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RESEARCH

Investigations on Contact Pressures and Durability of Metal-Polymer Dry Sliding Bearings with Miniature, Small and Large Diameters

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A B S T R A C T

Shaft diameters of sliding bearings, including metal-polymer, can be much differentiated. The carrier capacity and durability of metal-polymer (MP) sliding bearings of miniature, small, medium and large diameters were investigated according to author's calculation method. Such main types of thermoplastic polymeric materials are used for bushings: PA polyamide and its composites, PTFE polytetrafluoroethylene composites, PEEK polyetheretherketone and its composites, PET polyethylene terephtalate and its composites. The main thesis of author's calculation methods for studying the contact strength of polymer bushing, wear and durability of MP bearings, the method of model triboexperimental studies of metal-polymer pairs under dry sliding friction are presented. The research results are presented in the form of wear resistance diagrams and the wear resistance characteristics of polymer materials established by them. Quantitative and qualitative results of the MP bearings shaft diameters influence on the maximum contact pressures and durability are presented. It was established that for the range of shaft diameters 2 ... 20 mm there is a nonlinear dependence of the decrease in maximum contact pressures. For larger diameters, this dependence will be close to linear. Similar patterns in the durability of bearings with different polymer bushing materials are observed. Their durability depends significantly on the type of polymer bushing.

1. INTRODUCTION

Different types of metal-polymer (MP) sliding bearings are widely used in modern mechanical engineering and other spheres of human activity. Metal-polymer sliding bearings are often used where it is impossible to use metal sliding bearings. In particular, they can work reliably under dry friction where lubrication cannot be used or provided. The range of shaft diameters used is significant: from less than 2 mm to 100 and more.

Different combinations of coupled elements are used for MP sliding bearings. The shaft is made steel (cast iron), and the bushing is of different

groups of polymeric materials, usually reinforced with different types of fillers. In particular, the following polymers were studied: filled polyamide PA6 + 30GF (30% fiberglass), PA6 + 30CF (carbon fiber), PA6 + MoS_2 (molybdenum disulfide); polytetrafluoroethylene PTFE + 15CF (carbon fiber), PTFE + 10C + 10CF (graphite + carbon fiber), PTFE + 15C + 5MoS_2 (graphite + molybdenum disulfide); highly efficient polymers - polyetheretherketone PEEK, PEEK + 30CF (carbon fiber) and polyethylene terephtalate ethylene PET, PET + 20PTFE. There are known other new types of composite materials that can be used for MP bearings.

It should be noted that the selection of optimal types of polymers for MP bearings in accordance with their operation conditions practically is not based on the scientifically sound principles and results of thorough tribological research. The results of such studies of polymeric materials are known in the literature. However, they can be used to a limited extent when designing MP sliding bearings. The vast majority of them are obtained by determining their wear resistance according to ISO 7148-2 under one contact pressure value [1]. Such data on the material wear resistance of the material is impossible to extrapolate to other operating conditions of the bearings.

Various results are available in the literature on experimental studies to the scheme pin-on-disk under dry friction, concerning tribological behavior of different polymeric materials. In particular the wear of polyamide PA6-steel AISI 02 tribocouple was studied in [2]. The most large-scale investigations are given in [3], where the results on twenty-one engineering commercial polymers studies, both unfilled and composite are presented. The friction and wear of PTFE, polyacetal POM-H, PA6 + GF, PA6 + CF on 40CrMnNiMo8 steel with different roughness were studied in [4]. Wear of various polymers (PTFE, PA6, polyacetal POM and others) under dry friction on stainless steel was studied in [5]. Influence of sliding speed and loading on wear of PTFE, PTFE + GF, PTFE + Bronze + C on stainless steel AISI 440C under dry friction was studied in [6]. The friction and wear of the PTFE + 25Bronze composite (bronze powder) were studied in a similar way [7]. Some experimental results on tribological behavior of highly efficient polymers under friction on steel are presented in [8-12]. Wear of REEK under dry friction on steel according to the scheme of end friction is established in [8]. The unfilled PEEK, PEEK + 10CF + 10Graphite, PEEK + 10CF + 10Graphite + 10PTFE at different contact pressures and sliding speeds under dry sliding friction on steel AISI 52100 (100Cr6) were investigated in [9]. Polymer wears according to the scheme of end friction in pair of PEEK-316L stainless steel with different surface roughness was studied in [10]. Accordingly, in [11], the tribological behavior of the cast polyamide PA6 + Oil, PA6 + MoS_2, POM + Aluminium, PET + PTFE, PTFE + Bronze, PTFE + Graphite polymer composites were studied under dry friction according to the ball-on-plate scheme. Friction coefficient, wear loss, friction energy and temperature in MP model sliding bearings with bushing with PEEK, PEEK + 30CF, PEEK + 30GF, PEEK + 10CF + 10C + 10PTFE were studied in [12]. Testing of the tribological behavior of three types of Iglidur polymer materials paired with 100Cr6 steel for automotive sliding bearings was carried out in [13] using the ball-on-disk scheme under dry friction conditions. Archard wear coefficient and friction coefficient were determined.

