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RESEARCH

Investigation of the Effect of Temperature on the Contact Characteristics and Resource of Metal-Polymer Plain Bearings

M. Chernets[a,*], A. Kornienko[a], S. Fedorchuk[a], Yu. Chernets[a]

[a]*Aerospace Faculty, National Aviation University, Lubomyr Huzar 1, 03058 Kyiv, Ukraine.*

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* Corresponding author:

Myron Chernets
E-mail: myron.czerniec@gmail.com

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A B S T R A C T

Using the developed methodology for calculating hybrid (metal-polymer) plain bearings, a study of the effect of temperature on the contact parameters and their resource was conducted. Metal-polymer bearings with a bushing made of polyamide PA6 reinforced with glass or carbon dispersed fibers were considered. The calculation of the maximum initial contact pressures and their change due to wear of the bushing was performed. The predictive estimate of resource of the investigated bearings at various loadings was carried out. Quantitative and qualitative patterns of changes in the maximum initial contact pressures and bearing resource with increasing temperature were established. Studies show that the decrease in the modulus of elasticity due to increasing temperature leads to a decrease in the maximum pressure and increase the resource of metal-polymer bearings, which is useful. This phenomenon is due to a decrease in the rigidity of the polymer composite.

1. INTRODUCTION

Metal-polymer plain bearings (MP) are increasingly used in the technosphere (mechanical engineering, aircraft building, instrument engineering, automobile production, food processing, pharmaceutical, textile, chemical, pulp and paper industry, etc.) and in many other areas of human activity. The use of such bearings ensures trouble-free operation in technical dry friction, where it is impossible to use or provide lubrication. Their widespread use is due to: simplified manufacturability of elements; ease of maintenance or no need for maintenance at all; the ability to operate in special technological conditions (equipment in the food processing, pharmaceutical, paper and pulp, chemical industry, etc.), which make it impossible to lubricate due to contamination of products with lubricants; critical operating conditions (low or high temperatures, vacuum operation); the possibility of lubrication with various substances or liquids that are the working medium; insignificant generated noise and other positive qualities.

The bushings of such bearings are mainly made of various polymers (polyamides - PA, polyacetals - POM, polytetrafluoroethylene - PTFE, polyimides - PI, PAI, polyetherketones - PEEK, etc.) and filled composites based on them. In order to reduce wear and increase the resource of MP plain bearings, modifications of polymer matrix of different type and structure by particles / fibers (molybdenum disulfide, graphite, glass and carbon fiber (dispersed, short, long), bronze powder, polyethylene, polytetrafluoroethylene, etc.) with different volume content are used. This minimizes the disadvantages of unmodified polymers. The mechanical properties of polymers and polymer based composites are differently influenced by the temperature of the working environment. This is especially true of the Young's modulus, which is an important mechanical property of polymer composites because it characterizes their rigidity. A significant decrease in the Young's modulus under the influence of ambient temperature and humidity of the working environment (in particular for polyamide PA6 and PA6 based composites [1]) can undoubtedly have a significant impact on the contact and tribotechnical characteristics. Therefore, an important issue is the study of the effect of decreasing the elastic modulus of certain polymer materials with increasing temperature on these performance characteristics of MP bearings. For this purpose, the developed method [2-6] of calculated evaluation of contact strength and frictional stability characteristics of sliding tribosystems was used. Appropriate effective methods for calculating the MP bearings in the literature are actually absent, especially those that take into account the influence of the temperature factor. However, the known computational [7-14] or numerical methods of research of both metal [1,15,16] and metal-polymer plain bearings [17-19] have limited practical application for calculations of their resource. This, in our opinion, is due to the fact that the authors of these methods use the well - known Archard`s law of abrasive - adhesive wear providing a linear dependence of wear on contact pressure and sliding speed. This type of wear is practically not allowed even in dry friction bearings and therefore the study of metal bearings by these methods are approximate given the inappropriate real mechanism of wear. Instead, the developed calculation method of research of MP bearings [20,21] is based on the fatigue mechanism of wear, and takes into account the change of conditions of contact interaction owing to wear. The developed method is based on the methodology of research of wear kinetics at sliding, as process of frictional-fatigue destruction of surface layers of tribomechanical system elements [2-4,6], and author's calculation methods of plain bearings with metal elements [2,3, 6,22,23].

