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ANALYTICAL STUDY OF DURABILITY AND CONTACT CHARACTERISTICS OF METAL-POLYMER COMPOSITE SLIDING BEARINGS FOR MEANS OF TRANSPORT

Summary. The field of application of metal-polymer bearings is vast: transport of various types, processing industry equipment, medical equipment, various types of maintenance equipment, etc. With the use of the developed generalized author's analytical method of metal-polymer plain bearings on the basis of which it is laid the author's research methodology of materials wear kinetics at sliding friction (dry, lubricated), calculation of their durability is carried out. Contact parameters are also determined. Metal-polymer bearings with a bushing made of PA6, PA66 polyamide and PA6 based composites filled with glass (PA6 + 30GF) and carbon (PA6 + 30CF) dispersed fibres, molybdenum disulphide (PA6 + MoS_2) and oil filled cast polyamide (PA6 + oil) are considered. These polyamides (unfilled and filled), as self-lubricating materials, are widely used in this type of dry friction sliding bearings. The predictive estimation of durability of investigated bearings depending on loading, polymer materials Young's modulus, their wear resistance and sliding friction coefficient is executed. Regularities of change from the specified factors of bearing's durability and the maximum contact

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pressures are established. Experimental indicators, diagrams, and wear resistance characteristics of the specified polymeric materials are presented. The results of researches of sliding friction coefficient and modulus of elasticity for carbon steel (0.45%C) – polyamides tribocouples are presented.

Keywords: metal-polymer sliding bearings, analytical method, polyamides, PA6, PA66, composites PA6 + 30GF, PA6 + 30CF, PA6 + MoS_2, PA6 + oil, Young's modulus, wear resistance, coefficient of friction, durability, contact parameter

1. INTRODUCTION

Metal-polymer plain bearings (MP) are used in a variety of machines, equipments, and devices in automotive, mechanical engineering, instrumentation, aerospace, rocket and space technologies, etc.; in all types of processing industry – food, textile, pharmaceuticals, chemical, etc. They are commonly found in household appliances, computer, office, medical, measuring equipment, and other related fields. They operate in water, gas, liquid, boundary and dry friction. The self-lubricating MP bearings offer numerous advantages in comparison to rolling bearings, including simplicity or lack of maintenance, high manufacturability, a wide range of shaft diameters (ranging from micro to large), small cross-sectional dimensions, significant shaft speed, ability to operate under dynamic and shock loads, high damping capacity, low noise generation, and the ability to operate in polluted technological environments or water, at elevated and lowered temperatures, in vacuum, and other characteristics.

The reliability of MP bearings is particularly important because many technical devices are forced to operate without the use of lubricants due to various reasons and production conditions. In fact, such conditions may be provided for in the operation of different types of vehicles or may arise under different circumstances.

Given that dry friction causes significantly more intense wear compared to the boundary friction, for MP bearings it is very relevant and necessary to calculate their durability and load carrying capacity in the design. However, the corresponding effective methods of calculation of MP bearings in the literature are virtually absent. Developed calculation [1-8] or numerical methods [9-11] of the study of metal plain bearings have not found practical application for calculating the durability and load carrying capacity of MP bearings. In the existing studies of MP sliding bearings [12-14], the authors use the well-known linear Archard`s law of abrasion / adhesive wear. However, even in MP dry friction bearings, not to mention metal bearings, where dry friction is almost non-existent, the mechanism of abrasive wear is unlikely. Regarding the presence of a dominant mechanism of adhesive wear in metal-polymer couples, [15] states that the wear of polymers at sliding friction is a combination of three mechanisms (fatigue, adhesive, and sometimes abrasive). Which of them will be the main and which are concomitant depends on the polymer properties, the friction conditions and the course of wear. Under certain conditions, the polymeric material is transferred to the metal counterpart, forming a thin adhesive film, which is a sign of adhesive wear as the main process [16-18]. However, in [16] it is noted that at friction coefficients greater than 0.3, characteristic of MP dry friction pairs, surface fracture (wear) of polymers occurs by the mechanism of surface fatigue under the influence of shear stresses (specific friction forces). Adhesion wear can also partially occur as a minor concomitant process. Considering this, the author's methodology for studying the materials wear kinetics during sliding friction [19-22] and, accordingly, the calculation method for studying the wear of metal [21-25] and MP bearings [26, 27] assumes a fatigue mechanism, and takes into account changes in conditions of contact interaction due to wear.

MP bearing bushings are made of various polymers (polyamides – PA, polyacetal – POM, polytetrafluoroethylene – PTFE, polyester ketones – PEEK, ultra-high molecular weight polyethylene – UHMWPE, etc.) and filled composites based on them [11]. Materials (glass and carbon fibres, molybdenum disulphide, graphite, bronze powder, polytetrafluoroethylene, etc.) with different volumetric contents are used as fillers in the polymer matrix, the function of which is increasing the wear resistance and durability of bearings. In particular, a widely used polymer for MP bearings is polyamide PA66 and partially PA6. Polyamide based composite materials are much more widely used, in particular PA6 filled with fibreglass (PA6 + 30GF), carbon fibre (PA6 + 30CF), molybdenum disulphide (PA6 + MoS_2), oil filled cast polyamide (PA6 + oil).

