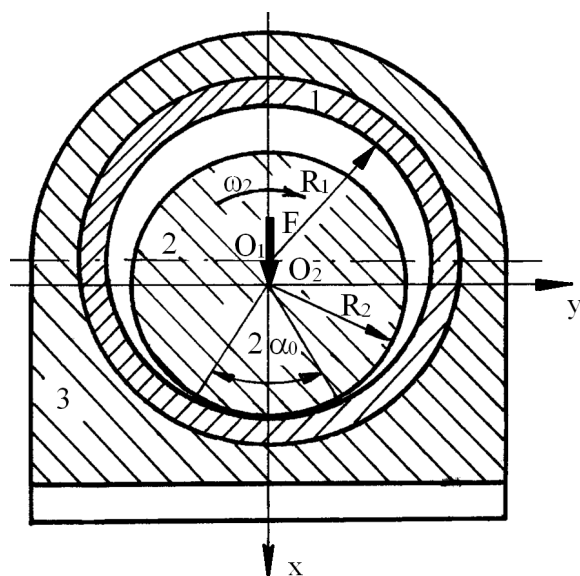


Fig. 1. Scheme of the sliding bearing

concentrated load $F=N/l$ is applied to the center O_2 of the nominally circular disk. There is a radial clearance $\varepsilon=R_1-R_2 \geq 0 \ll R$ between it and the ring 1 (bearing bushing). The shaft is made of steel and the bushing is made of the polyamide composite reinforced with dispersion glass or carbon fibers. Therefore, their wear resistance and elastic properties will be significantly different. Disc 2 rotates at a constant angular velocity ω_2 . Under the loading of disk 2 by the force F , an initial contact pressure p_α is arising in the contact area $2R_2\alpha_0$. The distribution of the pressure p_α and the contact angle $2\alpha_0$ are unknown and should be determined as a result of the problem solution. Under wear, both the pressure value and the angle of contact will change, which is also subject to study.

The calculation of the initial static contact pressures p_α is performed by means of the singular integro-differential equation [1]:

$$c_1 \int_{-\alpha_0}^{\alpha_0} \text{ctg} \frac{\alpha-\theta}{2} p'_\theta d\theta = c_2 p_\alpha + c_3 \int_{-\alpha_0}^{\alpha_0} p_\alpha d\alpha + c_4 \cos \alpha \int_{-\alpha_0}^{\alpha_0} p_\alpha \cos \alpha d\alpha + \frac{\varepsilon}{R^2} \quad (1)$$

where: $p'_\theta = dp/d\theta$;

α is the polar angle;

$0 \leq \alpha \leq \theta$;

$$c_1 = \frac{1}{8\pi} \left(\frac{1+\kappa_1}{G_1 R_1} + \frac{1+\kappa_2}{G_2 R_2} \right);$$

$$c_2 = \frac{1}{4} \left(\frac{1-\kappa_1}{G_1 R_1} - \frac{1-\kappa_2}{G_2 R_2} \right);$$

$$c_3 = \frac{1+\kappa_1}{8\pi G_1 R_1}$$

$$c_4 = \frac{1}{2\pi} \left(\frac{\kappa_1}{G_1 R_1} + \frac{1}{G_2 R_2} \right); \quad R_1 \approx R_2 = R; \quad G_1, G_2$$

are the shear modulus of materials;

ν_1, ν_2 are the Poisson coefficients;

$\kappa=3-4\nu$ is the Muskhelishvili constant.

