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Prediction of the Contact Pressures and Resource of Metal-Polymer Linear Cylindrical Plain Bearings

Different types of linear bearings (guides) are widely used in practice in various equipment, as in many other areas of human activity. In particular, this applies to cylindrical guides (linear plain bearings) of reciprocating motion. However, despite their considerable distribution, in fact, there are no reasonable methods for calculating the wear and service life of not only metal but also metal-polymer linear bearings. According to the author's method, the influence of load, base diameter and radial clearance on the maximum initial pressures in this bearing was investigated within the plane contact problem of the theory of elasticity. Further, using the computational method according to the author's tribokinetic model of wear during sliding friction, the effect of the composite bushing wear on the change of the initial contact characteristics (contact pressures and contact angle) was evaluated. The forecast calculation of the service life of the bearing depending on the above factors is also carried out. Quantitative and qualitative regularities of dependence of contact pressures and a resource on the accepted factors are established.

Keywords: *metal-polymer linear cylindrical plain bearings, calculation method of contact parameters, resource calculation method, polymer composites*

1. INTRODUCTION

Linear plain bearings (plain and cylindrical guides), which provide reciprocating motion in mobile connections, are widely used both in mechanical engineering and in many other areas. In particular in forging and pressing equipment, metal - cutting machines, hoisting - transport mechanisms, agricultural machines, packing equipment, equipment of food and processing industry, equipment for wood and plastics processing, aerospace technology, positioning drives, laboratory and medical equipment, measuring devices, etc.

Cylindrical guides are also quite common. They have a base made of steel or cast iron, and a slider bus–hing made of bronze or anti-friction cast iron. However, a hybrid combination of materials friction pair is increasingly used, when one of its elements may be made of composite polymeric materials. In particular, such technologies are used to restore worn contact sur-faces by applying a layer of composite material. In these hybrid friction pairs, the materials will have fundamentally different strength, elastic characteristics and wear resistance.

However, even for the classical type of linear plain bearings made of metallic materials, no appropriate calculation methods have been developed, although this is necessary for the needs of engineering practice. To study the contact strength, wear and resource of this type of linear bearings with metal elements, the known computational methods [1-6] and numerical methods (Ansys) [7-9] were not used. With regard to metal-polymer linear bearings, there are no effective and sound computational methods for their study. Only in [10-12] experimental studies and numerical modeling by FEM of contact pressures and their changes due to wear in polymer composite bearings were performed. As for the methods of calculation (simulation) of the resource of this type of bearings, they are absent. Also by the method of triboelements [13] a certain attempt was made to estimate the wear of a thin elastic layer on the rigid bushing of the sliding bearing during reciprocating motion.

In practice, plain bearings with rotational shaft motion are also widespread, where hydrodynamic lubrication is realized, in particular, the study of this type of bearings is presented in [14-16]. However, due to the occurrence of the oil film bearing layer at a steady rotational motion, there is no mechanical contact between the shaft and the bushing, which is a feature of such bearings. Accordingly, the wear of its contact parts will be absent here, unlike plain bearings of rotary and linear motion operating in conditions of boundary or dry friction. In hydrodynamic friction, it is also customary to determine the pressure in the oil film and the thickness of the oil layer, rather than contact pressures, as a criterion for their bearing capacity. Hydrodynamic friction mode cannot be provided in linear plain bearings.

Instead, according to the developed author's methods for the study of plain bearings with metal elements [17-22], in [23-25] the wear kinetics of cylindrical linear plain bearings with metal elements operating in con–

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ditions of boundary friction was studied. Regarding the study of this structures type with reciprocating motion of one of the elements, they are known in the literature regarding pneumatic cylinders [26-29] and hydraulic cylinders [30–33]. This is not about wear, but about determining the friction force in the piston and rod seals, taking into account their geometry, diameter, operating conditions in the cylinders (pressure, speed, type of friction, etc.).

