 DOI: 10.20858/tp.2016.11.4.7

bearing; friction; heat generation; convective heat transfer; temperature

Igor KOLESNIKOV, Sergey DANILCHENKO*
Rostov State Transport University
Narodnogo Opolcheniya sq. 2, Rostov-on-Don, 344038 Russia
Elena KOLOSOVA, Mikhail CHEBAKOV, Alexander LYAPIN
Southern Federal University
B. Sadovaya 105/42, Rostov-on-Don, 344006 Russia
**Corresponding author* E-mail: sergey.a.danilchenko@gmail.com

MODELLING OF THERMOELASTIC TRANSIENT CONTACT INTERACTION FOR BINARY BEARING TAKING INTO ACCOUNT CONVECTION

Summary. Serviceability of metal-polymeric "dry-friction" sliding bearings depends on many parameters, including the rotational speed, friction coefficient, thermal and mechanical properties of the bearing system and, as a result, the value of contact temperature. The objective of this study is to develop a computational model for the metallic-polymer bearing, determination on the basis of this model temperature distribution, equivalent and contact stresses for elements of the bearing arrangement and selection of the optimal parameters for the bearing system to achieve thermal balance.

Static problem for the combined sliding bearing with the account of heat generation due to friction has been studied in [1]; the dynamic thermoelastic problem of the shaft rotation in a single and double layer bronze bearings were investigated in [2, 3].

МОДЕЛИРОВАНИЕ ТЕРМОУПРУГОГО НЕСТАЦИОНАРНОГО КОНТАКТНОГО ВЗАИМОДЕЙСТВИЯ В БИНАРНОМ ПОДШИПНИКЕ СКОЛЬЖЕНИЯ С УЧЕТОМ КОНВЕКТИВНОГО ТЕПЛООБМЕНА

Аннотация. Работоспособность полимерных подшипников скольжения «сухого трения» зависит от многих параметров, включающих в себя скорость вращения вала, коэффициент трения, термо-механические свойства элементов подшипниковой системы и, как следствие, величины результирующих контактных температур. Целью данного исследования является разработка расчётной модели работы бинарного подшипника скольжения «сухого трения» с полимерными цилиндрическими вставками, определения на ее основе распределения температур, эквивалентных и контактных напряжений в элементах подшипниковой системы и подбор оптимальных параметров подшипниковой системы, при которых достигается тепловой баланс.

Статическая задача для комбинированного подшипника скольжения с учётом тепловыделения от трения была исследована в работе [1], динамическая термоупругая задача о вращении вала в однослойном подшипнике из бронзы была исследована в работе [2], двухслойного с антифрикционным покрытием в [3].

1. INTRODUCTION

Clarification for the behavior of surface layers in metal-polymeric thermoelastic tribocontact – one of the central problems in tribotechnology. Therefore, for a deeper knowledge of the processes on the contact surface, it is necessary to develop not only the experimental methods of diagnosis but adequate theoretical models. At present, there is a single point of view that the determining factor in the operational mode of the metal-polymer interface is a thermal stress at friction node.

In recent years, in domestic and foreign scientific literature much attention is paid to the theoretical (numerical, analytical) and experimental studies of the bearing performance of polymeric "dry friction" bearings [4-12]. The widely used polymer in the manufacture of bearing is PTFE-4. The main reason that caused the interest to this material is the fact that during "dry friction" of metals and PTFE-4 at low sliding speed the friction coefficient is very small and does not usually exceed normal coefficients of friction in metal bearings in the presence of lubricants. Pure PTFE has good chemical resistance, low friction and a wide operating temperature range, but it is subject to deformation under load and heavy wear. Fillers introduced into PTFE improve wear resistance about a thousand times, the resistance of the load pressure 2-5 times, and thermal expansion is reduced by 2-3 times. The investigation of the influence of sliding speed, bearing pressure and temperature on the friction and wear in sliding bearings made of PTFE with fillers is subject of [13].

In contrast to the above-mentioned works, the present article examines metallic-polymer plain bearing with PTFE inclusions in unsteady interaction of bearing components, taking into account the heat generation from friction and convective heat transfer, based on a 3D model.

Bearing type with anti-friction PTFE-4 inclusions is shown in Fig. 1.

Fig. 1. The view of studying bearing
Рис. 1. Исследуемый подшипник

2. THE MATHEMATICAL FORMULATION OF THE PROBLEM

Let us describe the nonstationary dynamic coupled contact problem of thermoelastic interaction of elastic homogeneous cylinder (hereinafter - the shaft) with the inner surface of the cylindrical layer of finite length with inserts (hereinafter - the bearing). The geometry of bearing is in a cylindrical coordinate system and $O_1 r\varphi z$ a cylindrical layer ($R_1 \leq r \leq R_2$, $0 \leq \varphi \leq 2\pi$, $-l/2 \leq z \leq l/2$) (Fig. 2a) shows a three-dimensional examined object (Figure 1b) - statement of the problem in a front plane.