Until now, effective and scientifically sound methods for the estimated assessment of carrying capacity, wear and durability of MP sliding bearings for their use in design have also not been developed. There is also no appropriate standard for the calculation of such bearings are not known, and known calculation [14-17,23,25] or numerical methods of metal [21,22,24] and metal-polymer [18-20] sliding bearings investigations so far here have not found practical application. They are based on Archard's law of abrasive wear, which assumes a linear dependence of wear on contact pressure and sliding speed. However, this type of wear by the micro-cutting mechanism is not allowed in bearings, even under dry friction. In addition, the mechanical properties, in particular the Young's modulus of the shaft and bushing materials are significantly different up to 250 times (Table 1). It is obvious that these fundamentally affects the contact characteristics and wear MP bearings bushing.

Therefore, the development of adequate to different operating conditions effective methods of estimated evaluation for the above mentioned MP bearings characteristics of remains an urgent task. For studying MP bearings the developed method of their characteristics calculation [26-28] was used below. It is based on the methods of wear calculation of sliding bearings with metal contact

elements [29-36]. The methodology for investigation of wear kinetics under sliding friction as a process of friction-fatigue fracture of materials surface layers of tribosystem elements was used in turn, for their development [29-31].

The investigations of MP bearings diameter influence on the specified service characteristics are absent in the literature in connection with the lack of appropriate calculation methods. This publication is devoted to the solution of this practical problem by the mentioned developed method. This investigation is dedicated to the solution of this practical issue using the developed methods of analytical calculation.

2. ANALYTICAL METHODS OF CALCULATION

The calculation of the initial maximum contact pressures $p(0)$ in MP sliding bearing was carried out by the author's method of contact mechanics about the internal contact of cylindrical bodies with close radii [29-33]. The general view and scheme of MP bearing are shown in Fig. 1.

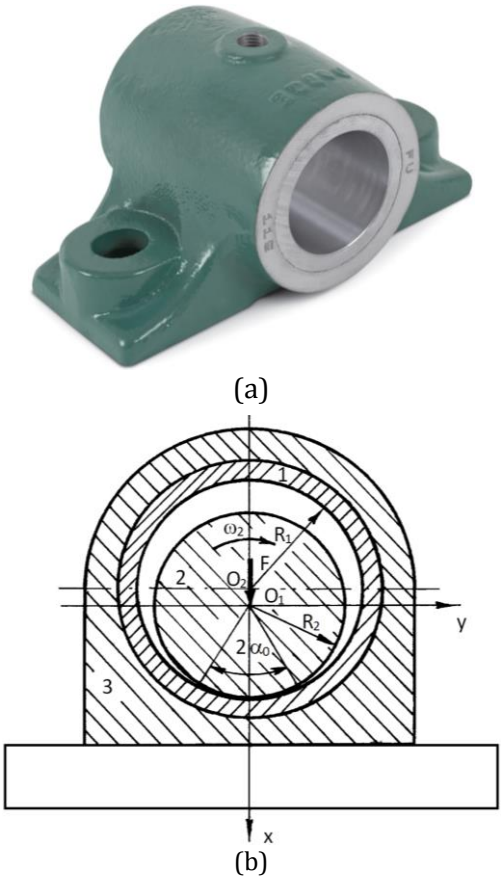


Fig. 1. Metal-polymer sliding bearing: (a) - general view; (b) - scheme (1 - composite bushing, 2 - shaft, 3 - housing).

The shaft 2 of MP bearing is subjected to static load *N*. Contact 3D problem for this tribosystem (Fig. 1a) is reduced to 2D contact problem of elasticity theory (Fig. 1b). Since a plane contact problem of elasticity theory is considered, the total load *N* on the bearing is reduced to a unit width, that is $F = N / l$, where l - length of the shaft journal. The initial radial clearance $\varepsilon = R_1 - R_2 \geq 0$ must be guaranteed to ensure the bearing work. Accordingly, R_1 is the internal radius of the bushing 1, and R_2 - the radius of the shaft 2 (Fig. 1b). The shaft in the MP bearing is metallic and the bushing - polymeric. The strength characteristics of shaft and bushing materials are differing significantly (8 - 10 times), and Young's modulus (60 - 250 times). Polymeric materials have up to 3 orders of magnitude lower than wear resistance of steel. The shaft 2 under the influence of external loading contacts the bushing 1 by arc $2R_2\alpha_0$, where the contact pressure $p(\alpha)$ will occur. Both the contact angle $2\alpha_0$ and pressures distribution $p(\alpha)$ should be determined when solving this problem.

The singular integrodifferential equation is given in [29-31] to solve this problem. Its approximate solution is carried out by collocation method, and the contact pressure function is selected in this form:

$$p(\alpha) \approx E_0 \varepsilon \sqrt{\tan^2 \frac{\alpha_0}{2} - \tan^2 \frac{\alpha}{2}}, \qquad (1)$$

where $E_0 = (e / R_2)\cos^2(\alpha_0 / 4)$ - correlation for collocation coefficient [31], $e = 4E_1E_2 / Z$, $Z = (1+\kappa_1)(1+\nu_1)E_2 + (1+\kappa_2)(1+\nu_2)E_1$; α - polar angle, E_1, E_2 – modulus of elasticity of bushing and shaft materials; ν_1, ν_2 – Poisson's ratio of these materials; $\kappa = 3 - 4\nu$ - Kolosov-Muskhelishvili constant.