Below are the main principles of the developed computational method for the study of contact pressures and resource of metal-polymer linear cylindrical plain bearings with a bushing of polymer composites Moglice and DK6. The results of evaluation of the contact parameters from the load, the diameter of the base of the linear bearing, the radial clearance in the connection, taking into account the wear and service life of the bearing are also presented.

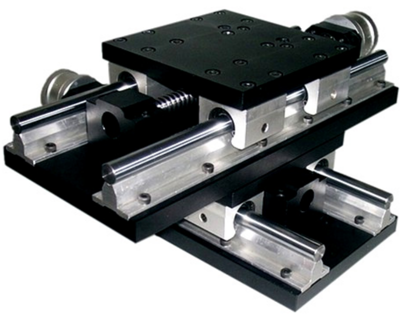
2. PROBLEM FORMULATION

There are different types of linear cylindrical guides with one (Fig. 1a), two (Fig. 1b) and four (Fig. 1c, d) bases.

A linear cylindrical sliding bearing (cylindrical guide) with one base (Fig. 1a) is modeled by an elastic base 2, along which an elastic slider 3 with a non-metallic bushing 1 performs a rectilinear reciprocating motion at a constant speed v (Fig. 2).

a)

b)



c)

d)

Figure 1. Linear cylindrical plain bearings

In the guide (Fig. 2) between the polymer bushing 1 with an inner radius R_1 and the steel base of the radius R_2, there is a radial clearance $\varepsilon = R_2 - R_1$. The polymer bushing of the slider and the steel base of the guide have significantly different elastic characteristics and different wear resistance. A static force F is applied to the slider 3, under the influence of which contact pressures $p(\alpha)$ arise in the contact area $2\alpha_0$, the magnitude and distribution of which is unknown. Further study of this guide was carried out as a plane contact problem of the theory of elasticity for cylindrical bodies, which are in internal contact under the action of the consolidated radial force $N = F/l_1$.

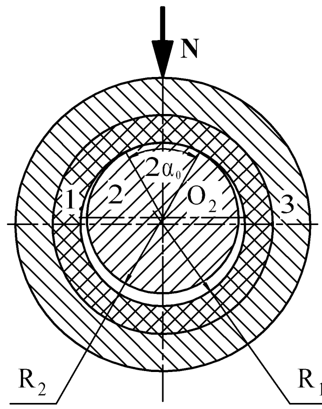


Figure 2. Calculated scheme of linear cylindrical plain bearing

3. CALCULATION METHOD OF CONTACT PRESSURES AND RESOURCE

In accordance with [17, 18, 25] an integro-differential equation of a certain type is used to determine the magnitude and distribution of the initial contact pres– sures $p(\alpha)$ in the investigated conjugation of cylind– rical bodies of close radii. For its approximate solution by the collocation method, the function $p(\alpha)$ is taken in the form

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

where $\alpha = \pm 0.5\alpha_0$ - the collocation points; $E_0 = \left(e_4 / R\right)\cos^2\left(\alpha_0 / 4\right)$ - the collocation coefficient; $Z = \left(1+\kappa_1\right)\left(1+\mu_1\right)E_2 + \left(1+\kappa_2\right)\left(1+\mu_2\right)E_1$, $e_4 =$ $e_4 = 4E_1E_2 / Z$,; E - Young's modulus; $\kappa = 3 - 4\mu$ - the plane strain state; μ - Poisson's ratio.

The contact zone $2R_2\alpha_0$ of the guide elements is set if you know the contact semiangle α_0, which is found at a given value of the load N according to the following equation of equilibrium of forces applied to the base 2:

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

$$0 < \alpha_0 < 90° \qquad (2)$$

When $\alpha = 0$, there is a maximum initial contact pressure $p(0)$, which characterizes the load carrying capacity of the guide

$$p(0) = E_0\varepsilon \tan(\alpha_0/2). \qquad (3)$$

When the polymer bushing wears out, the contact angle $2\alpha_0$ will increase and the initial contact pressures $p(0)$ will decrease. Maximum contact pressure $p(0, t, h)$ in the guide during wear

$$p(0,t,h) = p(0) - p(0,h). \qquad (4)$$

The change $p(0, h)$ of the maximum initial contact pressure due to wear of the polymer bushing is calculated by the formula [18, 20, 22, 25]

$$p(0,h) = E_0 C_h \varepsilon_h \tan\left(\frac{\alpha_{0h}}{2}\right). \qquad (5)$$

where $\varepsilon_h = h_{1*}(K_1 + h_1')$; $C_h -$ the wear rate indicator; $h_1' = \frac{h_2}{h_1} = \frac{B_1\tau_{01}^{m_1}(\tau-\tau_{02})^{m_2}}{B_2\tau_{02}^{m_2}(\tau-\tau_{01})^{m_1}}K_2$ [20], $\tau = fp(0)$ - maximum Coulomb specific friction force; h_{1*} - allowable wear of the bushing;

B, m, τ_0 - characteristics of wear resistance of mate–rials in a tribopair, determined by the least squares method based on the results of experimental studies of wear at different levels of specific friction forces τ [20, 22, 25]; $K_1 = 1$, $K_2 \langle 1$ - coefficients of mutual overlap of the guide elements during moving contact.

An equation of the following type is used to establish the contact semiangle α_{0h} at wear

$$N = 4\pi R_2 E_0(\varepsilon + C_h\varepsilon_h)\sin^2\frac{\alpha_{0h}}{4}, \qquad (6)$$

The resource of the cylindrical sliding guide is defined in the form of the limiting path of sliding friction L_{1*} of the composite bushing

$$L_{1*} = \frac{-B_1\tau_{01}^{m_1}}{C_h\tau_{0h}(1+h_1')(1-m_1)K_1} * \qquad (7)$$
$$*\left\{(\tau-\tau_{01})^{1-m_1} - \left[(\tau-\tau_{01}) - h_{1*}(1+h_1')C_h\tau_{0h}\right]^{1-m_1}\right\}.$$

where $L_{1*} = vt_*$, t_* - service life of the guide in hours; $\tau_{0h} = fp(0,h)$.

4. NUMERICAL SOLUTION, RESULTS AND DISCUSSION

Data for calculation: F = 500, 750, 1000, 2000 N; $N = F/l_1$ = 5, 7.5, 10, 20 N/mm; ε = 0.05, 0.075, 0.1 mm (clearance fit H9/d9); D_2 = 40, 50 mm; l_1 = 100 mm - bushing length, l_2 = 500 mm - base length; $K_2 = 0.2$, $K_1 = 1$; f = 0.09 – boundary friction; h_{1*} = 0.5 mm. Materials of elements of a cylindrical guide:

slider bushing: polymer composites Moglice and DK6(2) Moglice – E_M = 11200 MPa, μ_M = 0.4, σ_c = 120 MPa, DK6 – E_{DK} = 6500 MPa, μ_{DK} = 0.4, σ_c = 140 MPa; $B_1 = 1.2\cdot 10^{11}$, m_1 = 1.9, τ_{01} = 0.05 MPa – their wear resistance characteristics;

base: carbon steel S45 - E_2 = 210000 MPa, μ_2 = 0.3; $B_2 = 2.2\cdot 10^{12}$, m_2 = 2.1, τ_{02} = 0.1 MPa.

Moglice is the anti-friction polymer epoxy-based material of German company «Diamant», which is used for sliding surfaces renovation. Material DK6 is created on the basis of an epoxy matrix with various fillers and contains fillers, in particular molybdenum disulfide, graphite.