The approximate solution of Eq. (1) for determining p_α is performed by the collocation method for one collocation point $\alpha=0,5\alpha_0$. The contact pressure function is determined as [1]

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

where: $E_0 = (e_4 / R_2) \cos^2(\alpha_0 / 4)$

$$e_4 = 4E_1 E_2 / Z$$

$$Z = (1 + \kappa_1)(1 + \nu_1)E_2 + (1 + \kappa_2)(1 + \nu_2)E_1$$

$$E = 2G / (1 + \nu).$$

The maximum contact pressures p_0 occur at $\alpha = 0$. Then

$$p_0 \approx E_0 \varepsilon \tan(\alpha_0 / 2) \quad (3)$$

The unknown contact angle α_0 is determined by the equilibrium of the forces applied to the disk

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

The tribocontact pressures acting in the bearing during the elements wear are determined as follows:

$$p_{oh} = p_\alpha + p_h, \quad (5)$$

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

In order to find them, the following dependence is used

$$p_h = E_h \varepsilon_h \sqrt{\tan^2 \frac{\alpha_{0h}}{2} - \tan^2 \frac{\alpha}{2}} \quad (6)$$

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

In order to determine the half tribocontact angle α_{0h} characterizing the contact zone during wear, a condition is used

$$F = 4\pi R_2 E_0 (\varepsilon + c_{oh} \varepsilon_h) \sin^2(\alpha_{0h} / 4) \quad (7)$$

where: $\varepsilon_h = h_{k \max} (-K_t^{(k)} + h_k')$;

$h_1' = h_2 / h_1$, $h_2' = h_1 / h_2$ are relative wears in the tribosystem;

$K_t^{(1)}, K_t^{(2)}$ are the mutual overlapping factors bearing elements in the moving contact;

c_{oh} is the rate in the tribocontact angle increase.

$$h_1' = \frac{h_2}{h_1} = \frac{\Phi_1(\tau)}{\Phi_2(\tau)} = \frac{B_1 \tau_{10}^{m_1} (\tau - \tau_{20})^{m_2}}{B_2 \tau_{20}^{m_2} (\tau - \tau_{10})^{m_1}} K_t^{(2)}, \quad (8)$$

$$h_2' = \frac{h_1}{h_2} = \frac{\Phi_2(\tau)}{\Phi_1(\tau)} = \frac{B_2 \tau_{20}^{m_2} (\tau - \tau_{10})^{m_1}}{B_1 \tau_{10}^{m_1} (\tau - \tau_{20})^{m_2}} K_t^{(1)}; \tau = fp_0$$

B, m, τ_0 are wear resistance characteristics of materials set by triboexperimental researches, $k=1$ for the ring and 2 for the disc, respectively.

The calculated ratio for the predictive estimate of the bearing durability at the given bushing wear $h_1 = h_{k \max}$ is [1]

$$t = \frac{-B_k \tau_{k0}^{m_k}}{\nu c_h \tau_h \Sigma_k (1 - m_k) K_t^{(k)}} \left\{ [\tau - \tau_{k0}]^{1 - m_k} - [(\tau - \tau_{k0}) + c_h h_{k \max} \Sigma_k \tau_h]^{1 - m_k} \right\}. \quad (9)$$

where: $\Sigma_1 = (-K_t^{(1)} + h_1')$, $\Sigma_2 = (K_t^{(2)} - h_2')$;

$$K_t^{(1)} = 1, K_t^{(2)} = \alpha_{0s} / \pi.$$

SOLUTIONS

The data for calculation: $N_{\max} = 1500$ N, $N_s = 1000$ N, $N_{\min} = 500$ N; $F = N/l$; $D_2 = 20, 25, 50$ 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, $F_s = 40$ N, $F_{\min} = 20$ N; at $D_2 = 30$ mm – $F_{\max} = 50$ N, $F_s = 33.3$ N, $F_{\min} = 16.67$ N; $\varepsilon = 0.2, 0.3$ mm; $n_1 = 60$ rpm; $f_{GF} = f_{CF} = 0.3$ – dry friction; $h_{1*} = 1.0$ mm – allowable wear of the bushing.

The metal-polymer bearings with the following materials of elements were investigated:

1. Shaft – steel 0.45% C normalized, grinding; modulus of elasticity and Poisson's ratio $E_2 = 210$ GPa, $\mu_2 = 0.3$; $B_2 = 10^{13}$, $m_2 = 2$, $\tau_{20} = 0.1$ MPa.