The literature presents a number of results of tribological experimental studies of different polymeric materials used in MP plain bearings [28-36]. In particular, in [28] the wear and friction coefficient of PA6 based polyamide composites coupled with AISI 02 steel (0.90% C, 1.6% Mn) were studied according to the pin-on-disk scheme during dry friction. In [29], the tribological behaviour of various bearings with polymer bushings coupled with steel shaft during dry friction was studied: polyacetal (POM-Delrin), cast polyamide PA6, and others. The work [30] is devoted to the study and evaluation of wear resistance of bearings based on PE (polyethylene), PA, POM, PTFE and Bakelite. Study [31] establishes the effect of sliding speeds on friction and wear in bearings made of composites PA66, PA66 + 18PTFE and PA66 + 20GF + 25PTFE at room temperature. Tribological analysis of the dry friction behaviour of cast nylon (PA6) for the connecting rod plain bearing was performed in [32]. G. Kalácska presents in [33] the results of pin-on-disk wear testing of twenty-one engineering polymers, both basic and composites. In [34], the patterns of friction and wear of PA6 and ABS (acrylonitrile butadiene styrene) sliding on metal under dry friction conditions were studied on a pin-on-disc tribometer. [35] is devoted to the study of the tribological properties of PA6 polyamide in dry friction at different speeds and loads. In [36], the coefficient of friction, friction force and temperature of six polymer composites were investigated according to the ball-on-disk scheme under dry friction: cast polyamide PA6G + oil, PA6G + MoS_2, polyacetal POM + Al, polyterephthalate ethylene PET + PTFE, PTFE + Bronze, PTFE + Graphite. Although the results of these studies are quite extensive, it is not possible to establish the wear resistance characteristics of polymers in order to calculate the durability of plain bearings.

The presented article is the result of the investigation conducted by the author's calculation method for metal-polymeric plain bearings.

- durability of the MP bearing at allowable wear of the polymeric bushing;
- initial maximum contact pressures;
- initial contact angle;
- wear resistance of basic polyamides PA6, PA66;
- wear resistance of filled composites PA6 + 30GF, PA6 + 30CF, PA6 + MoS_2, PA6 + oil;
- influence of fillers on the wear resistance of the studied polymeric materials;
- effect of contact pressure on the coefficient of sliding friction.

2. METHODS OF CALCULATION OF METAL-POLYMER BEARINGS

The plane contact problem of the linear theory of elasticity is studied. In the plain bearing (Fig. 1) the shaft 2, which is under the action of the reduced force $F = N / l,$ rotates at angular velocity ω = const. Under the influence of full external static load $N = \text{const}$, the shaft journal 1 and the bushing 2 located in the housing 3 are contacted in the area $2R_2\alpha_0$ on which act unknown in magnitude contact pressures p_α. A radial clearance $\varepsilon = R_1 - R_2 > 0$ will be guaranteed between the outer radius of the steel shaft R_2 and the inner radius of the polymer bushing R_1. The elements materials of MP bearing have significantly different strength, elastic characteristics, wear resistance (strength characteristics of 8… 10 times, Young's modulus of 50… 200 times, wear resistance by 2… 3 orders of magnitude). This fundamentally affects their contact strength, bearing capacity, contact parameters and, of course, recourse. When rotating, the shaft 2 wears along the contour $2\pi R_2$ and the bushing 1 - in the area $2\alpha_0 R_2$.

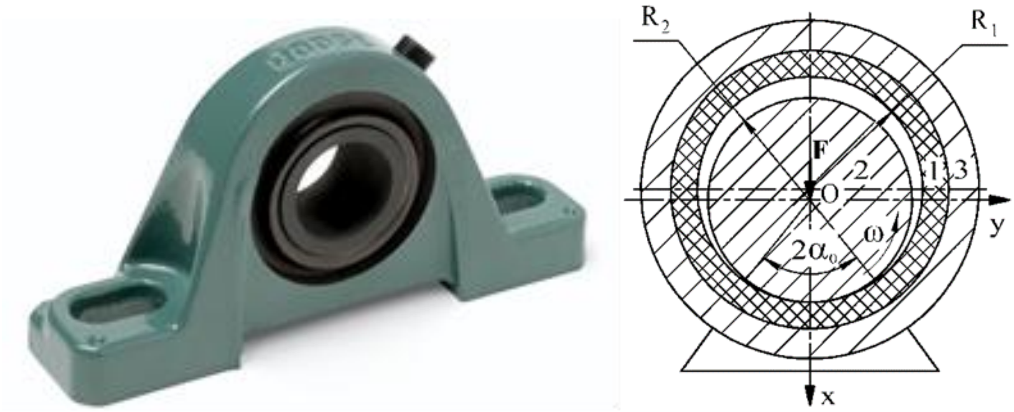

Fig. 1. Metal-polymer plain bearing: general view and scheme

According to the tribokinetic mathematical model of wear at sliding friction [20, 21] and the method of calculation of plain bearings [22-25] for the forecast estimation of durability at the given bushing wear $h_1 = h_{k\max}$ the following expression is used [20, 21]:

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

where B_k, m_k, τ_{k0} - wear resistance characteristics of tribocouple materials, which are established by the results of model triboexperimental studies [20, 21]; k is shaft or bushing index (Fig. 1); $v = \omega R_2$ - sliding speed; c_h is the wear rate coefficient; $\Sigma_1 = (-K_t^{(1)} + h_1'),\ \Sigma_2 = \left(K_t^{(2)} - h_2'\right);\ K_t^{(1)} = 1, K_t^{(2)} = \alpha_0 / \pi$ - respectively, the coefficients of mutual overlap of the bushing and the shaft; $h_1' = h_2 / h_1$, $h_2' = h_1 / h_2$ - relative wear in the tribosystem.

$$h_1' = \frac{B_1\tau_{10}^{m_1}\left(\tau-\tau_{20}\right)^{m_2}}{B_2\tau_{20}^{m_2}\left(\tau-\tau_{10}\right)^{m_1}}K_t^{(2)},$$
$$h_2' = \frac{B_2\tau_{20}^{m_2}\left(\tau-\tau_{10}\right)^{m_1}}{B_1\tau_{10}^{m_1}\left(\tau-\tau_{20}\right)^{m_2}}K_t^{(1)}, \tag{2}$$

The specific force of friction is determined by the Amontons-Coulomb law.

$$\tau = fp_0 \tag{3}$$

where f is the coefficient of sliding friction.

The initial maximum contact pressure p_0 will occur along the line of force F and is determined by the formula [20, 21]:

$$p_0 \approx E_0\varepsilon\tan\left(\alpha_0/2\right) \tag{4}$$

where $E_0 = e\cos^2\left(\alpha_0/4\right)/R_2$, $e = 4E_1E_2/Z$, $0\langle\alpha_0\langle 90^o$, $Z = (1+\kappa_1)(1+\nu_1)E_2 + (1+\kappa_2)(1+\nu_2)E_1$, $\kappa = 3-4\nu$; E_k, ν_k – Young's moduli and Poisson's ratios of the materials of the shaft 1 and bushing 2.

As a result of wear of the composite bushing, the initial maximum contact pressures p_0 will decrease, i.e. in the tribocontact there will be wear contact pressures $p_{0h}\langle p_0$, which are determined as follows:

$$p_{0h} = p_0 - \left|p_h\right| \tag{5}$$

where p_h – change in initial pressures due to wear.

To find them, we used the dependence given in [20, 21]:

$$p_h \approx E_h\varepsilon_h\tan\left(\alpha_{0h}/2\right) \tag{6}$$

where $E_h = c_h(e/R_2)\cos^2(\alpha_{0h}/4)$; c_h >0 – wear rate indicator; $\varepsilon_h = h_{1\max}\Sigma_1$.

The initial specific friction force $\tau = fp_0$ during the polymer bushing wear will decrease.

$$\tau_h = fp_{0h} = fE_h\tan(\alpha_{0h}/2), \tag{7}$$

where $\alpha_{0h}\rangle\alpha_0$ is the unknown contact semiangle occurring due to bushing wear.

In the studied type of contact problem, the initial contact angle $2\alpha_0$ is also unknown. Determination of the initial contact semiangle α_0 is carried out under the equilibrium condition of forces acting on the shaft 2.

$$F = R_2 \int_{-\alpha_0}^{\alpha_0} p_\alpha \cos\alpha d\alpha = 4\pi R_2 E_0 \varepsilon \sin^2(\alpha_0 / 4). \tag{8}$$

Taking into account the expression for the coefficient E_0, it is expressed as:

$$\alpha_0 = 2arc\sin\sqrt{F / \pi e\varepsilon} \tag{9}$$

To determine the tribocontact semiangle α_{0h}, which outlines the contact area during wear, the following condition is used:

$$\begin{aligned} F &= 4\pi R_2 E_0 (\varepsilon + c_{\alpha h}\varepsilon_h)\sin^2(\alpha_{0h} / 4) = \\ &= 4\pi e \cos^2(\alpha_0 / 4)(\varepsilon + c_{\alpha h}\varepsilon_h)\sin^2(\alpha_{0h} / 4) \end{aligned} \tag{10}$$

3. MATERIALS, TRIBOEXPERIMENT

To calculate the MP bearing durability according to (1) should be determined the wear resistance characteristics B_k, m_k, τ_{k0} of materials in tribocouple steel-polymer. The author's method of model research of materials wear [20, 21] involves several levels of contact pressure in the tribosystem and provides determination at the same time both wear (mass or linear) of samples, and specific force of friction τ. Model triboexperiments are performed according to the pin-on-disk friction scheme, which ensures the constancy of the initial external conditions during the study, namely the contact pressures and sliding speed, the contact temperature of the experimental tribocouple. This approach to triboexperimental research significantly expands the information on the wear resistance of materials in a certain range of contact pressures, in contrast to the standards ISO 7148-2 and ASTM G99, which provides to conduct experimental studies at one value of contact pressure. In the author's methodology of studying the material's wear kinetics, it is accepted that the level of τ unambiguously determines their wear rate. It should be noted that it is the coefficient of sliding friction f determines the level of t in the tribocontact. That is, at the same level of contact pressure p_0, its value can vary several times depending on the type of friction (dry, semi-dry friction, boundary).