Maximum contact pressures appear along the load line that is when $\alpha = 0$. Then from (1)

$$p(0) \approx E_0 \varepsilon \tan(\alpha_0 / 2) \ . \qquad (2)$$

The conditional equilibrium forces applied to the shaft, that is the external load *F* and contact pressures $p(\alpha)$ are used to find the contact semiangle α_0.

$$F = R_2 \int_{-\alpha_0}^{\alpha_0} p(\alpha)\cos\alpha d\alpha . \quad (3)$$

Considering the expression (2) and given correlation for E_0 the formula for contact semiangle determination will have a simple view

$$\alpha_0 = 2arc\sin\sqrt{F/\pi e\varepsilon} . \quad (4)$$

As composite bushing wears, the parameters of the initial contact (pressures $p(0)$ and contact angle $2\alpha_0$) will change when shaft rotates at a constant angular velocity. Accordingly, pressures $p(0)$ will reduce as a result contact area $2R_2\alpha_0$ increase.

That is maximum current (wear contact) pressures are calculated as

$$p(0,h) = p(0) - p(h), \quad (5)$$

where $p(h)$ – change of initial pressures owing to wear

Low of $p(h)$ variation is selected as [31,32]:

$$p(h) = E_h\varepsilon_h \tan(\alpha_{0h}/2), \quad (6)$$

where $E_h = c_h(e/R_2)\cos^2(\alpha_{0h}/4)$, c_h >0 – indicator of wear rate

To determine the variable semiangle α_{0h} due to bushing wear by successive approximations method the following equilibrium condition is used [31,32]:

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

where $\varepsilon_h = h_{1\max}\left(K_t^{(1)} - h_1'\right)$; $h_1' = h_2/h_1$ - comparative shaft wear relatively to bushing; $K_t^{(1)} = 1, K_t^{(2)} = \alpha_0/\pi$ – coefficients of mutual overlap of the bearing elements; $c_{\alpha h}$ - indicator of angle α_{0h} growth rate; $h_{1\max}$ - permissible wear of the bearing bushing 1;

$$h_1' = \frac{B_1\tau_{10}^{m_1}\left(\tau - \tau_{20}\right)^{m_2}}{B_2\tau_{20}^{m_2}\left(\tau - \tau_{10}\right)^{m_1}} K_t^{(2)}, \text{ [27,31,32]} \quad (8)$$

where $\tau = fp(0)$ - Coulomb specific wear force; f – coefficient of sliding friction; B_k, m_k, τ_{k0} – material wear resistance characteristics defined from the results of triboexperimental investigations; k – numbering of bearing elements (Fig. 1b).

Formulas in [27,28] are used to calculate MP bearing durability under permissible wear of the bushing $h_{1\max}$

$$t = \frac{-B_1\tau_{10}^{m_1}}{vc_h\tau(h)\Sigma_1\left(1-m_1\right)K_t^{(1)}} *$$

$$*\left\{\left[\tau - \tau_{10}\right]^{1-m_1} - \left[\left(\tau - \tau_{10}\right) + c_h h_{1\max}\Sigma_1\tau_h(h)\right]^{1-m_1}\right\}, \quad (9)$$

where $v = \omega_2 R$ - sliding speed; $\Sigma_1 = (K_t^{(1)} - h_1')$; $\tau(h) = fp(0,h) = fE_h\tan(\alpha_{0h}/2)$ - maximum specific wear force in tribocontact.

3. MATERIALS AND MODEL TRIBOEXPERIMENT

In the above method of MP bearing durability calculating according to formula (9), the characteristics of material wear resistance B_k, m_k, τ_{k0} in the steel-polymer tribocouple under the adopted conditions of sliding friction are used. They are determined as a result of appropriate data processing of model triboexperimental studies. This type of mentioned polymeric materials wear investigations is carried out according to the author's methodology [30-32], which provides several-step loading of the tribocouple. At the same time, two tribological factors are determined: samples wear (linear or mass) and specific friction force τ by Coulomb. Actually, in the author's methodology of investigating the wear kinetics under sliding fraction [29-31] by the mechanism of friction fatigue fracture of materials tribosystem elements materials wear rate just functionally depends on the level τ. The pin-on-disk friction scheme, which ensures that the contact pressures and the sliding rate do not change during the investigations, was used.

According to experimental studies, Indicators of wear resistance Φ_i for polymer materials under each contact pressure p_i stage are determined by investigation experiments

$$\Phi_i = L_i / h_i, \quad (10)$$

where L – friction path; h_i – linear wear of samples under i-th level of contact pressures p_i.

For triboexperimental studies of metal-polymer tribocouples, the following program was adopted: sliding speed v = 0.4 m/sec; contact pressure p = 2, 4, 6, 8 MPa; friction path L = 5000… 10000 m; diameter of both finger samples d = 3 mm. Polymeric sample temperature T = 23 ± 1°C at the air relative humidity HR 50 ± 5% (ISO 7148-2).

