

Investigation of the Effect of Young's Modulus on the Contact Strength of Metal Polymer Plain Bearings

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ABSTRACT

The load carrying capacity (contact pressures) of hybrid (metal-polymer) plain bearings is calculated in the article according to the author's research method, considering the differences of the Young's modulus of the bushing made of polymeric materials. Studies have been conducted for metal-polymer bearings with a bushing of two types of polymer composites: epoxy based antifriction composite materials DIAMANT Moglice (DIAMANT Metallplastic GmbH) and DK6 (PT) (fillers – graphite, MoS₂, Zr). The elastic constants of composite materials, in particular the Young's modulus, have a noticeable effect on the contact pressures in metal-polymer bearings. The Young's modulus is significantly different in these composite materials. In accordance with this factor, the influence of load, bushing diameter and radial clearance on the maximum contact pressures was studied. Quantitative and qualitative regularities of maximum contact pressures from the accepted factors of influence: Young's modulus, loading, bushing diameter, radial clearance, type of bushing material are established. The obtained results are compared with the results obtained by known conventional calculating methods of contact pressures

Keywords: metal-polymer plain bearings, epoxy composites Moglice and DK6 (PT), polyamide composites PA6+30%GF and PA6+30%CF, calculation method, Young's modulus, maximum contact pressures.

INTRODUCTION

Plain bearings of various types are used in mechanical engineering and in other various fields due to a number of advantages over rolling bearings. In particular, these are: high load capacity, a wide range of shaft diameters, a very large range of rotational speeds, good susceptibility to dynamic and shock loads, small lateral dimensions compared to their diameter, low noise level; possibility of operation with dry friction, in water or aggressive media, etc.

Along with bearings made of metallic materials, various non-metallic materials (porous and

cermet, carbographite, layered composites, polymers, filled composites on a polymer or other basis) are widely used for the bushings. The number of new types of composite materials that can be used in metal-polymer bearings is increasing.

A feature of metal-polymer (MP) bearings is a significant difference in the strength characteristics of the shaft and bushing materials (8–10 times) and Young's modulus (40–100 times), which fundamentally affects the contact pressure and bearing capacity. The calculation methods [1–3, 4–6] known in the literature for studying contact pressures in metal plain bearings are practically not used in engineering practice for MP

bearings. Only in [8–10] the contact pressures in metal-polymer plain bearings were investigated by numerical methods. Also in [7] the analysis of tribological efficiency of polymeric composite materials - steel hybrid joint at their application in tribosystems of dry sliding friction is carried out. Although they do not take into account the aspect of the effect of the Young's modulus of polymer materials on contact pressures when designing MP plain bearings.

This article is devoted to the study of this issue. Known methods [11–13] for calculating metal plain bearings are modified for the study of MP bearings [14, 15]. With their use, the calculation of the arising contact pressures was carried out taking into account the peculiarities of the contact interaction of a hybrid pair of materials with significantly different mechanical properties and elastic characteristics. It also effectively provides a qualitative and quantitative analysis of the arising maximum contact pressures and the establishment of regularities of their change from the influence of various factors. The influence of Young's modulus of polymeric materials on contact pressures is investigated in the article.

METHODS

The plain problem of the theory of elasticity about the internal contact of cylindrical bodies of close radii is to be studied. Below is a method for solving it. The bearing analytical model is shown in Figure 1b.

The bearing shaft is subjected to a static radial load F . In order to bring the 3D bearing assembly (Fig. 1a) to the 2D assembly (Fig. 1b), the total load is assigned to the length l of the bushing. That is, in the plane task reduced concentrated load $N = F/l$ will act on disk 2 (unit width of the shaft journal). A small radial clearance $\varepsilon = R_1 - R_2 \geq 0 \ll R$ is provided between the shaft journal 2 and the bushing l to ensure bearing lubrication. Shaft 2 with a radius R_2 is made of metal (steel), and the bushing l with a radius R_1 is made of polymer composites. The composite bushing is located in housing 3. When the bearing is loaded with force N , unknown contact pressures occur in the contact area $2R_2\alpha_0$. They are supposed to be found as a result of solving the problem.