2. Bushing:

- composite: glass-filled polyamide PA6 + 30GF, $E_{GF} = 3.90$ GPa, $\mu_{GF} = 0.42$, volume content of filler – 30%; $B_{1GF} = 6.67 \cdot 10^{10}$, $m_{1GF} = 1.9$, $\tau_{10} = 0.05$ MPa;
- composite: carbon-filled polyamide PA6 + 30CF, $E_{CF} = 5.20$ GPa, $\mu_{CF} = 0.42$, volume content of filler – 30%; $B_{1CF} = 24 \cdot 10^{10}$, $m_{1CF} = 1.9$, $\tau_{10} = 0.05$ MPa.

The results of the calculations are presented in Tables 1, 2 and Figures 3–6.

Bearing with glass-polymer bushing

Table 1 presents the results of the estimated evaluation of the tribocontact pressures during wear of the bushing. The steel shaft journal wears less by about three orders of magnitude.

The relative reduction of tribocontact pressures p_{oh}/p_0 during bushing wear in this bearing with a change in load N , radial clearance ε and shaft diameter D_2 is shown in Figures 2 and 3.

The results presented in the table and figures show that the reduction of the load leads to a

Table 1. Maximum tribocontact pressures in the bearing with glass-polymer bushing

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)

decrease in the value of the initial contact pressures p_0 (Table 1), and, accordingly, to a decrease in the change of tribocontact pressures p_{0h} , ie the ratio p_{0h}/p_0 (Fig. 2). This trend is manifested in both smaller and larger radial clearances in the bearing.

Similar quantitative and qualitative trends occur with increasing diameter of the shaft journal (Fig. 3).

Bearing with carbon polymer bushing

Table 2 presents the results of the estimated evaluation of the tribocontact pressures during wear of the bushing.

The relative decrease in the tribocontact pressures p_{0h}/p_0 during wear of the bushing in this bearing when changing the load N , the radial clearance ϵ and the diameter D_2 of the shaft is shown in Figures 4 and 5.