The experimental values of wear resistance indicators Φ_i of MP sliding bearings material are approximated by the following function [30-32]:

$$\Phi_k(\tau) = B_k \frac{\tau_{k0}^{m_k}}{(\tau - \tau_{k0})^{m_k}}. \quad (11)$$

In contrast to Archard's linear law of abrasive wear of materials under the influence of contact pressure p, a non-linear dependence of wear on the specific force of friction $\tau = fp$ is assumed here. To determine the wear resistance characteristics B_k, m_k, τ_{k0} of the tribocouple materials the least squares method is used.

For example, Fig. 2 presents the results of experimental studies of polyamides (markers indicators Φ_i) and their mathematical description (wear resistance diagrams $\Phi(\tau):\tau$) in the accepted range of the specific friction forces.

The established characteristics of polymers wear resistance B_1, m_1, τ_{10} are given in table. 1. Instead, the united wear resistance diagrams of all investigated polymeric materials are shown in Fig. 3. The qualitative nature of wear resistance change is almost the same except for the PTFE+15CF composite. While comparing these diagrams it is seen that PET+20PTFE composite has the highest wear resistance among polyamides. Polymers PEEK and PA6+30GF have the lowest.

According to these united diagrams it is possible to compare the wear resistance of various polymeric materials under the same specific friction $\tau = fp(0)$ forces.

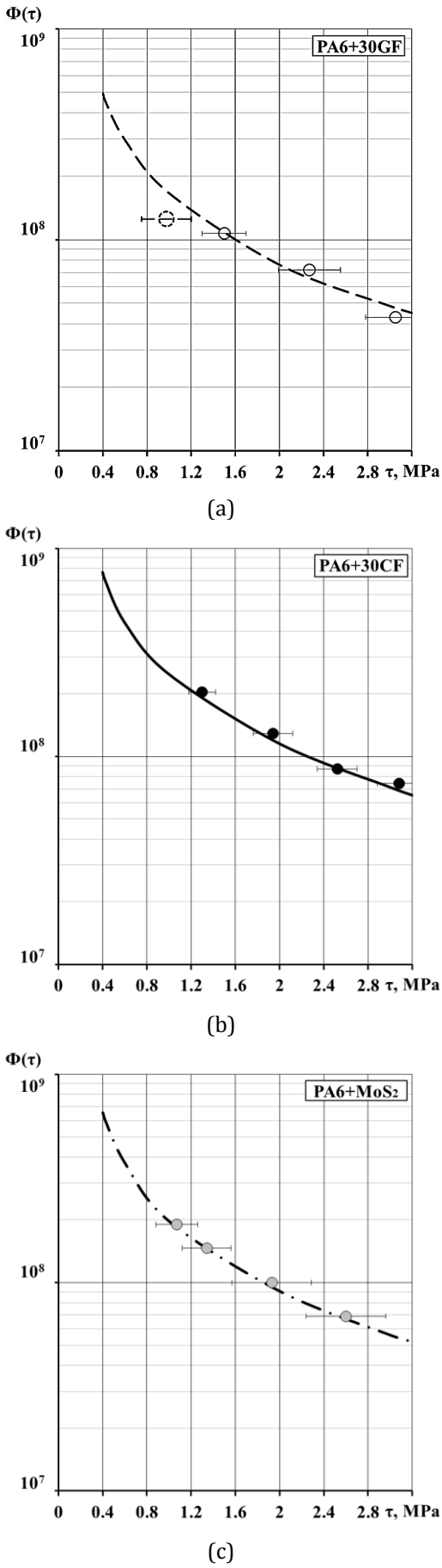


Fig. 2. Diagrams of polyamides wear resistance.

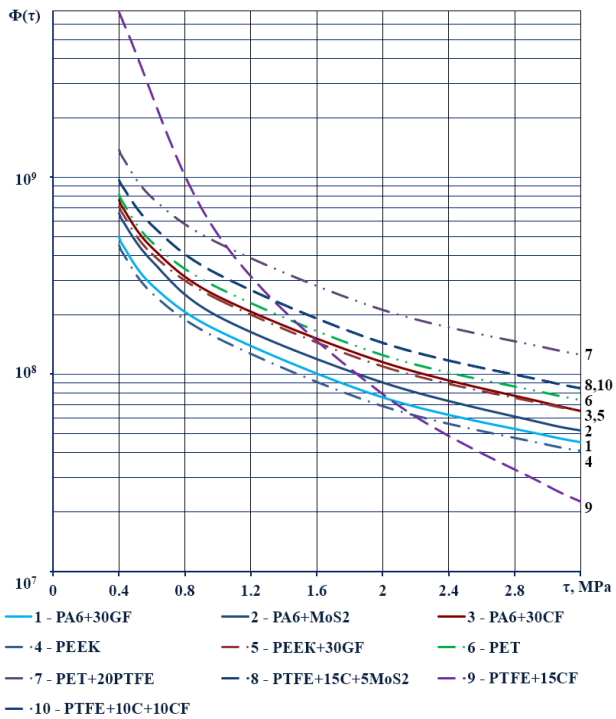


Fig. 3. United wear resistance diagrams of MP sliding bearings polymer materials.