To calculate the static contact pressures $p(\alpha)$ occurring in the bearing, the following integral-differential equation is used [11–13]:

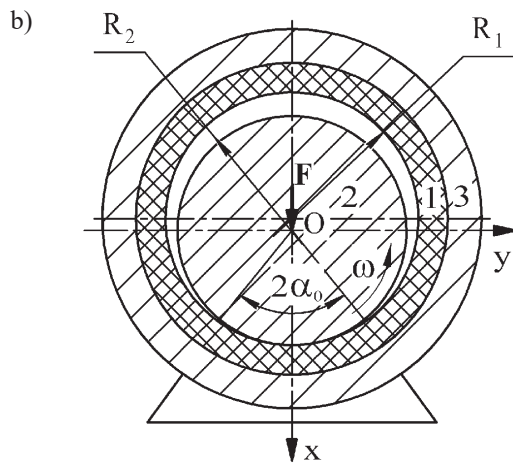


Fig. 1. Metal-polymer plain bearing: a) general view, b) analytical model

$$c_1 \int_{-\alpha_0}^{\alpha_0} \cot \frac{\alpha - \theta}{2} p'_0 d\theta = c_2 p(\alpha) + c_3 \int_{-\alpha_0}^{\alpha_0} p(\alpha) d\alpha + c_4 \cos \alpha \int_{-\alpha_0}^{\alpha_0} p(\alpha) \cos \alpha d\alpha + \frac{\varepsilon}{R^2} \tag{1}$$

where: $p'_\theta = dp / d\theta$; $0 \leq \alpha \leq \theta$, $0 \leq \theta \leq \alpha_0$;
 $c_1 = \frac{1}{8\pi} \left(\frac{1+\kappa_1}{G_1 R_1} + \frac{1+\kappa_2}{G_2 R_2} \right)$; $c_2 = \frac{1}{4} \left(\frac{1-\kappa_1}{G_1 R_1} - \frac{1-\kappa_2}{G_2 R_2} \right)$;
 $c_3 = \frac{1+\kappa_1}{8\pi G_1 R_1}$; $c_4 = \frac{1}{2\pi} \left(\frac{\kappa_1}{G_1 R_1} + \frac{1}{G_2 R_2} \right)$; $R_1 \approx R_2 = R$,
 $\kappa = 3 - 4\nu$, $E = 2G(1 + \nu)$.

The approximate solution of equation (1) for determining the contact pressures $p(\alpha)$ is performed by the collocation method for two symmetric collocation points with a polar angle $\alpha = \pm 0.5\alpha_0$. The contact pressure function is taken as [11–13].

$$p_\alpha \approx E_0 \varepsilon \sqrt{\tan^2 \frac{\alpha_0}{2} - \tan^2 \frac{\alpha}{2}}; 0 < \alpha_0 < 90^\circ \tag{2}$$

where: $E_0 = (e/R_2)\cos^2(\alpha_0/4)$, $e = 4E_1E_2/Z$,
 $Z = (1+\kappa_1)(1+\nu_1)E_2 + (1+\kappa_2)(1+\nu_2)E_1$.

The maximum contact pressure p_0 characterizing the strength of the bearing, occurs when $\alpha = 0$.

$$p(0) \approx E_0 \varepsilon \tan(\alpha_0 / 2) \quad (3)$$

Substituting in (3) the expression for E_0 it is obtained that:

$$p(0) = (\varepsilon \varepsilon / R_2) \cos^2(\alpha_0 / 4) \tan(\alpha_0 / 2) \quad (4)$$

The contact pressure depends on the radius of the shaft, the radial clearance, the contact angle and the elastic characteristics of the materials.