The article presents the study results of the influence of the Young's modulus change of polyamide composites due to the increase of temperature in the friction unit (hereafter referred to as the temperature) on the maximum contact pressures and resource of MP bearings according to the developed calculation method. Composites PA6 + 30%GF and PA6 + 30%CF, respectively, filled with glass and carbon fine fibers (Ø 6… 10 µm, l = 3… 6 mm) are considered. The problem in such a statement has not been considered in the literature, so this is a new analytical study, which also has practical significance.

2. INVESTIGATION OF THE EFFECT OF TEMPERATURE ON THE MODULUS OF ELASTICITY

Fig. 1 shows the dependences of the modulus of elasticity on temperature. The studies of the effect of temperature on the modulus of elasticity of polyamide PA6 and composites PA6 + 30% GF, PA6 + 30% CF were carried out for the range of 25 - 55 ° C at a relative humidity of 50% [1,20,21 and own results].

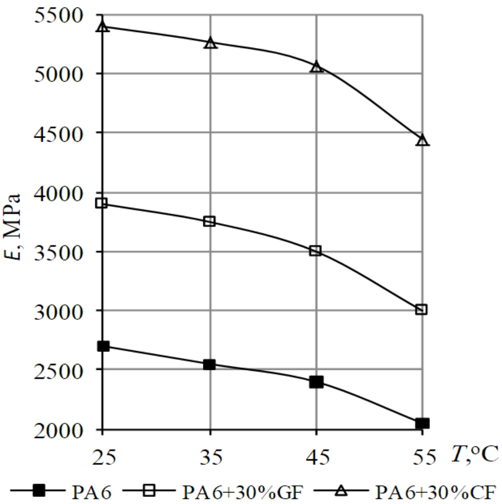

Fig. 1. Effect of temperature on the modulus of elasticity.

A more active decrease in the Young's modulus is observed approximately equally for these polymers starting from a temperature of 40 °C. It should be noted that the operating temperature in metal-polymer bearings is higher than under normal conditions, but should not exceed 60°C, because it can lead to thermal aging and reduced strength of these polyamide composites.

A decrease in humidity leads to an increase in the modulus of elasticity, and vice versa. Increasing the temperature in the MP bearing can also lead to changes in the coefficient of friction in different ways depending on the type of polymer materials and operating conditions [24-32]. In the current study, the coefficient of friction is assumed to be constant.

3. METHOD OF CALCULATION OF INITIAL AND WEAR CONTACT PRESSURES

The main points of the method of calculating the maximum contact pressures are presented in detail in [2-6]. The solution of this contact problem is carried out as a plane contact problem of the elasticity theory about the internal contact of close radii cylindrical bodies (Fig. 2).

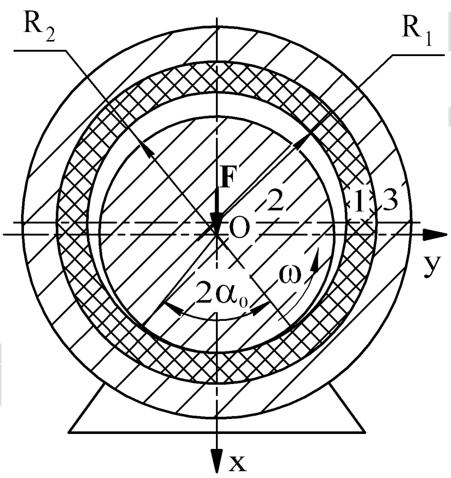

Fig. 2. Scheme of the sliding bearing.