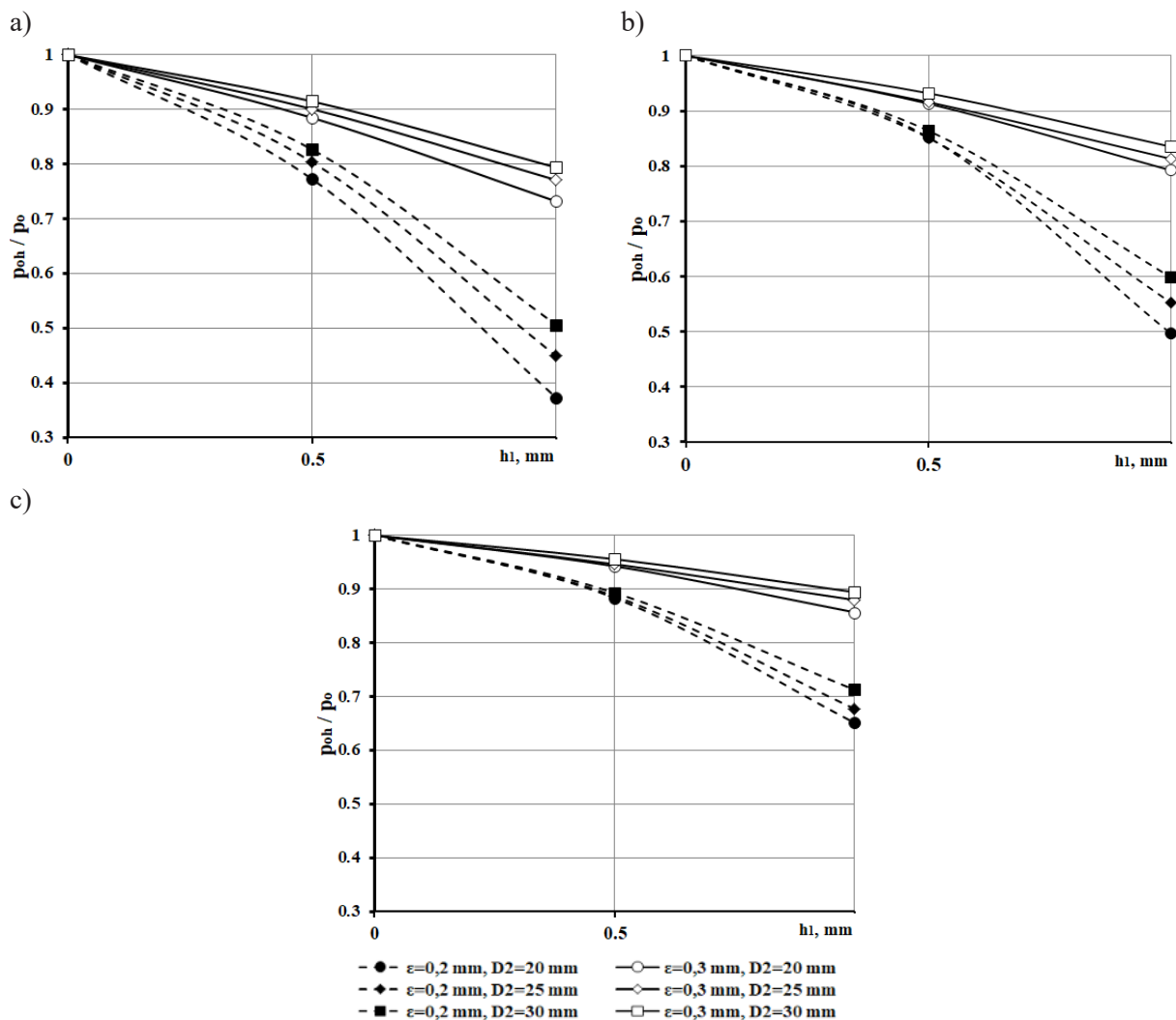


Fig. 2. The influence of load on the relative reduction of tribocontact pressures: a) 1500 N; b) 1000 N; c) 500 N