4. SOLUTION NUMERICAL RESULTS AND DISCUSSION

Data for MP bearing calculations: D_2 = 2, 10, 20, 50 mm; D_2 = 2 mm - 2ε = 0,09 mm, D_2 = 10 mm - 2ε = 0,18 mm , D_2 = 20 mm - 2ε = 0,24 mm, D_2 = 50 mm - 2ε = 0,36 mm F = 2,5 D_2 (N/mm), $N = Fl$, $l = D_2$; ω_2 = 6,2832 sec^{-1}, T = 25°C, HR 50%, h_{1max} = ε. Bearings with such shaft and bushing materials were investigated: shaft – normalized carbon steel 0.45%C, grinding, R_a= 0.8-0.9 μm is the roughness of the steel sample; E_2 = 210000 MPa, ν_2 = 0,3; B_2 = 10^{13}, m_2 = 2, τ_{20} = 0.1 MPa; bushing – polymer materials (Table 1).

The characteristics of elasticity and wear resistance of the specified polymer materials were determined by the authors as a result of conducted research.

Dry friction coefficient f, determined experimentally for steel-polymer tribocouple, is given in Table 2.

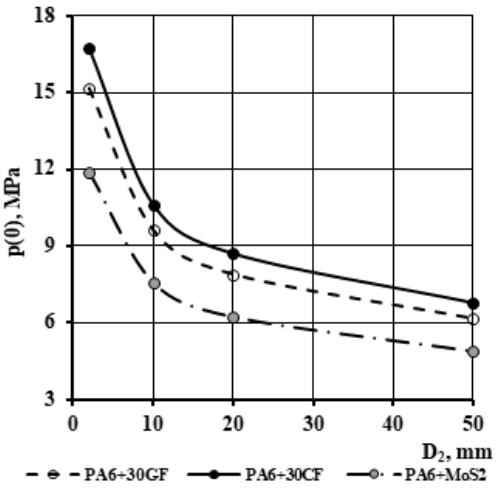
Figures 4-7 present the calculation results. Fig. 4 presents the dependence of maximum contact pressures on shaft diameter.

Table 1. Data about polymer materials.

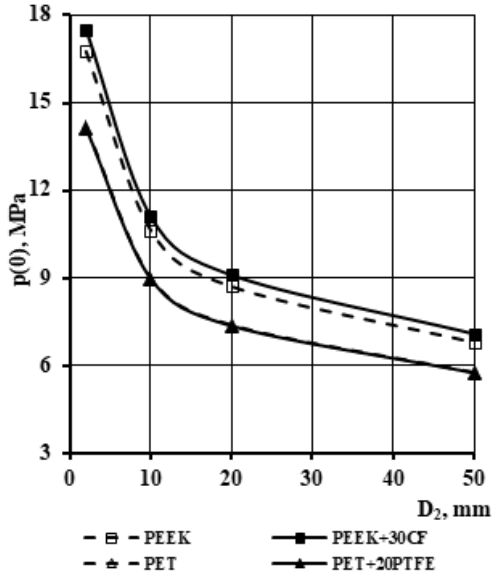
Polymer materials (manufacturer)	Elasticity and wear characteristics				
	E_1, MPa	ν_1	B_1 ·10^{10}	m_1	τ_{10}, MPa
PA6+30GF (Sustamid 6 GF 30 Rochling)	2700	0,41	4,12	1,09	0,05
PA6+30CF (Sustamid 6 ESD 60 Rochling)	3300	0,41	6,53	1,09	0,05
PA6+MoS_2 (Sustamid 6 MO Rochling)	1660	0,4	5,58	1,1	0,05
PTFE+15CF (Fibracon® PTFE + 15% Carbon Fibre Rochling)	880	0,46	108	2,6	0,05
PTFE+10C+10CF (Fibracon PTFE + 10% Carbon + 10% Carbon Fibre Rochling)	850	0,46	8,2	1,1	0,05
PTFE+15C+5MoS_2(Fibracon PTFE + 15% Carbon + 5% MoS2 Rochling)	850	0,46	8,2	1,1	0,05
PEEK (Sustapeek Rochling)	3650	0,4	3,76	1,09	0,05
PEEK+30CF(Sustapeek CF 30 Rochling)	3400	0,38	5,95	1,09	0,05
PET(Sustadur PET Rochling)	2300	0,43	6,81	1,09	0,05
PET+20PTFE (Sustadur PET GLD 130 Rochling)	2260	0,44	11,45	1,09	0,05

Table 2. Coefficient of sliding friction.