The material's wear resistance characteristics B_k, m_k, τ_{k0} in the tribocouple are determined by the results of model triboexperimental studies as invariant wear resistance parameters in the selected range $\tau = fp_0$. In the following, they are necessary to calculate the durability or wear of plain bearings by the above method.

According to the method of model triboexperiments, the calculation of experimental indicators of wear resistance (wear resistance functions) Φ_i of polymeric materials for each level of contact pressure p_i and correspondingly specific friction force τ was carried out by using the formula:

$$\Phi_i = L_i / h_i \tag{11}$$

where L is the path of friction, h_i is the linear wear of samples at the i- th level of pressures p_i.

Research program: sliding speed v = 0.4 m/s; contact pressure p = 2, 4, 6, 8 MPa; friction path L = 5000… 10000 m; sample diameter d = 3 mm. Forced cooling of the friction unit was carried out to ensure the temperature of the end surface of the polymer sample T = 23 ± 1°C at a relative humidity of 50 ± 5% (standard ISO 7148-2).

For approximation of the wear resistance experimental indicators Φ_i the next type of ratio was used [20, 21]:

$$\Phi_k(\tau) = B_k \frac{\tau_{k0}^{m_k}}{(\tau - \tau_{k0})^{m_k}} \tag{12}$$

The wear resistance characteristics of tribocouple materials are determined by using the method of the least squares, where τ_0 is the limiting value of the specific friction force at which wear at the macro level will be virtually absent.

According to the wear resistance characteristics determined in this way, graphical indicators (diagrams) of wear resistance are plotted for each polymer (Fig. 2).

4. RESULTS, DISCUSSION

The source data for calculation of durability and parameters of contact at dry friction: N = 5000, 3000 N; D_2 = 50 mm; $l = D_2$; $\varepsilon = 0.2$ mm; ω = 6.28 rad / sek; h_{1*} = 0.5 mm.

Metal-polymer bearings from the following materials of the elements are investigated:

Shaft 2 – carbon steel 0.45%C, normalized, roughness R_a = 0.8-1 µm; E_2 = 210 GPa, ν_2 = 0.3; $B_2 = 10^{13}$, $m_2 = 2$, τ_{20} = 0.1 MPa;

polymeric bushing 1 (Table 1).

Tab. 1

Parameters of bushing materials

Wear resistance characteristics	Thermoplastic polymeric materials					
	PA6	PA66	PA6+ 30GF	PA6+ MoS_2	PA6+ 30CF	PA6+oil
$B_1 \cdot 10^{10}$	2.26	3.37	4.12	5.58	6.53	7.03
m_1	1.09	1.09	1.09	1.1	1.1	1.1
τ_{10}, MPa	0.05	0.05	0.05	0.05	0.05	0.05
Young's module E_1, GPa	2.0	2.3	2.7	1.66	3.3	1.96
Poisson's ratio ν_1	0.4	0.4	0.41	0.4	0.41	0.4

Note: for PA6 + 30GF the volume content of filler (fibreglass) is 30%.

The results of triboexperimental studies and numerical solution are presented in Fig. 2-9. Fig. 2 presents diagrams of wear resistance of the above polymeric materials, and Fig. 3 presents their relative wear resistance.

The figure shows the determined experimental functions of the wear resistance of polyamides Φ_i with different markers at four values of the specific force of friction $\tau_i = fp_i$ in the tribocontact. Their location along the τ axis at the same contact pressures depends on the coefficient of friction in the studied tribocouple. However, the qualitative patterns of changes in wear resistance in the form of wear resistance diagrams are practically the same for these polymer materials. These graphic indicators of wear resistance - wear resistance diagrams of polyamides are obtained by approximation of Φ_i according to (12). They make it possible to conduct a visual comparative assessment of the wear resistance of these polyamides at a specific value of τ. From the analysis of Fig. 2 it is noticeable which of the polyamides has low wear resistance (PA6), and which has the highest wear resistance (PA6 + oil). It should be noted that the wear resistance of polyamides depends non-linearly on the effective specific force of friction.