A radial load *N* is applied to the bearing journal. In the considered plain problem it is attributed to the journal length *l*. Accordingly, the reduced concentrated force *N = F/l* is applied in the center of the shaft (disk). Shaft 2 (Fig. 1) rotates with a constant angular velocity ω. Between the shaft journal 2 with radius R_2 and the bushing 1 with radius R_1 located in the housing 3, there is a radial clearance $\varepsilon = R_1 - R_2 \geq 0 \ll R$. The shaft is made of steel and the bushing is made of polyamide composite PA6 + 30%GF or PA6 + 30%CF, so their wear resistance and elastic properties will be significantly different. When the bearing is loaded, an initial contact pressure $p(\alpha)$ occurs in the contact area 2 $R_2\alpha_0$, the distribution law of which is unknown. During wear of the composite bushing, the initial contact pressure $p(\alpha)$ will decrease due to the increase in the contact angle $2\alpha_0$.

According to [2, 3, 6], the contact pressure function is described by the following expression:

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

where $E_0 = (e / R_2)\cos^2(\alpha_0 / 4)$, $e = 4E_1E_2 / Z$, $Z = (1+\kappa_1)(1+\nu_1)E_2 + (1+\kappa_2)(1+\nu_2)E_1$, α is the polar angle, E_1, E_2 are the modulus of elasticity of materials; ν_1, ν_2 are their Poisson's ratios; $\kappa = 3 - 4\nu$.

Maximum contact pressures $p(0)$ occur when α=0. Then

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

To determine the unknown initial contact semiangle α_0, the force equilibrium condition is used, i.e. the external load *F* and the contact pressure $p(\alpha)$ applied to the shaft

$$F = R_2 \int_{-\alpha_0}^{\alpha_0} p(\alpha)\cos\alpha d\alpha = 4\pi R_2 E_0 \varepsilon \sin^2(\alpha_0 / 4). \qquad (3)$$

The initial contact pressures $p(\alpha)$ will decrease due to wear of the composite bushing, i.e. in the tribocontact there will be wear contact pressures $p(\alpha, t, h)$, which are determined as follows [2–5, 22]:

$$p(\alpha, t, h) = p(\alpha) + p(h), \qquad (4)$$

where $p(h)$ is the change in initial pressures due to wear.

The following dependence is used to describe them

$$p(h) = E_h \varepsilon_h \sqrt{\tan^2 \frac{\alpha_{0h}}{2} - \tan^2 \frac{\alpha}{2}} \, , \qquad (5)$$

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

To determine the semiangle of the tribocontact α_{0h} characterizing the contact zone during wear the condition similar to the equilibrium condition (3) is used

$$F = 4\pi R_2 E_0 (\varepsilon + c_{\alpha h}\varepsilon_h)\sin^2(\alpha_{0h}/4), \quad (6)$$

where $\varepsilon_h = h_{k\max}\left(-K_t^{(k)} + h_k'\right)$; $h_1' = h_2/h_1$, $h_2' = h_1/h_2$ – the relative wear in the tribosystem; $h_{k\max}$ - allowable wear of its elements; $K_t^{(1)} = 1, K_t^{(2)} = 2\alpha_0/2\pi$ are the coefficients of mutual overlap of the bearing elements during moving contact; $c_{\alpha h}$ is the rate of the tribocontact angle increase;

$$h_1' = \frac{B_1\tau_{10}^{m_1}(\tau-\tau_{20})^{m_2}}{B_2\tau_{20}^{m_2}(\tau-\tau_{10})^{m_1}}K_t^{(2)},\ h_2' = \frac{B_2\tau_{20}^{m_2}(\tau-\tau_{10})^{m_1}}{B_1\tau_{10}^{m_1}(\tau-\tau_{20})^{m_2}}K_t^{(1)}, \quad (7)$$

where $\tau = fp(0)$ is the Coulomb specific friction force, f is the sliding friction coefficient.

4. CALCULATION OF RESOURCE OF BEARING

The phenomenological tribokinetic mathematical model of wear during sliding friction [2-6] was used for analytical study of the effect of changes in the modulus of elasticity due to increasing temperature on the resource of metal-polymer plain bearing.