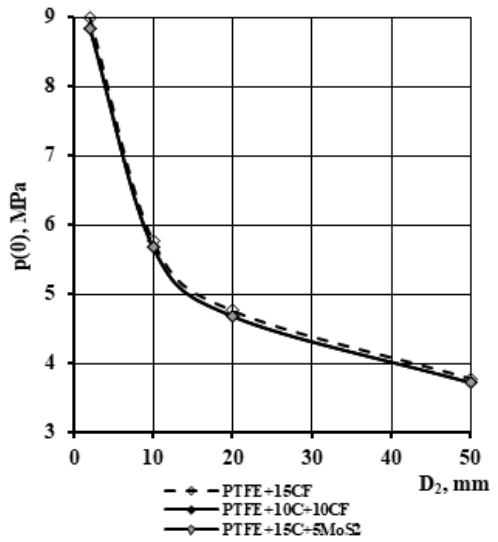
Polymer materials	F = 5 N/mm	F = 25 N/mm	F = 50 N/mm	F = 125 N/mm
PA6+30GF	0,36	0,48	0,52	0,59
PA6+30CF	0,3	0,39	0,42	0,47
PA6+MoS_2	0,21	0,27	0,29	0,34
PTFE+15CF	0,3	0,35	0,41	0,51
PTFE+10C+10CF	0,33	0,41	0,47	0,53
PTFE+15C+5MoS_2	0,3	0,35	0,41	0,51
PEEK	0,28	0,36	0,4	0,48
PEEK+30CF	0,3	0,38	0,44	0,5
PET	0,22	0,25	0,29	0,35
PET+20PTFE	0,21	0,28	0,3	0,35



(a)



(b)



(c)

Fig. 4. Influence of shaft diameter on contact pressures.

When the miniature diameter of the shaft is D_2 = 2 mm contact pressures reach the highest values as compared to larger diameters, although the external load increases linearly with the mentioned coefficient of proportionality, that is F = 2.5 D_2. Instead, the contact semiangle α_0 increases nonlinearly according to the law (4) and, accordingly, the area of contact $D_2\alpha_0$ increases also. The above standard diameter clearance 2 ε in the MP bearings depends nonlinearly on diameter D_2 (Fig. 5) at its miniature and small values. Therefore, the diagrams of pressure changes have the presented form.

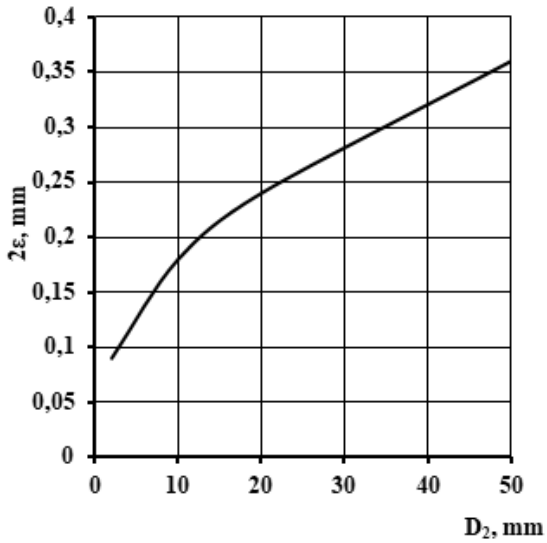


Fig. 5. Maximum clearances in MP bearings

The highest maximum contact pressures $p(0)$ occur in bearings with PTFE bushings (Fig. 4b). Almost the same contact pressures occur for PA6 + 30CF bushing. Young's modulus for these polymers is the largest. In the case of polytetrafluoroethylene composites bushings with low modulus of elasticity the pressures $p(0)$ will be 1.76… 1.94 times lower than in the case of mentioned polymers. Correlation of Young's modules between them is in the range of 3.9… 4.1 times. It should be noted that for identical types of polymers the filler type has little effect on the level of contact pressures in the bearing. Young's modulus here is decisive. This is shown in Fig. 4b, where polyethylene terephtalate (PET, PET + 20PTFE) have similar Young's modules, and Fig. 4c, where three polytetrafluoroethylene composites (PTFE + 15CF, PTFE + 10C + 10CF, PTFE + 15C + 5MoS_2) have very close modules of elasticity.

From the above results of pressure $p(0)$ calculations it follows that at the specified normative values of radial clearances in MP bearings, pressures increase significantly with the decrease of D_2 from 50 mm (large) to D_2 = 2 mm (miniature). It was found that this increase for different types of polymer materials with much differentiated modules of elasticity and slightly different Poisson's ratio (Table 1) is close in magnitude. In particular: for polyamides - 2.46 times, for polytetrafluoroethylenes - 2.38 times, for polyetheretherketones - 2.47 times, for polyethylene terephtalates - 2.46 times.

In the case of bushings made of polytetrafluoroethylene composites with a low modulus of elasticity, the pressures $p(0)$ are about 2 times lower than for other studied polymers. When the modulus of elasticity increases, the pressure level increases.

It should be noted that in engineering practice in the design calculation of metal plain bearings as a criterion for their load carrying capacity the conditional contact pressure p is widely used. It is taken as uniformly distributed over the contact area $2R_2l$, i.e.

$$p = \frac{F}{2R_2 l} = \frac{N}{D_2} \le [p], \tag{12}$$

where [p] is the allowable contact pressure of a less strong material (reference parameter).