To determine the unknown semiangle of contact α_0 , the equilibrium condition of the forces applied to the shaft is used

$$R_2 \int_{-\alpha_0}^{\alpha_0} p(\alpha) \cos \alpha d\alpha = 4\pi R_2 E_0 \varepsilon \sin^2(\alpha_0 / 4) \quad (5)$$

Given the expression for the collocation coefficient E_0 , condition (5) to determine the contact semiangle will take the form

$$N = \pi \varepsilon \sin^2(\alpha_0 / 2) \quad (6)$$

Accordingly, after the transformations

$$\alpha_0 = 2 \arcsin \sqrt{N / \pi \varepsilon} \quad (7)$$

Analysis of this dependence shows that the contact angle in the joint does not depend on the shaft radius, but on the load, radial clearance and elastic characteristics of the materials.

The analytical solution is simple and can be implemented by various software tools, including Microsoft Excel. That is, the given calculation method is easy to implement in engineering design calculations. The sequence of the solution is as follows:

1. According to Eg. (6) at the given values of parameters the contact semiangle α_0 is determined. Accordingly, the contact arc (area) will be $2R_2\alpha_0$.
2. According to Eg. (3) or (4), the maximum contact pressure is calculated.

In engineering practice in the design calculation of metal plain bearings as a criterion for their load carrying capacity the average pressure p is widely used. It is taken to be evenly distributed over the contact area $2R_2l$, ie

$$p = \frac{F}{2R_2l} = \frac{N}{D_2} \leq [p] \quad (8)$$

From the analysis of Eg. (8) it follows that the formula lacks such parameters as the radial clearance $\varepsilon = R_1 - R_2 > 0$ and the elastic characteristics E, ν of the shaft and bushing materials. As mentioned above, these parameters have a decisive influence on both the contact pressure and the contact arc. In this conventional calculation method, the contact arc has a length of $2R_2 = D_2$ corresponding to a constant contact angle of $2\alpha_0 = 360^\circ / \pi \approx 114.6^\circ = \text{const}$. However, such a significant contact angle is achieved with significant loads and minimal radial clearances. This is known from the literature [1-2, 11-13], etc.] and the author's works [11-13], obtained by the methods of contact theory of elasticity.

A slightly different modified formula for determining the maximum contact pressure p_{max} in a bearing has been proposed in [16]

$$p_{max} = \frac{4 F}{\pi D_2 l} = \frac{4 N}{\pi D_2} \quad (9)$$

It is assumed that the contact angle $2\alpha_0 = 180^\circ$, and the pressure is distributed according to the cosinusoidal law. The maximum pressure p_{max} according to (9) will be 1.273 times higher than the average pressure p according to (8). It also does not take into account the elastic characteristics of materials. And at a contact angle $2\alpha_0 = 180^\circ$ the radial clearance is zero. Such initial assumptions in this technique do not correspond to reality, because in the plain bearing must be a certain radial clearance, because without it its reliable operation is impossible.

Therefore, the simplified (conventional) methods of calculating metal plain bearings according to the criterion of contact pressure p do not allow objectively assessing the actual pressure level. With regard to metal-polymer plain bearings, the use of these techniques is virtually unfounded given that they do not take into account the significant differences in the elastic characteristics of steel (shaft) and composite bushing, as indicated in the introduction. For this purpose, it is reasonable to use the above classical analytical method of the mechanics of contact of close radii cylindrical bodies with internal contact, where these simplifications are absent.

SOLUTIONS

Data for calculation: $F = 500, 750, 1000, 2000 \text{ N}$; $N = F / l = 5, 7.5, 10, 20 \text{ N/mm}$, $l = 100 \text{ mm}$; $\varepsilon = 0.05, 0.075, 0.1 \text{ mm}$; $D_2 = 40, 50 \text{ mm}$.

Bearing materials: bushing - polymer composites Moglice, DK6 (PT) (Table 1); shaft - steel 45, martempering - $E_2 = 210000$ MPa, $\nu_2 = 0.3$.