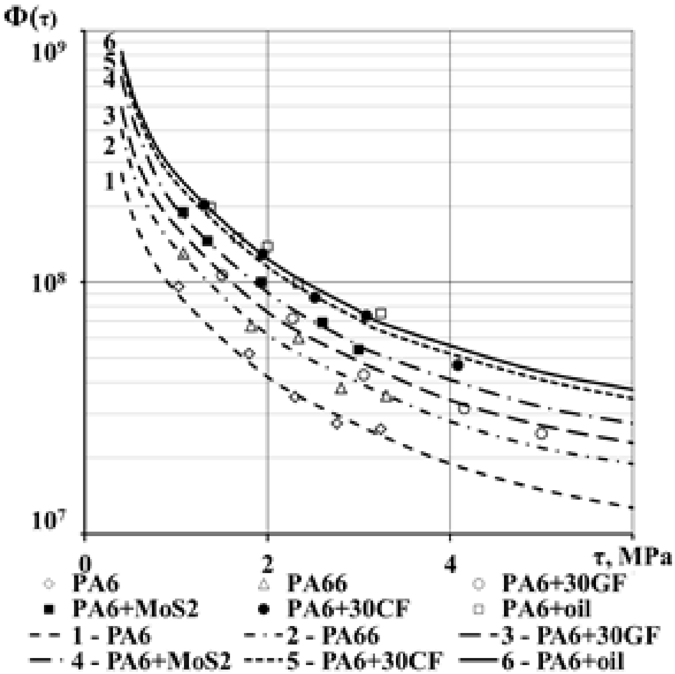

Fig. 2. Diagrams of polyamides wear resistance

For example, the ranking according to the criterion of wear resistance of the studied polyamides at 2 MPa relative to the least resistant polyamide PA6 at $\tau = 2$ MPa is shown in Fig. 3.

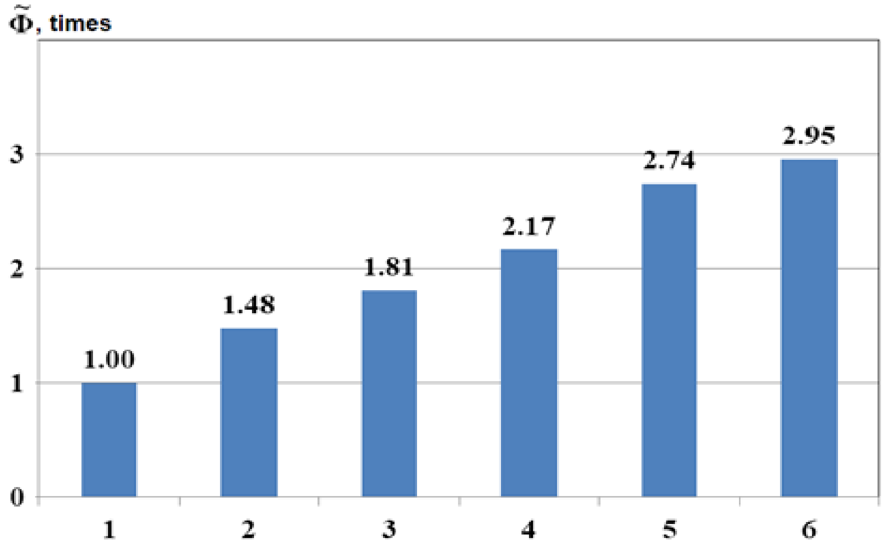

Fig. 3. Relative wear resistance of polyamides
1 - PA6, 2 - PA66, 3 - PA6 + 30GF, 4 - PA6 + MoS_2, 5 - PA6 + 30CF, 6 - PA6 + oil

Since the wear resistance index m (Table 1) of all studied polyamides is almost the same, in the whole range of changes in τ their relative wear resistance $\tilde{\Phi}$ will be almost the same as shown in Fig. 3. Increasing the durability of polyamides is close to linear.

According to the given data, the calculation of durability of MP bearings with various materials of bushings (Fig. 4) is carried out.

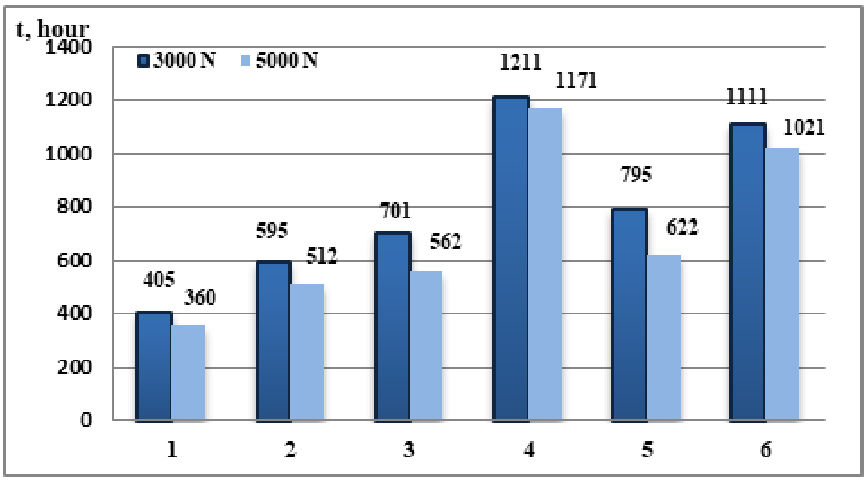

Fig. 4. Durability of MP bearings

There is no increase in the durability in the order of increasing their wear resistance (Fig. 2) from polymer 1 to polymer 6, although the wear resistance characteristic B is increasing (Table 1). This type of sequence does not exist either in the coefficients of sliding friction (Fig. 5 and 6) or in the value of the Young's moduli (Table 1, Fig. 9). As a result, the MP bearing with a polymer sleeve made of a more durable material (PA6 + oil) will not have the highest durability. The highest durability has a composite PA6 + MoS_2. It depends on the complex influence of these three characteristics. This, in fact, applies to composites PA6 + 30CF and PA6 + oil.