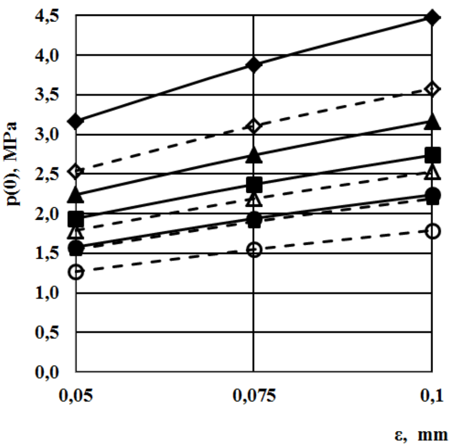
According to the above computational methods, the calculation of the maximum contact pressures $p(0)$ and their transformation $p(0, t, h)$ due to wear was performed; angles α_0 of the initial contact and their magnitude α_{0h} during wear of the slider, and service life of the guide L_1. Their results are given in Fig. 3 – 7. Figures (a) correspond to Moglace composite, and figures (b) – DK6 composite.

Fig. 3 shows the quantitative and qualitative dependencies of $p(0)$ on radial clearance ε and at different loads N on the guide. With a smaller diameter of the bushing with radial clearance increasing, the pressures $p(0)$ increase almost linearly, and with a larger diameter, this dependence becomes close to linear. Quantitative analysis of the obtained results of the calculation shows that: increasing the radial clearance ε twice leads to an increase in $\sqrt{2}$ times the pressures $p(0)$ regardless of the change in the load N and the base diameter D_2; a 4-fold increase in the load leads to an increase in 2 times the maximum contact pressures $p(0)$ regardless of changes in the values of the radial clearance, the base diameter D_2 and the type of composite material. In the guide with the Moglice bushing, the Young's modulus which is 1.3 bigger than DK6, there are pressures more than $\approx\sqrt{E_M/E_{DK}}$ times.

Fig. 4 represents the change in the maximum contact pressure $p(0)$ when the bushing wears. The nature of its change will be nonlinear with increasing radial clearance.

The qualitative decrease of $p(0)$ will be 3.43…3.49 times depending on the load, radial clearance and diameter. With increasing load N 4 times $p(0, t, h)$ will also double, as will the pressures $p(0)$ (Fig. 3).

Fig. 5 shows the qualitative and quantitative dependences of angle α_0 on the radial clearance and the load.



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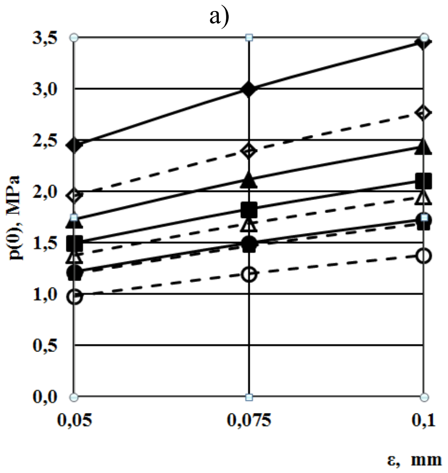

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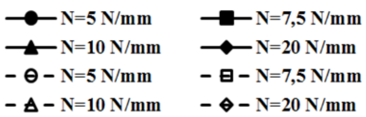
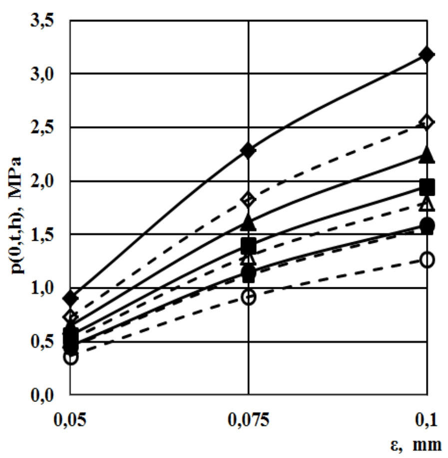
Figure 3. Influence of the radial clearance and load on the maximum contact pressures: D_2 = 40 mm – solid lines, D_2 = 50 mm – dashed lines



It is established that α_0 will decrease nonlinearly with clearance and load increasing. Increasing the radial clearance twice helps to reduce the angle α_0 by $\sqrt{2}$ times at all load values. Accordingly, increasing the load four times gives an increase in angles α_0 twice.