The results of the calculation of the maximum contact pressures $p(0)$ are given in Figs. 2–4. Figures (a) correspond to the Moglice composite, and figures (b) correspond to the DK6 (PT) composite. Solid lines show graphs for $D_2 = 40$ mm, and dashed lines for $D_2 = 50$ mm.

Figure 2 shows the dependence of the maximum contact pressures $p(0)$ on the load N when changing the radial clearance ϵ in the joint.

There is an almost linear dependence of pressures $p(0)$ on the load $N = 5 \dots 20$ N/mm ($F = 500$,

750, 1000, 2000 N) at different radial clearances ϵ and the studied shaft diameters D_2 . When the load N in the specified range is increased 4 times, the maximum contact pressures $p(0)$ increase 2 times regardless of the change in the values ϵ of the radial clearance ϵ and the shaft diameter D_2 . The diameter D_2 increase by 1.25 times causes a proportional decrease in pressure $p(0)$. In a bearing with a Moglice composite bushing, the pressures will be about 1.3 times higher than in a bearing with a DK6 (PT) composite bushing. This is due to the greater rigidity of Moglice due to the higher value of Young's modulus.

The dependence of the maximum contact pressures $p(0)$ on the radial clearance ϵ when the load N changes is shown in Figure 3.

Increasing the radial clearance ϵ leads to an almost linear increase in the maximum contact pressures $p(0)$ at all loads N and shaft diameters D_2 . The doubling of the radial clearance ϵ leads to a $\sqrt{2}$ -fold increase in the pressure $p(0)$ regardless of the change in the magnitude of the load N and the shaft diameter D_2 .

Table 2 summarizes the maximum contact pressures $p(0)_M$ and $p(0)_{DK}$. The relative increase \tilde{p} of pressures $p(0)_M$ in relation to $p(0)_{DK}$ is indicated.

These results show that the ratio is almost unchanged ($\tilde{p} = const$) with changes in load, radial clearance in the joint and shaft diameters.

Table 1. Physical and mechanical characteristics of the Moglice and DK6 (PT) composites

Characteristics	Units	Value Moglice	Value DK6 (PT)
Density	g/sm ³	1.7	2.0
Young's modulus	MPa	11200	6500
Poisson's ratio	-	0.4	0.4
Compressive strength	MPa	120	150
Flexural strength	MPa	64	60
Shore hardness (maximum)		90	90
Heat resistance: - short-term - long-term	°C	-40 ... +125 -20 ... +60	-40 ... +125 -20 ... +90

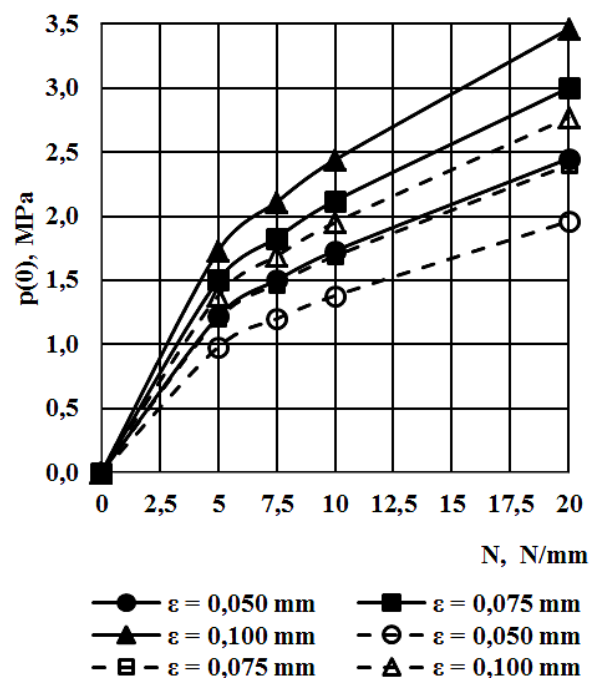
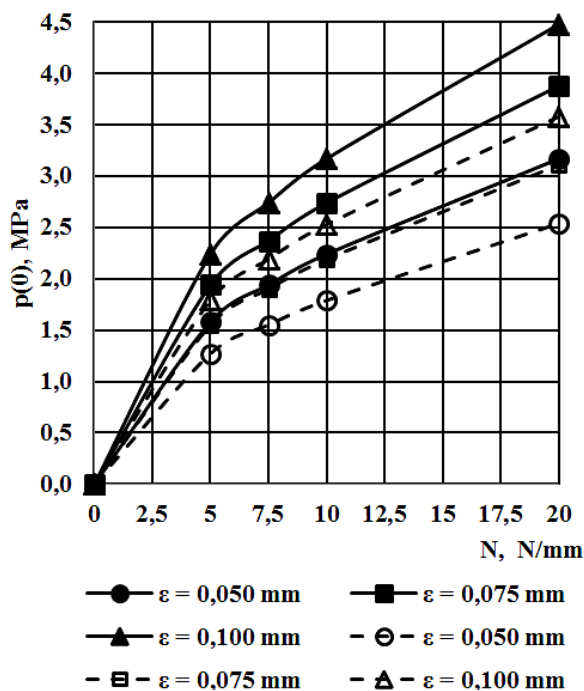