Fig. 5 shows the relative durability (ranking by durability) of MP bearings relative to polyamide PA6.

If the durability (Fig. 4) at higher loads is lower for all materials, then in the case of relative durability (Fig. 5) there is no such pattern for two polyamide composites - PA6 + MoS_2, PA6 + oil.

As a result of triboexperimental studies, the non-linear nature of the decrease in the friction coefficient with increasing contact pressure was established (Fig. 6).

The highest is the coefficient of friction in the pair PA6 + 30GF - steel, slightly lower - in the pair PA6 + 30CF - steel, and the lowest - in the pair PA6 + MoS_2 - steel.

Accordingly, at contact pressures caused by loads N = 3000, 5000 N, their values are shown in Fig. 7.

By using the presented method, the maximum contact pressures p_0 (Fig. 8) and contact angles $2\alpha_0$ (Fig. 9) were estimated.

The value of the maximum contact pressure p_0 under the same conditions depends only on the value of the modulus of elasticity E of polymeric materials.

As the Young's modulus increases, the rigidity of the polymer material increases, because of which the contact angle $2\alpha_0$ decreases and the maximum contact pressure p_0 increases.

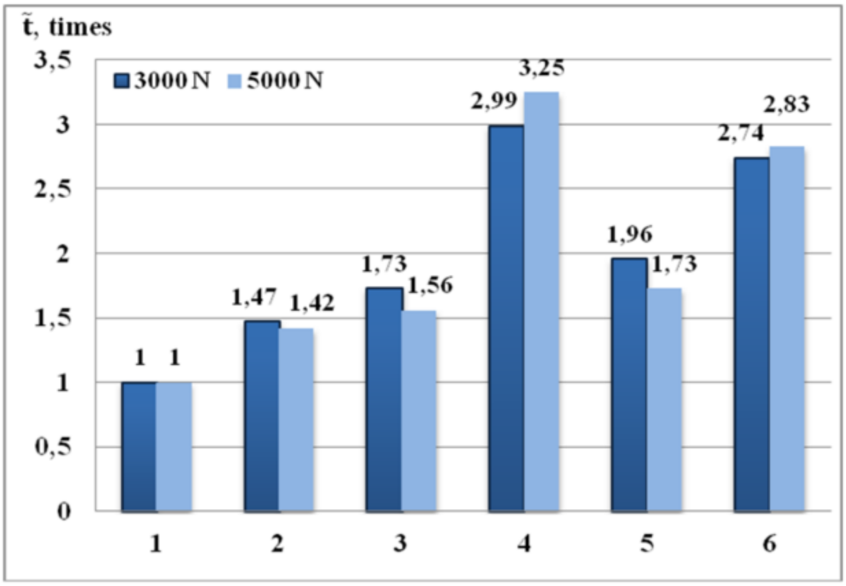

Fig. 5. Relative durability of MP bearings

It is known that in the design calculation of metal plain bearings in engineering practice, the conventional method based on the average pressure p as a criterion for their bearing capacity, calculated according to a simplified method, is used. It is assumed that $p = \text{const}$ when the contact angle $2\alpha_0 \approx 114.6^{\circ}$ = const, i.e.

$$p = \frac{N}{2R_2 l} = \frac{F}{D_2} \leq [p] \tag{13}$$

where $[p]$ is the allowable contact pressure for less durable material.

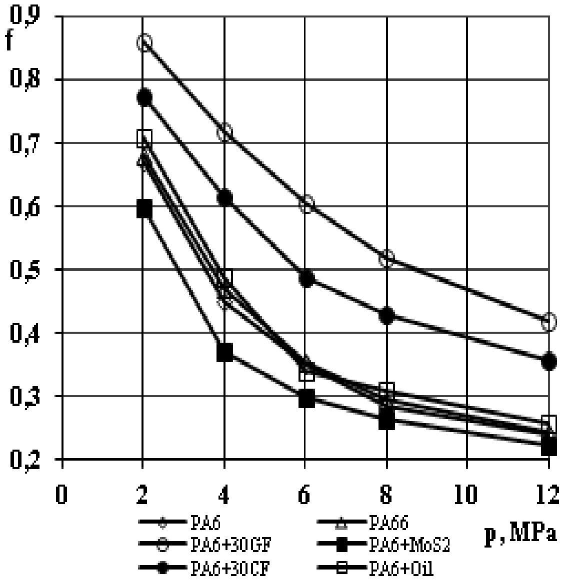

Fig. 6. Experimental dependence of the average coefficient of sliding friction on the pressure in polyamide-steel tribocouples