Factors such as radial clearance $\varepsilon > 0$ and elasticity characteristics E, ν of shaft and bushing materials are not taken into account here. As mentioned above in paragraph 2, these parameters have a decisive influence on both the maximum contact pressures and the contact arc. In this conditional calculation, the contact arc has a length of $2\alpha_0 R_2 = D_2$ corresponding to the constant contact angle of $2\alpha_0 = 360^0 / \pi \approx 114.6°$ = const. However, such a significant contact angle is achieved with significant loads on the bearing and minimal radial clearances, which is known from the literature on solutions obtained by methods of the contact theory of elasticity. A modified method for estimating the maximum contact pressure p_{max}, which is even less close to actual contact conditions, is presented in [37].

$$p_{max} = \frac{4}{\pi}\frac{F}{D_2 l} = \frac{4}{\pi}\frac{N}{D_2} \tag{13}$$

It has the same disadvantages as the above method. Here it is assumed that the contact angle $2\alpha_0$ = 180°, and the pressure is distributed according to the cosine law, which assumes zero radial clearance.

The initial assumptions in these methods are far from real. A radial clearance must be guaranteed in the sliding bearing, because without it its reliable operation is impossible. Also, neglecting the elastic properties of the materials of the shaft and bushing does not correspond to reality, and even more so to the laws of linear mechanics of the contact of elastic deformable bodies. Here, the results of assessment of contact parameters are significantly (many times) distorted. That is, such methods are extremely simplified and very conditional, since they do not allow to objectively assess the actual pressure level in the sliding bearing.

Regarding metal-polymer sliding bearings, the use of this conventional calculation method is groundless in view of the very significant difference in Young's modulus of elasticity of steel (shaft) and composites (bushing) (Table 1), which was indicated in the introduction. Such a task is effectively solved for both metal and metal-polymer bearings by the classical analytical method of linear mechanics of contact of elastic deformable cylindrical bodies of close radii with internal contact, presented above, where the specified simplifications are absent.

In order to compare the maximum contact pressures $p(0)$ according to the developed method with the pressures p, p_{max} according to conventional methods of calculation in [38], the studies of MP bearings with bushings made of several polymer epoxy and thermoplastic materials, as well as a classic metal bearing with a pair of carbon steel (0.45C) - bronze Sn-Zn-Pb 6-6-6 were conducted. Accepted data for calculation: N = 2000 N; $F = N / l$ = 20 N/mm, l = 100 mm; ε = 0.1 mm; D_2 = 40 mm. The following results of this assessment were obtained [38]:

Table 3. Contact parameters, comparison of contact pressure.

Shaft - bushing	Calculated by		
	Author's methods	Eq. (12)	Eq. (13) [37]
Steel – PA6+30%GF [p] = 11 MPa	2.73 / 26.9	0.5 / 114.6 (5.46 times)	0.636 / 180 (4.29 times)
Steel – PA6+30%CF [p] = 11 MPa	3.14 / 23.4	0.5 / 114.6 (6.28 times)	0.636 / 180 (4.94 times)
Steel – DK6 (PT) [p] = 12 MPa	3.46 / 21.2	0.5 / 114.6 (6.92 times)	0.636 / 180 (5.44 times)
Steel – Moglice [p] = 14 MPa	4.48 / 16.3	0.5 / 114.6 (8.69 times)	0.636 / 180 (7.04 times)
Steel – Bronze [p] = 50 MPa	11.66 / 6.3	0.5 / 114.6 (23.0 times)	0.636 / 180 (18.05 times)

Note: PA6+30%GF - E_{GF} = 3900 MPa, ν_{GF} = 0.42; PA6+30%CF - E_{CF} = 5200 MPa, ν_{CF} = 0.42; epoxy composite Moglice - E_M = 11200 MPa, ν_M = 0.4; epoxy composite DK6 (PT) - E_{DK} = 6500 MPa, ν_{DK} = 0.4; E_{Br} = 114000 MPa; ν_{Br} = 0.34; composites filled with dispersed fibers PA6+30%GF, PA6+30%CF were dried; in the numerator – contact pressures (MPa), and in the denominator – contact angles (degrees).

The load carrying capacity at the allowable contact pressure [p] in MP bearings with polymer bushing materials according to conventional calculation methods is exhausted much later than it is expected according to the studies carried out according to the author's method. That is, the load carrying capacity of the bearings is unreasonably overestimated.

Fig. 6 presents the calculation results of durability for MP bearings with different diameters of the shaft.

The durability of the bearings is differentiated in the case of each investigated polymer. The highest durability is for bearings with bushings made of polyamide composites - PA6 + MoS_2, polyethylene terephtalate - PET + 20PTFE composite, and polytetrafluoroethylene composites PTFE + 10C + 10CF, PTFE + 15C + 5MoS_2 have approximately the same durability. The bearing with the PTFE + 15CF composite bushing will have very low durability.