Fig. 2. Effect of the load on the maximum contact pressures

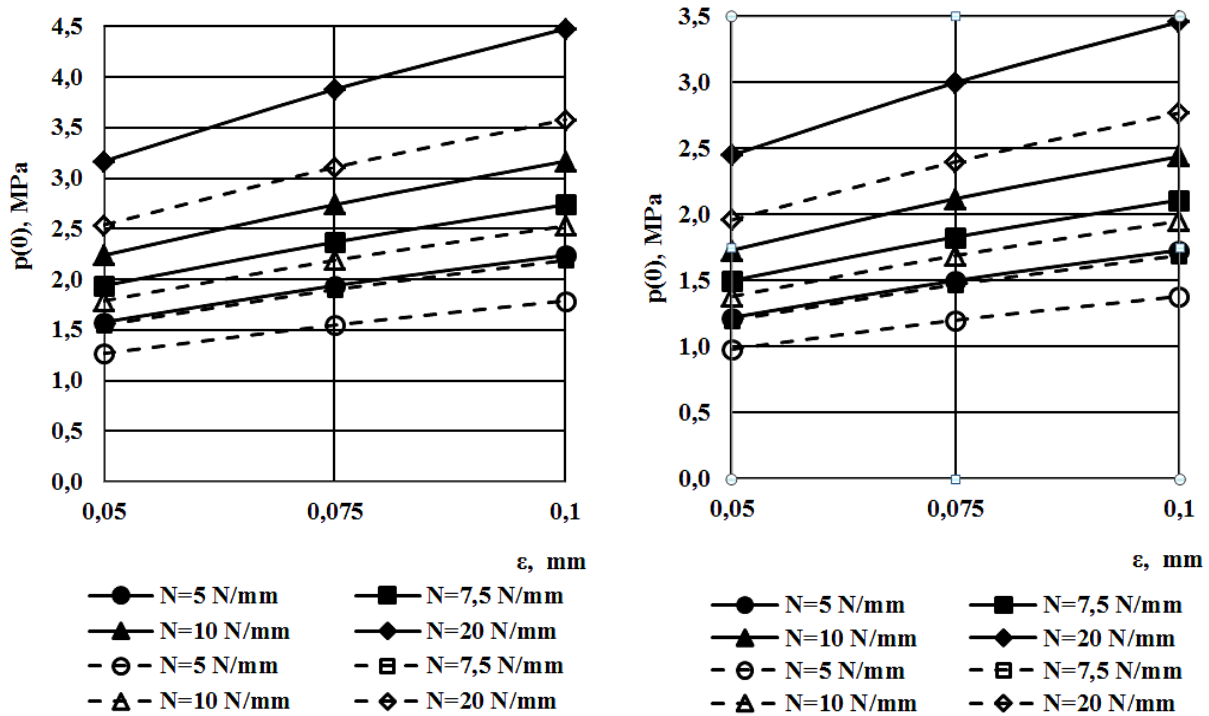


Fig. 3. Effect of the radial clearance on the maximum contact pressures

We also compared the maximum contact pressures $p(0)$ with the pressures p and p_{max} calculated by Eq. (8) and Eq. (9) correspondingly [14, 15] in MP bearings with bushings made of Moglice, DK6, PA6+30%GF (polyamide filled with 30% dispersed fiberglass), PA6+30%CF (polyamide filled with 30% dispersed carbon fiber) composites. The following data are accepted for calculation: $F = 2000$ N; $N = F/l = 20$ N/mm, $l = 100$ mm; $\epsilon = 0.1$ mm; $D_2 = 40$ mm. In addition to these metal-polymer bearings, a metal bearing with a pair of steel 45 – bronze (Sn – 6%, Zn – 6%, Pb – 6%, Cu – 82%) was also investigated. The results of the calculations are given in Table 3.

According to the data [3] ($F = 100$ N, $D_2 = 19$ mm, $\epsilon = 0.25$ mm, $l = 10$ mm, $N = 10$ N/mm, $E = 970$ MPa, $\nu = 0.35$, bushing material – polyamide PA) MES found that in the MP bearing $p_{max} = 2.9$ MPa, and according to the above method – 3.48 MPa.

Accordingly, Fig. 4 shows the dependence of the maximum contact pressures in the investigated bearings with increasing Young’s modulus of the bushing materials.

The data obtained by the given author’s method show that the Young’s modulus has a significant effect on the contact parameters. The Poisson’s ratio may also have a slight effect. Instead,

Table 2. Maximum contact pressures and their ratios

Load N , N	Contact pressures	Radial clearance ϵ , mm		