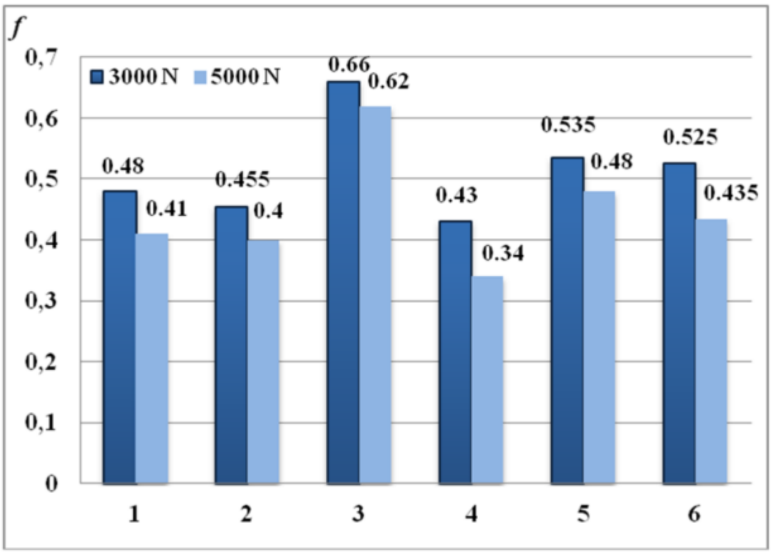

Fig. 7. The effect of load and type of polyamide on the coefficient of sliding friction

A modified formula for maximum contact pressure $p_{\max}$ is given in [37], where the cosine law of its distribution by contact area at angle $2\alpha_0 = 180^{\circ}$ is laid down.

$$p_{\max} = \frac{4}{\pi} \frac{N}{D_2 l} = \frac{4}{\pi} \frac{F}{D_2} \tag{14}$$

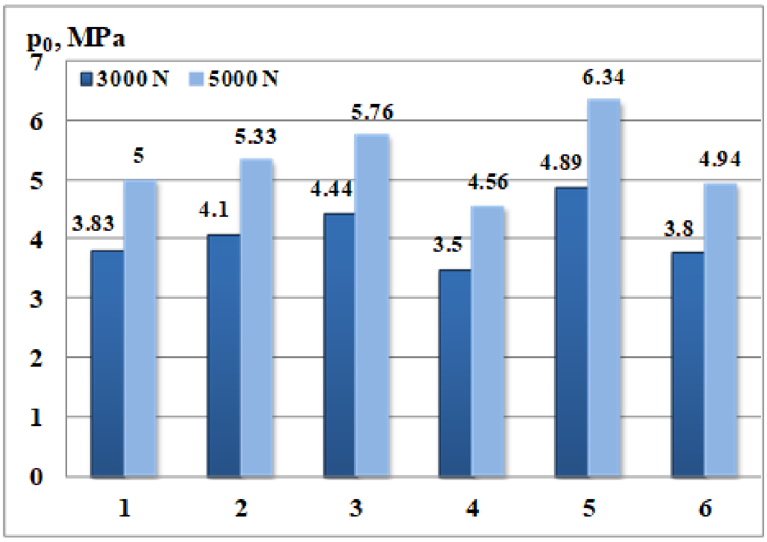

Fig. 8. Maximum contact pressures in the MP bearing with various types of polyamide bushings

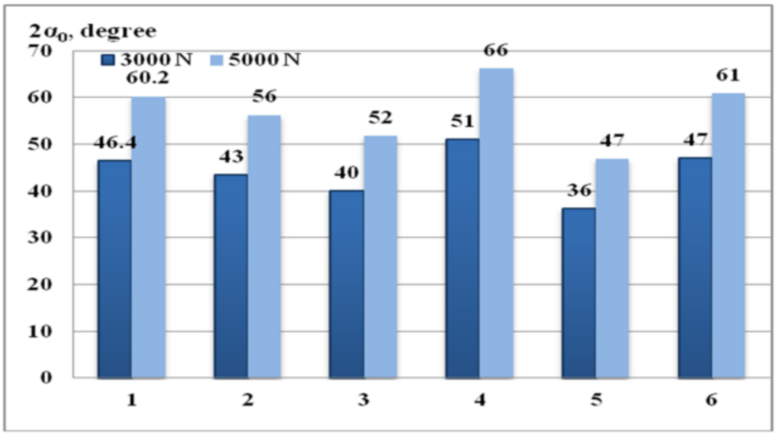

Fig. 9. Contact angles

These formulas do not take into account both the radial clearance $\varepsilon > 0$ and the elastic characteristics E, ν of the materials of bearing elements.

As was shown above, the characteristics of the bearing and materials have a decisive influence on the contact parameters - p_0, $2\alpha_0$. The specified contact conditions of the bearing elements in these methods are inappropriate. There is always a certain radial clearance here, which ensures reliable operation of the bearing. It should also be noted that the contact angle $2\alpha_0 \approx 114.6^{\circ}$ can be achieved only with a significant load on the bearing and small radial clearances, as is known from the literature and from the works of the author [20, 21, 23]. The contact angle $2\alpha_0 = 180^{\circ}$ is not achieved even at zero clearance. In contrast, in MP plain bearings, the use of these techniques is unreasonable given the significant difference between the Young's modulus of the steel shaft and the composite bushing, as mentioned above.