To predict the resource of the plain bearing at a given wear of the bushing used the following calculated ratio:

$$t = \frac{-B_k\tau_{ko}^{m_k}}{v c_h \tau(h)\Sigma_k(1-m_k)K_t^{(k)}} *$$

$$*\left\{\left[\tau-\tau_{k0}\right]^{1-m_k} - \left[\left(\tau-\tau_{k0}\right) + c_h h_{k\max}\Sigma_k\tau_h\right]^{1-m_k}\right\}, \quad (8)$$

where B_k, m_k, τ_{k0} are characteristics of wear resistance of tribocouple materials under the accepted external conditions of triboexperimental researches; k is the numbering of bearing elements (Fig. 2); v is the sliding speed; c_h is the wear rate factor; $\Sigma_1 = (-K_t^{(1)} + h_1')$, $\Sigma_2 = \left(K_t^{(2)} - h_2'\right)$; $K_t^{(1)} = 1, K_t^{(2)} = \alpha_0/\pi$.

Accordingly, the specific force of friction during wear

$$\tau(h) = fp(0,t,h) = fE_h\tan(\alpha_{0h}/2). \quad (9)$$

5. RESULTS OF NUMERICAL SOLUTION OF WEAR CONTACT PROBLEM, DISCUSSION

Metal-polymer bearings with the following materials of elements are investigated:
Shaft - steel 45 normalized, grinding; E_2 = 210 GPa, μ_2 = 0,3; B_2= 10^{13}, m_2 = 2, τ_{20}= 0.1 MPa. Bushing: 1) carbon-filled polyamide PA6+30%CF, E_{CF} = 5.20 GPa, μ_{CF} = 0.42; 2) glass-filled polyamide PA6+30%GF, E_{GF} = 3.90 GPa, μ_{GF} = 0.42; volumetric filler content – 30%. Wear resistance characteristics of polyamide composites paired with steel, determined by the results of our triboexperimental studies [20,21]: PA6+30%CF: B_{1CF} = $24\cdot10^{10}$, m_{1CF} = 1.9, τ_{10}= 0.05 MPa; PA6+30%GF: B_{1GF} = $6.67\cdot10^{10}$, m_{1GF} = 1.9, τ_{10} = 0.05 MPa.

Data for calculation: N =1500, 1000, 500 N; $N = F/l$; D_2 = 30 mm; $l = D_2$; ε = 0.2 mm; n_2 = 60 rpm; $f_{GF} = f_{CF}$ = 0.3 – dry friction; $h_{1\max}$ = 1.0 mm – allowable wear of the bushing.

The results of the calculations are presented in Fig. 3 - 7. The solid lines indicate the graphs for carbon-filled PA6 + 30%CF, and dashed lines - for glass-filled PA6 + 30%GF polyamide composites.

Fig. 3 shows the dependences of the change of maximum contact pressures $p(0)$ on the temperature at different values of the working load.

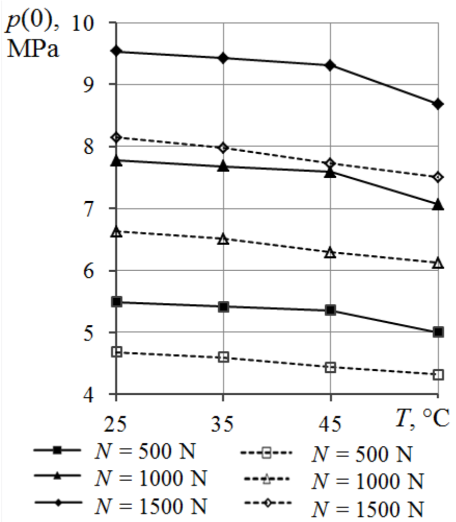

Fig. 3. Dependence of the maximum contact pressures on the temperature. Solid lines – PA6 + 30%CF, dashed lines - PA6 + 30%GF.