		0.05	0.075	0.1
500	$p(0)_M$, MPa	1.58 / 1.27	1.94 / 1.555	2.24 / 1.785
	$p(0)_{DK}$, MPa	1.22 / 0.98	1.50 / 1.20	1.73 / 1.38
	\bar{p} , times	1.295 / 1.295	1.293 / 1.296	1.295 / 1.293
750	$p(0)_M$, MPa	1.94 / 1.555	2.37 / 1.904	2.73 / 2.19
	$p(0)_{DK}$, MPa	1.50 / 1.20	1.83 / 1.47	2.11 / 1.69
	\bar{p} , times	1.293 / 1.296	1.295 / 1.295	1.293 / 1.295
1000	$p(0)_M$, MPa	2.24 / 1.785	2.74 / 2.19	3.16 / 2.525
	$p(0)_{DK}$, MPa	1.73 / 1.38	2.12 / 1.69	2.44 / 1.95
	\bar{p} , times	1.294 / 1.293	1.292 / 1.295	1.295 / 1.295
2000	$p(0)_M$, MPa	3.17 / 2.54	3.88 / 3.11	4.48 / 3.584
	$p(0)_{DK}$, MPa	2.45 / 1.96	3.00 / 2.40	3.46 / 2.77
	\bar{p} , times	1.293 / 1.295	1.293 / 1.295	1.294 / 1.294

Note: in columns the data in the numerator at $D_2 = 40$ mm, and in the denominator at $D_2 = 50$ mm.

Table 3. Contact parameters, contact pressure ratio

Shaft - bushing	Calculated by		
	Author's methods	Ex. (8)	Ex. (9)
Steel – PA6+30%GF [ρ] = 11 MPa	2.73 / 26.9	0.5 / 114.6 (5.46 times)	0.636 / 180 (4.29 times)
Steel – PA6+30%CF [ρ] = 11 MPa	3.14 / 23.4	0.5 / 114.6 (6.28 times)	0.636 / 180 (4.94 times)
Steel – DK6 (PT) [ρ] = 12 MPa	3.46 / 21.2	0.5 / 114.6 (6.92 times)	0.636 / 180 (5.44 times)
Steel – Moglice [ρ] = 14 MPa	4.48 / 16.3	0.5 / 114.6 (8.69 times)	0.636 / 180 (7.04 times)
Steel – Bronze [ρ] = 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, $\nu_{GF} = 0.42$; PA6+30%CF - $E_{CF} = 5200$ MPa, $\nu_{CF} = 0.42$; $E_{Br} = 114000$ MPa; $\nu_{Br} = 0.34$; in the numerator – contact pressures (MPa), and in the denominator – contact angles (degrees)