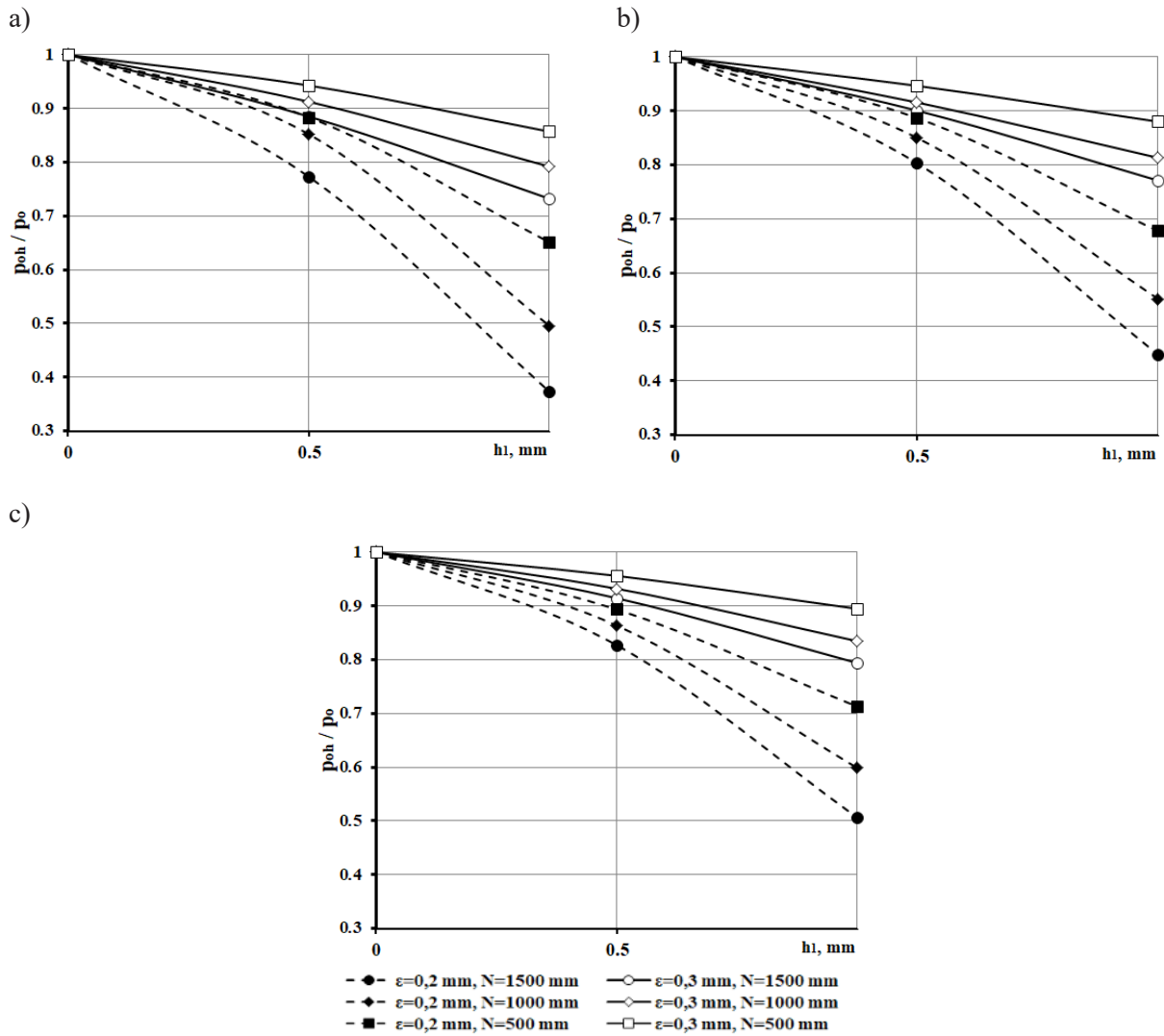


Fig. 3. The influence of the journal diameter on the relative reduction of tribocontact

The qualitative trends in the influence of the studied parameters on the change of initial contact pressures and tribocontact pressures are similar to those given in Figures 2 and 3. The comparison of the results in Table 1 and 2 shows that in bearings with a carbon-polymer bushing, the initial contact pressures are on average higher in $\sqrt{E_{CF} / E_{GF}}$ times than with a glass-polymer

bushing. A similar ratio will also be in the case of tribocontact pressures.

The influence of the above-mentioned factors on the durability of hybrid bearings has also been studied. The results of the calculated evaluation of the bearing durability t with the accepted allowable wear h_{1*} of the bushing are presented in Figure 6.

Table 2. Maximum tribocontact pressures in the bearing with carbon polymer bushing