There is a certain decrease in pressure with increasing temperature *T*. The dependence is nonlinear at *T*> 45 °C for carbon composite, and it is linear for glass composite. In the accepted temperature range, the decrease will be approximately the same - for the composite PA6 + 30%CF 1.1 times, for the composite PA6 + 30%GF 1.085 times at all values of loads. In the case of a carbon composite bushing, the maximum contact pressures in the bearing will be about 1.17 times greater than in the glass composite, regardless of the load on the bearing.

The reduction of the initial maximum contact pressures in the bearing, ie wear contacts pressures, is shown in Fig. 4. The patterns of change of the maximum wear contact pressures $p(0, t, h)$ will be almost the same as in the case of the initial maximum contact pressures $p(0)$ for both qualitative and quantitative changes. The pressure $p(0)$ reduction is 1,408 times after the composite bushing reaches the accepted allowable wear h_{1*} = 1.0 mm. This quantitative reduction $p(0)$ is almost the same for the composite PA6 + 30%CF and PA6 + 30%GF. Qualitative patterns of change of the maximum wear contact pressures $p(0, t, h)$ will be practically the same as in the case of the initial maximum contact pressures $p(0)$.

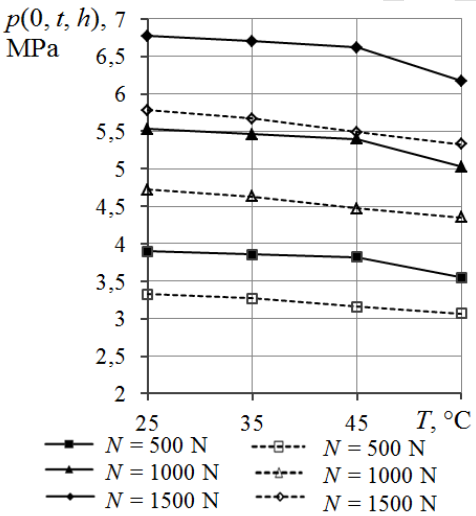

Fig. 4. Effect of the temperature on change of the maximum wear contact pressures. Solid lines – PA6 + 30%CF, dashed lines - PA6 + 30%GF.

The initial contact angles $2\alpha_0$ (semiangles α_0) will increase inversely proportional to the decrease in the initial maximum contact pressures $p(0)$ in both qualitative and quantitative terms (Fig. 5). Their interrelation is obvious, ie the decrease in contact pressures is due to an increase in the contact angle in the conjunction.

Wear of the composite bushing leads to an increase in the initial contact semiangles α_0 (Fig. 6). Qualitative and quantitative regularities of their increase, inverted to those which are observed at decrease in wear contact pressures remain $p(0, t, h)$.

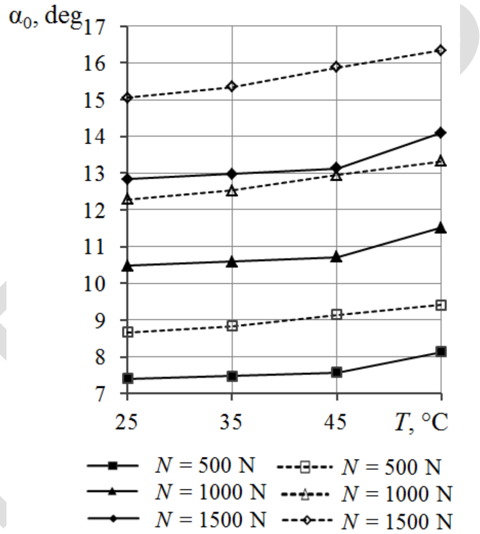

Fig. 5. Dependence of the contact semiangles on the temperature. Solid lines – PA6 + 30%CF, dashed lines - PA6 + 30%GF.