In fig. 6 is shown the nature of the influence of the bushing wear on the angle of the tribocontact.



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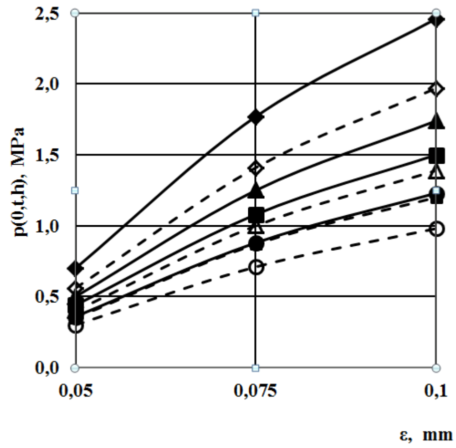

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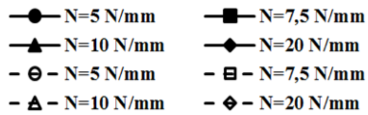
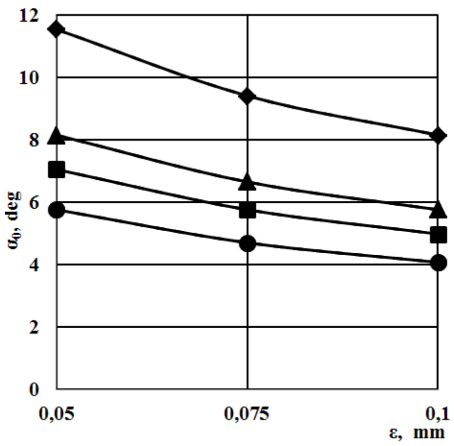
Figure 4. Influence of the wear on the maximum contact pressures





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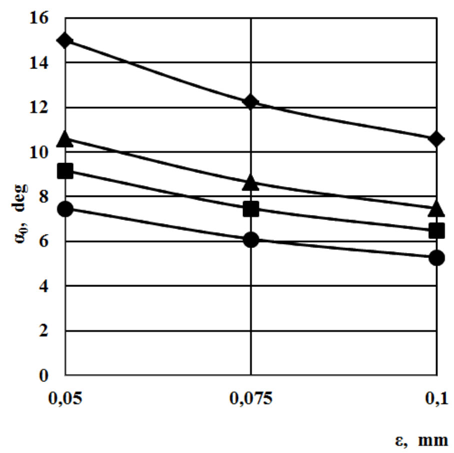

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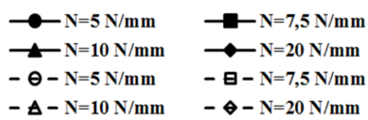
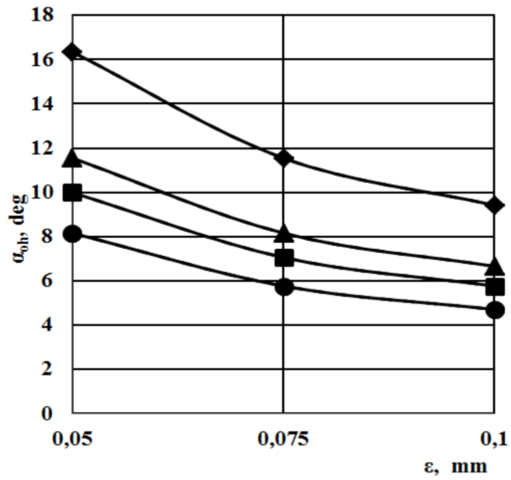
Figure 5. Influence of the radial clearance and the load on the contact angles