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

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

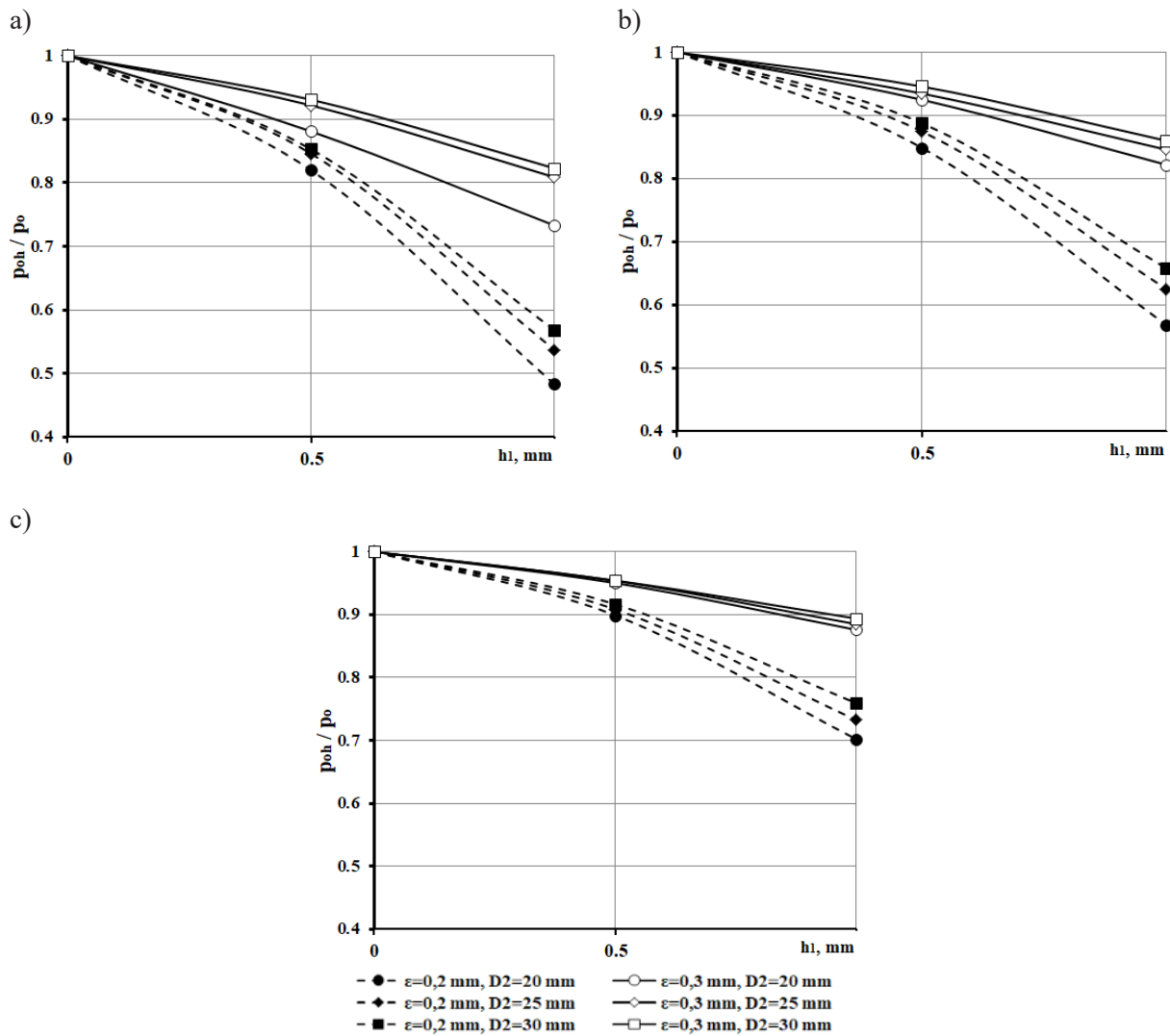


Fig. 4. The influence of load on the relative reduction of tribocontact pressures: a) 1500 N; b) 1000 N; c) 500 N

The calculated durability of the bearing with a carbon-polymer bushing is 2.54...2.78 times greater than the durability of the bearing with a glass-polymer bushing, i.e. on average 2.66 times. This corresponds to the ratio of the Young's modules of materials $2E_{CF}/E_{GF} = 2.67$ times.

CONCLUSIONS

The presented computational method of hybrid (metal-polymer) sliding bearings provides an effective and reasonable evaluation of their load capacity and durability at the design stage. The maximum contact pressures and their quantitative reduction at wear of the bushing, and also qualitative patterns of their relative change with change of the basic investigated factors of influence were established.