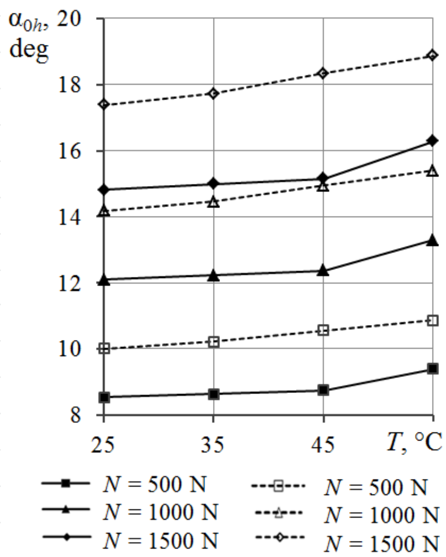

Fig. 6. Influence of the temperature on change of the contact semiangles. Solid lines – PA6 + 30%CF, dashed lines - PA6 + 30%GF.

An important applied aspect of the study by this method is the analysis of the effect of reducing the modulus of elasticity of composites under the influence of temperature on the bearing life. The results of the calculation are presented in Fig. 7. As the temperature increases, the resource of the bearing also increases: nonlinearly for bushings made of PA6 + 30%CF, and linearly for bushings made of PA6 + 30%GF. For a bearing with a carbon composite bushing, it will be 1.2 times higher, and with a glass composite bushing - 1.17 times higher when the bushing reaches the allowable wear h_{1*} = 1.0 mm along the line of acton of external load F (Fig. 2, α = 0). This is due to a decrease in contact pressures (Fig. 3)

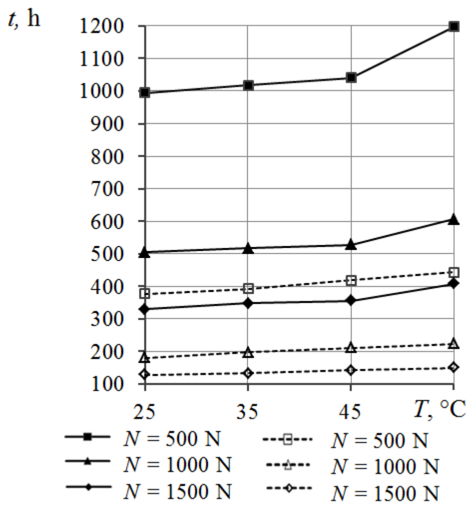

Fig. 7. The influence of the temperature on the bearing resource.

In addition, the resource of the bearing with a carbon composite bushing will be much higher than with a glass composite bushing. The resource ratio will be approximately the same for all loads at T = 25-45 °C: 2.48..2.64 times at N = 500 N, 2.5… 2.64 times at N = 1000 N, 2.5… 2.6 times at N = 1500 N. And at T = 55 °C this ratio will be slightly higher - 2.71 times regardless of the load.

6. CONCLUSION

1. It is established that the increase in the temperature leads to a decrease in the rigidity of polyamide composites PA6 + 30%CF and PA6 + 30%GF, manifested by a decrease of Young's modulus. As a result, the level of the initial maximum contact pressures $p(0)$ decreases. That is, an increase in temperature of 2.2 times has a positive effect on the bearing capacity of MP bearings (up to 1.1 times).
2. When the bushing wears, the patterns of further reduction of the initial maximum contact pressures $p(0)$ are preserved.
3. The increase in the temperature increases the resource of MP bearings. That is, the higher temperature has a positive effect on the resource of MP bearings (up to 1.2 times).
4. In addition, it was found that the calculated resource of the bearing with a carbon composite bushing will be 2.5… 2.7 times higher than with a glass composite bushing.

The developed analytical method of the elasticity theory for calculation of MP sliding bearings provides at a design stage an effective forecast estimation of their contact parameters and a resource taking into account influence of temperature in the friction unit on decrease in the modulus of elasticity of polymeric composites. That is, this method is a practical engineering method for calculating the bearing capacity of the MP bearing by the criterion of maximum contact pressures, and the calculation of its resource by the criterion of the bushing allowable wear.

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