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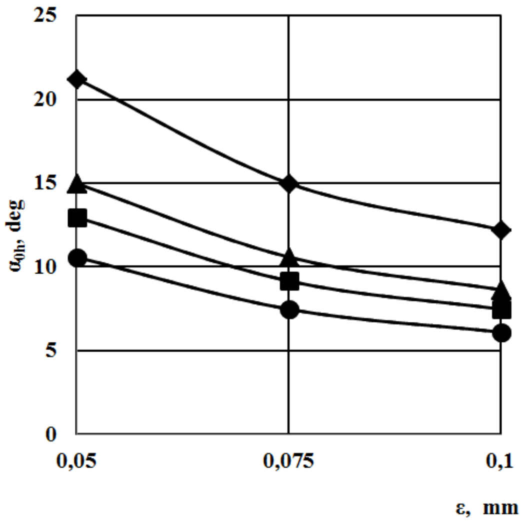

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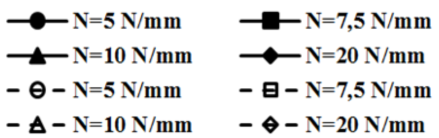
Figure 6. Influence of wear on the angles of the tribocontact

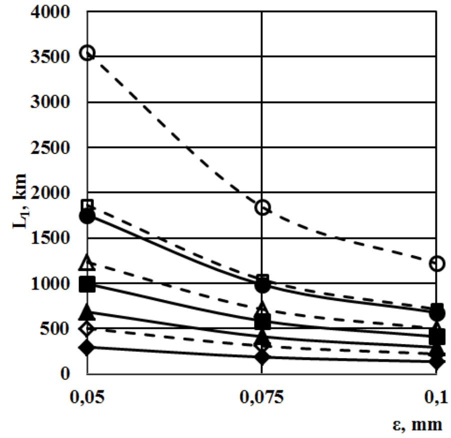


It is also observed a nonlinear nature of this dependence, the largest at the maximum accepted load. Increasing the clearance twice leads to a decrease in the angle α_{0h} by 1.73 times, which is approximately equal to E_M/E_{DK} . Instead, when the load is quadrupled, the angle α_{0h} will be twice as large.

Fig. 7 shows the qualitative and quantitative impact of the radial clearance and the load on the resource of the guide L_1 until the bushing 1 reaches the allowable wear $h_{1*} = 0{,}5$ mm.

The increase in the radial clearance causes a significant decrease in the resource of the guide due to the increase in maximum contact pressures. As the load increases, this reduction in resource will be more significant. Reducing the diameter also significantly reduces the durability (Fig. 7).

Doubling the radial clearance causes the sliding distance to decrease by 2.19 … 2.91 times (Moglice) and 2.40 …3.40 times (DK6) depending on the load and the base diameter (Table 1; 2).



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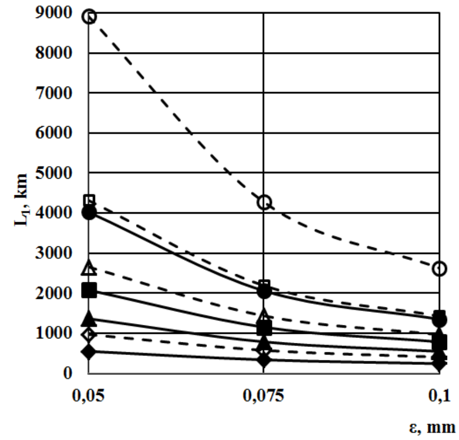

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Figure 7. Dependence of the guide service life on the radial clearance

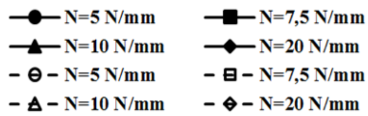

Table 1. Guide resource $L_{1(M)}$ for a bushing made of Moglice

ε, mm	Sliding distance L_1, km							
0,05	1756	996	685	296	3548	1863	1234	502
0,075	987	586	413	186	1845	1042	715	308
0,1	677	412	294	135	1219	713	498	221
N, N/mm	5	7,5	10	20	5	7,5	10	20
Increase ε, times	D_2 =40 mm				D_2 = 50 mm			
	Decrease L_1, times							
2	2,59	2,42	2,33	2,19	2,91	2,61	2,47	2,27