according to simplified (conventional) methods, the pressures do not depend on the combination of shaft and bushing materials, which, of course, does not correspond to reality, and, moreover, the regularities of linear contact mechanics of elastic deformed bodies. The load carrying capacity at the allowable contact pressure of the bearings with polymer bushings (Table 3) according to the conditional calculation is exhausted much later than it will be according to the above method according to the results of research.

CONCLUSIONS

Based on the results of the numerical solution, the maximum contact pressures in MP bearings with an epoxy composite bushing are determined. The quantitative and qualitative regularities of their change have been established, taking into account the following factors: Young's modulus, load, shaft diameter, radial clearance in bearings, type of composite materials.

An increase in the load N on the shaft by a factor of 4 leads to a decrease in $p(0)$ by a factor of $\sqrt{4} = 2$, and an increase in the shaft journal diameter D_2 by a factor of 1.5 leads to a decrease in $p(0)$ by a factor of $1.5 \sqrt{1.5}$. The established ratios are independent of radial clearance and bushing material. The contact pressure $p(0)$ will

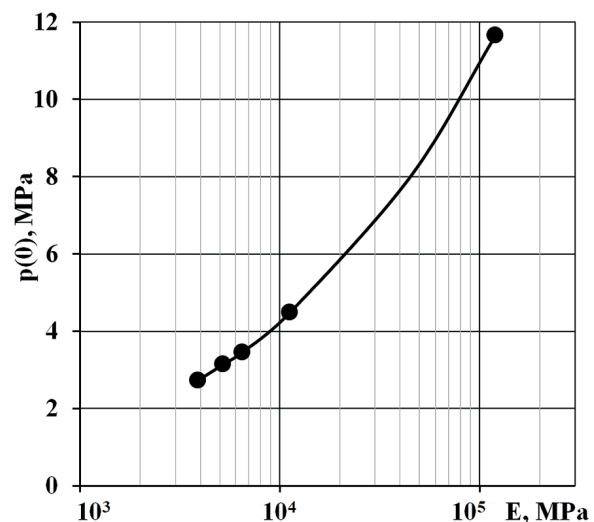


Fig. 4. Effect of the Young's modulus on the contact pressures in bearings

increase in proportion to the increase in the radial clearance ϵ . Since the Moglice composite has a higher Young's modulus than the DK6 (PT) composite, the ratio of maximum pressures $p(0)$ in bearings with such bushing materials is approximately $\sqrt{E_M / E_{DK}} = 1.31$ times. In this case, about 1.293...1.296 times (Table 2). A typical conventional calculating method of contact pressures gives a significant underestimation of the values of contact pressures in MP bearings (Table 3): with a bushing made of Moglice ($E_M = 11200$ MPa) in 8.7 times, DK6 (PT) ($E_{DK} = 6500$ MPa) in 6.9 times, PA6+30%CF ($E_{CF} = 5200$ MPa) in 6.28 times, PA6+30%GF ($E_{GF} = 3900$ MPa) in 5.46 times. In the case of a metal bearing with a bronze bushing ($E_{Br} = 114000$ MPa), this ratio is very large (23.3 times).

NOMENCLATURE

- α is the polar angle;
- G_1, G_2 are the shear modules of materials;
- ν_1, ν_2 are the Poisson's ratios;
- κ is the Muschelishvili constant for plain deformation;
- E is the Young's module;
- E_0 is the collocation coefficient,
- $[p]$ is the allowable value of contact pressure of less durable material (reference parameter).

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