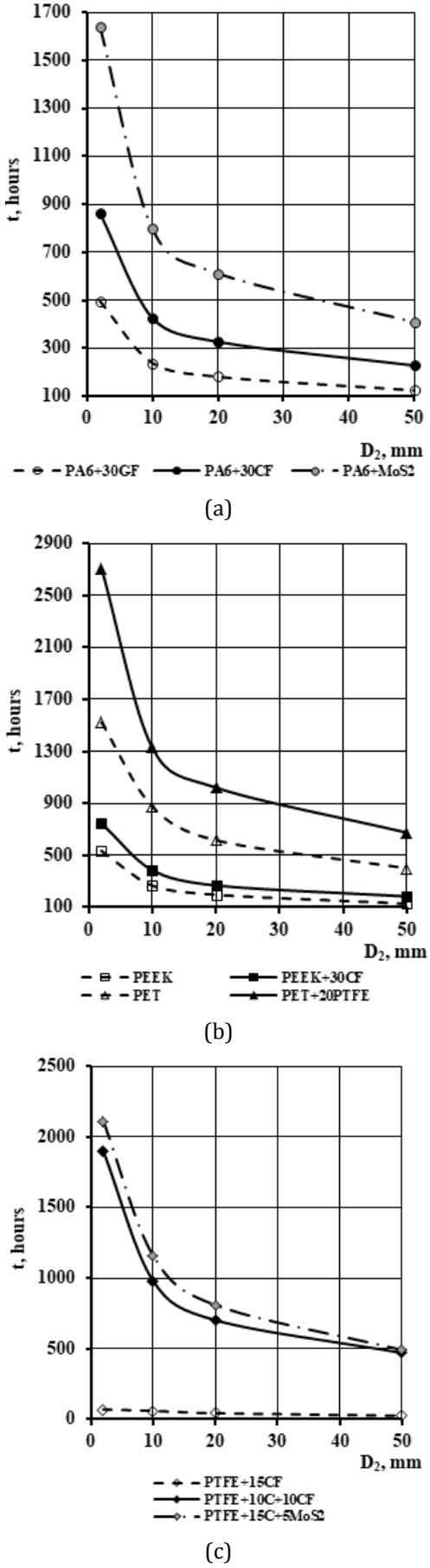


Fig. 6. Influence of shaft diameter on bearings durability.

Fig. 7 presents the comparison of durability values for MP bearings with the bushings made of different types of polymer materials.

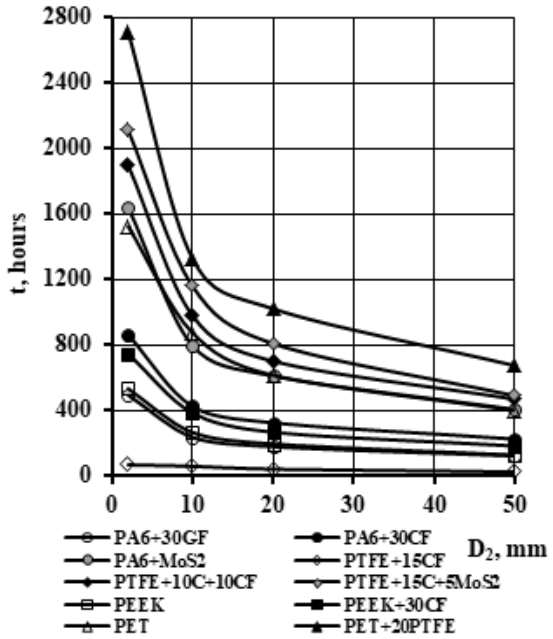


Fig. 7. Comparative durability of bearings.

The highest durability will have MP bearing with PET + 20PTFE, slightly lower with PTFE + 15C + 5MoS_2 (1.28… 1.34 times), PTFE + 10C + 10CF (1.43 times), even lower with PA6 + MoS_2 (1.65 times), PET (1.78… 1.68 times). Other polymer materials have a mediocre durability (3 or more times lower) as compared with mentioned this highly effective polymer composite. The bearing with PTFE + 15CF will have unsatisfactory durability (71… 28 h).

5. CONCLUSION

As a result of the carried out analytical research by the given method of calculation for MP bearings with bushings made from four groups of polymeric materials and their model triboexperimental researches the following results are received.

1. The nonlinear dependence of maximum contact pressures decreases on the shaft diameter in the MP bearings, especially in the range D_2 = 2… 20 mm is observed. Further with the shaft diameter increase, it becomes closer to linear.
2. Similarly, also the durability of bearings with bushings made of the tested polymeric materials decreases except for PTFE + 15CF composite when it decreases linearly. However, the bearing with the bushing made of this PTFE composite has a very low durability under dry friction.
3. According to of research it is established that when base polymers are modified with solid lubricants (PTFE, MoS_2, and graphite) their durability is high. On the other hand, when the filler is one – fibreglass or carbon fibber – the durability of MP bearings is much lower.
4. The calculated durability of MP sliding bearings, established according to the presented analytical method, is a function of a complex of various factors: shaft diameter, radial clearance, contact pressures and specific frictional forces, elastic properties of polymers, their wear resistance characteristics and sliding friction coefficient. They are included in a relation (9) to estimate bearing durability.
5. According to the developed calculation methods, it is possible to optimize the MP bearing in a fairly simple way according to various parameters: contact pressures, ultimate durability, type of polymeric bushing materials, and their wear resistance. Partially obtained research results can also be used for this purpose.

A characteristic feature of the presented research methods is their availability for effective use in the practice of design calculation and optimization of MP sliding bearings. Prototypes or similar methods of the indicated types of research on MP bearings have not been developed. There are no data on developed prototypes or similar methods of the specified types of MP bearing research in the literature

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