The contact pressures p and p_{max} in the considered MP bearings are estimated: N = 5000 N: p_0 = 4.94 – 6.34 (by Eq. (4)), p = 2.0 MPa (by Eq. (13)), p_{max} = 2.55 MPa (by Eq. (14)); N = 3000 N: p_0 = 3.5 – 4.89 (by Eq. (4)), p = 1.2 MPa (by Eq. (13)), p_{max} = 1.53 MPa (by Eq. (14)). That is, according to the conditional method of calculation of bearing capacity without taking into account the type of polymeric material in the bearing, there will be much lower contact pressures than determined by the above author's method of calculation.

Fig. 10 shows the Young's modulus of unfilled (base) polyamides and polyamide-based composites established by the Oliver - Farr method.

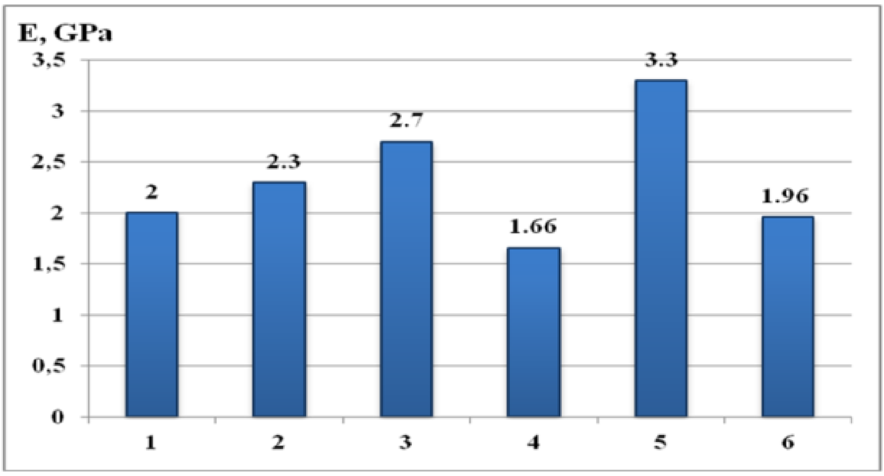

Fig. 10. Modulus of elasticity of polyamides and polyamide composites

Different polyamide fillers PA6 have different effects on the Young's modulus of composites. Fibreglass and carbon fibre increase it, molybdenum disulfide reduces it, and oil has almost no effect on it.

The methods of calculation of durability and bearing capacity of MP bearings presented together with a technique of model triboexperimental research of materials at sliding friction provide an opportunity of their effective research at designing.

Program for calculations was developed in PYTHON.

5. CONCLUSIONS

The main research results are:

1. For polyamides widely used in MP bearings (PA6, PA66, PA6 + 30GF, PA6 + 30CF, PA6 + MoS_2, PA6 + oil) the wear resistance was experimentally investigated according to the pin-on-disk friction scheme (ISO 7148-2) and wear resistance indicators were determined. Wear resistance diagrams were plotted using them. The wear resistance characteristics of polymeric materials as basic parameters of their wear kinetics at dry sliding friction were determined. Due to the modification of polyamide PA6 with different fillers, its wear resistance increased almost threefold (oil filled cast polyamide PA6 + oil).
2. The influence of load, wear resistance and friction coefficient, Young's modulus on the durability of the bearing and its contact parameters was investigated. It was established that the durability of the MP bearing depends on the complex effect of these three characteristics, and not only on the wear resistance of the bushing polymeric material.

It was found that the durability of the bearing with a bushing made of filled composites increases and was three times greater than for the composite PA6 + MoS_2 than PA6.
3. It was established that at the same loads on the bearing, the contact parameters differed markedly depending on the type of polyamide. This was due to their different elastic properties (rigidity), which are characterized by the Young's modulus.
4. Quantitative and qualitative regularities of the influence of contact pressure on the coefficient of sliding friction in the investigated metal – polymer tribocouples were established. The confirmation of the general regularity of its significant decrease upon an increase in contact pressure in an experimental tribocouple was obtained.
5. A typical method of calculating contact pressures gives a significant understatement for MP bearings (in 2.5… 4.1 times) depending on the load.
6. One important property of the developed methods for calculating the durability and bearing capacity of MP bearings was that the solution to the problem is presented in a closed form, which allows its practical implementation to be carried out using simple software tools, starting from Microsoft Excel. Based on these methods, it was also possible to carry out, when designing bearings, an optimization according to the criteria of durability, contact strength, wear, as well as the optimal choice of materials.
7. The results of the research can be used in practice to optimize the choice of polyamides in moving sliding friction assemblies of vehicles in terms of load-bearing capacity, wear resistance, and durability.
8. In particular, polyamides of the studied type were used in the engine connecting rod bearing [32], clutch discs [38], shaft-bushing bearing of an automotive turbocharger wastegate system [39], for the bearing of an automatic car belt tensioner [40].

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