Table 2. Guide resource $L_{1(DK)}$ for a bushing made of DK6

ε, mm	Sliding distance L_1, km							
0,05	4026	2078	1364	548	8918	4320	2655	968
0,075	2060	1150	785	335	4274	2190	1432	573
0,1	1350	783	545	240	2624	1429	963	403
N, N/mm	5	7,5	10	20	5	7,5	10	20
Increase ε, times	D_2 =40 mm				D_2 = 50 mm			
	Decrease L_1, times							
2	2,59	2,42	2,33	2,19	2,91	2,61	2,47	2,27

Table 3. The impact of the radial clearance on the relative reduction of $L_{1(DK)}$ / $L_{1(M)}$

Composite	ε, mm	Sliding distance L_1, km							
DK6	0,05	4026	2078	1364	548	8918	4320	2655	968
Moglice	0,05	1756	996	685	296	3548	1863	1234	502
$L_{1(DK)}$ / $L_{1(M)}$, times		2,29	2,09	1,99	1,85	2,51	2,32	2,15	1,93
DK6	0,075	2060	1150	785	335	4274	2190	1432	573
Moglice	0,075	98 7	586	413	186	1845	1042	715	308
$L_{1(DK)}$ / $L_{1(M)}$, times		2,09	1,96	1,90	1,80	2,32	2,10	2,00	1,86
DK6	0,1	1350	783	545	240	2624	1429	963	403
Moglice	0,1	677	412	294	135	1219	713	498	221
$L_{1(DK)}$ / $L_{1(M)}$, times		1,99	1,90	1,85	1,78	2,15	2,00	1,93	1,82
Load N, N/ mm		5	7,5	10	20	5	7,5	10	20
Diameter D_2, mm		40				50			

The resource of the guide bushing made of DK6 will be 1,77 … 2,51 times greater than the resource of the guide bushing made of Moglice (Table 3) depending on the radial clearance in the joint and the base diameter.

5. CONCLUSION

The use of hybrid pairs of materials with significantly different strength, elastic characteristics, deformability and wear resistance in linear plain bearings requires the creation of appropriate calculation methods by which at the design stage it would be possible to assess contact pressures, wear and service life. Due to their absence, the authors conducted research aimed at developing an appropriate method for solving the contact problem of the theory of elasticity with wear of a hybrid tribosystem according to the author's tribokinetic model of material wear kinetics.

Using polymer composites Moglice and DK6, the influence of the load, the bushing diameter and the radial clearance on the quantitative and qualitative patterns of change of contact parameters (initial and while the bushing wears) and the resource of this type of linear bearings from the specified factors was investigated. The above calculation results indicate the effectiveness of the presented method for estimating the contact parameters in engineering practice in the design of metal-polymer cylindrical linear bearings.

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ПРЕДВИЂАЊЕ КОНТАКТНОГ ПРИТИСКА И РЕСУРСА МЕТАЛНО-ПОЛИМЕРНИХ ЛИНЕАРНИХ ЦИЛИНДРИЧНИХ КЛИЗНИХ ЛЕЖАЈЕВА

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Различити типови линеарних лежајева (вођица) имају широку примену код опреме као и у многим другим областима људске активности. Ово се посебно односи на цилиндричне вођице (линеарне клизне лежајеве) са наизменичним кретањем. И поред широке примене не постоје адекватне методе за прорачун хабања и дужину радног века не само метала, већ и метално-полимерних линеарних лежајева. Према ауторовом методу извршено је истраживање утицаја оптерећења, основног дијаметра и радијалног зазора на максимални почетни притисак код овог типа лежаја у оквиру проблема површинског контакта у теорији еластицитета. Применом методе прорачуна, према ауторовом трибокинетичком моделу хабања у току трења, измерен је утицај хабања композитиних чаура на промену почетних карактеристика контакта (контактни притисак и контактни угао). Такође је прорачуном, на основу наведених фактора, предвиђена дужина трајања радног века лежаја. Утврђене су квантитативне и квалитативне правилности код зависности контактног притиска и ресурса од прихваћених фактора.