Increasing the load 3 times leads to an increase ≈ 1.73 times in the maximum contact pressures, and increasing the diameter 1.5 times reduces them by about 1.84 times. Increasing the radial clearance causes them to grow in direct proportion. The maximum contact pressures in the bearings with a carbon-filled composite bushing will be greater than in the bearings with a glass-filled composite bushing $\approx \sqrt{E_{CF}/E_{GF}}$ times. The quantitative and qualitative patterns of the changes in the durability of bearings with allowable wear, taking into account the diameter of the bushing, radial clearance and radial load were established.

Increasing the load 3 times leads to the same reduction in the durability of bearings with a glass-polymer bushing, and 3.08 times for a bearing with a carbon-polymer bushing. The diameter increase 1.5 times increases the durability of

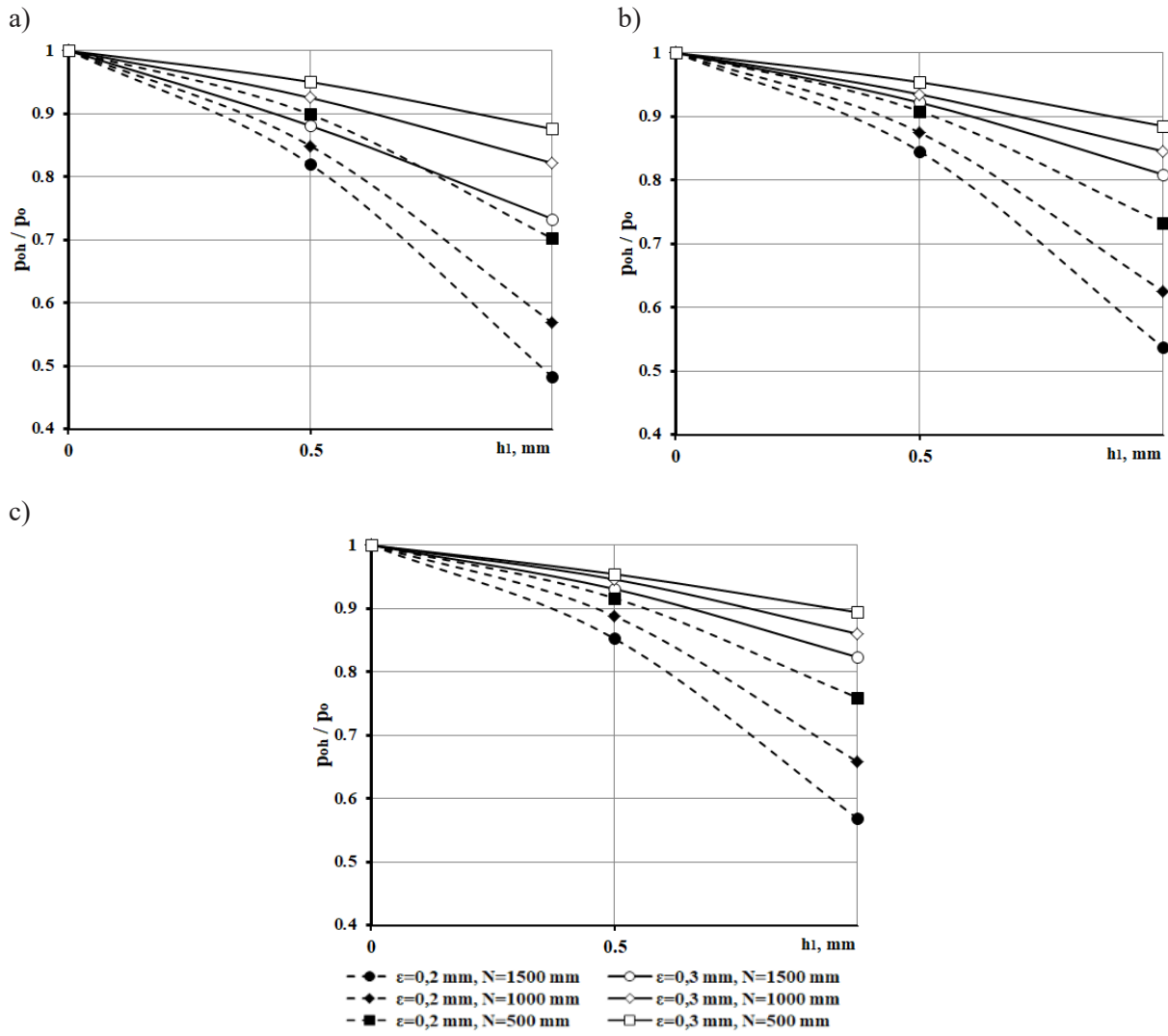


Fig. 5. The influence of the journal diameter on the relative reduction of tribocontact pressures: a) 20 mm; b) 25 mm; c) 30 mm

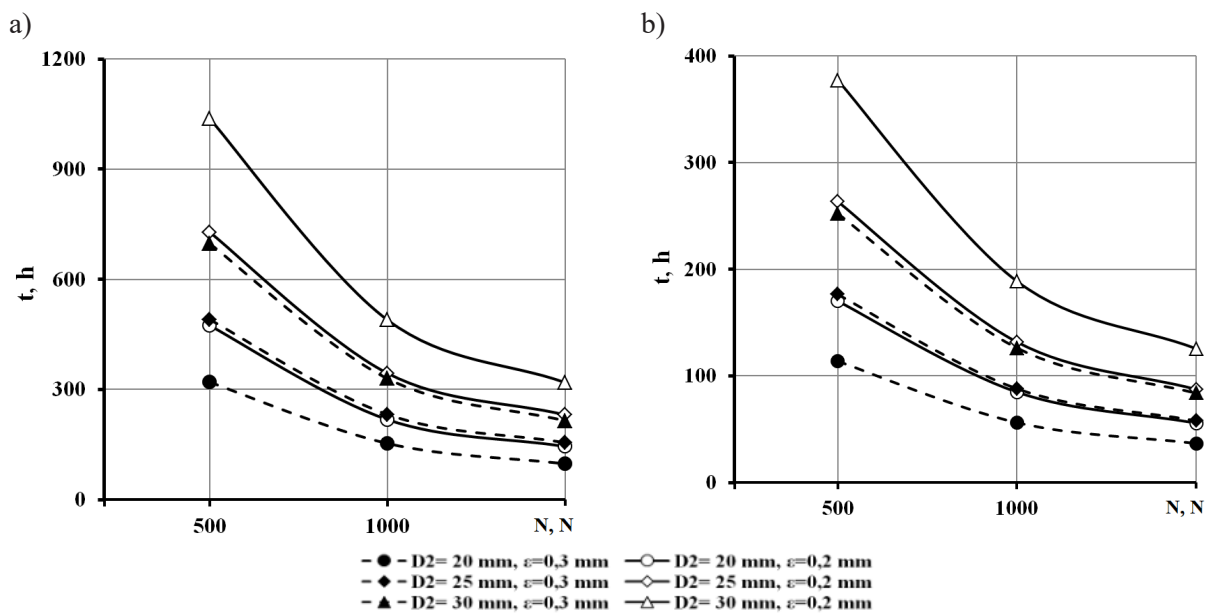


Fig. 6. Durability of hybrid bearings: a) bushing – carbon-filled polyamide PA6 + 30CF
b) bushing – glass-filled polyamide PA6 + 30GF

bearings ≈ 2.25 times. Increasing the radial clearance by 1.5 times reduces the durability of bearings in direct proportion. The calculated durability of bearings with a carbon polymer bushing will be on average $2E_{CF}/E_{GF}$ times greater than the durability of bearings with a glass polymer bushing.

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