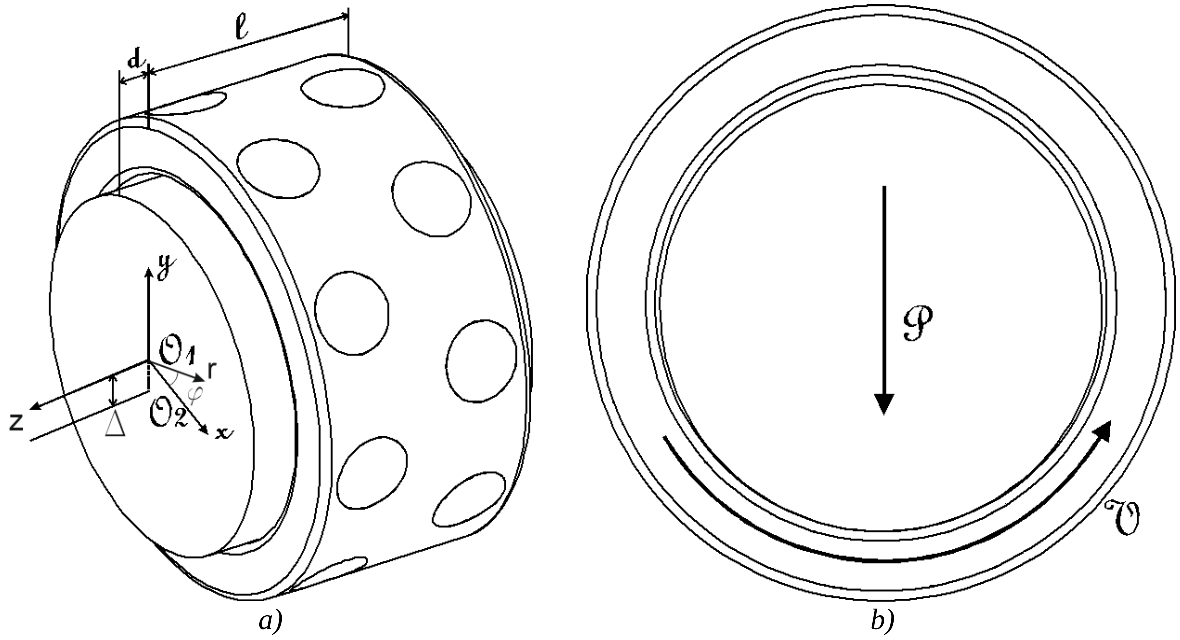

Fig. 2. Statement of the problem
Рис. 2. Постановка задачи

The outer surface of the bearing S^{ext} $(r = R_2,\ 0 \le \varphi \le 2\pi,\ -l/2 \le z \le l/2)$ is fixed in all directions except the rotational. The surface $r = R_1$ is under pressure of shaft with the radius $R_0 = R_1 - \Delta$ ($\Delta \ge 0, \Delta \le 1$), taking the volume ($-l/2 - d \le z \le l/2 + d$) with the center in O_2, and shaft axis is parallel to bearing axis, with the line of initial contact ($r = R_1$, $\varphi = 0$, $-l/2 \le z \le l/2$).

In local cylindrical coordinate system $O_2 r_1 \varphi z$ (Fig.2 (a), this system corresponds to the shaft) on the surfaces $S_1 = (r_1 = R_0,\ 0 \le \varphi \le 2\pi,\ -l/2 - d \le z \le -l/2)$ and $S_2 = (r_1 = R_0,\ 0 \le \varphi \le 2\pi,$ $l/2 \le z \le l/2 + d$) of the shaft, there are applied loads, and each is $P/2$ and acts vertically down. Bearing rotates counterclockwise with an angular velocity ω (rad/sec) during T^{int} seconds. Between the shaft and the bearing, there are Coulomb friction forces with the coefficient k. The corresponding values of friction coefficients for stainless steel/PTFE (Teflon) bearings could be found in [14]. Let $S_{\pm}^{shaft}$ be surfaces of shaft ends, lying in planes $z = \pm l/2 \pm d$, respectively.

It is assumed that at the shaft and bearing surfaces that make contact with the environment there is defined convective heat exchange with a convective heat transfer coefficient a. Let the initial state of the shaft temperature and the bearing be the same as the environment and equal to $0\ °C$.

As a result, we come to the solution of non-stationary coupled contact thermoelastic problem with the classical equations of motion for the thermoelastic medium [15] (index $i = 1$ refers to the shaft, the index $i = 2$ relates to a bearing):

$$(\lambda_i + 2\mu_i)\nabla\nabla \cdot \mathbf{u}^{(i)} - (\lambda_i + \mu_i)\nabla \times \nabla \times \mathbf{u}^{(i)} - \gamma_i \nabla \theta^{(i)} - \rho_i \ddot{\mathbf{u}}^{(i)} = 0,$$
$$\Lambda_i \nabla \cdot \nabla \theta^{(i)} - C_i \dot{\theta}^{(i)} - t_{0i}\gamma_i \frac{\partial}{\partial t} \nabla \cdot \mathbf{u}^{(i)} = 0; \tag{1}$$

Constitutive relations for coupled thermoelasticity have the form:

$$\begin{aligned}
\sigma_{11}^{(i)} &= (2\mu_i + \lambda_i)\varepsilon_{11}^{(i)} + \lambda_i(\varepsilon_{22}^{(i)} + \varepsilon_{33}^{(i)}) - \gamma_i\theta^{(i)}, \sigma_{12}^{(i)} = 2\mu_i\varepsilon_{12}^{(i)}, \\
\sigma_{22}^{(i)} &= (2\mu_i + \lambda_i)\varepsilon_{22}^{(i)} + \lambda_i(\varepsilon_{11}^{(i)} + \varepsilon_{33}^{(i)}) - \gamma_i\theta^{(i)}, \sigma_{23}^{(i)} = 2\mu_i\varepsilon_{23}^{(i)}, \\
\sigma_{33}^{(i)} &= (2\mu_i + \lambda_i)\varepsilon_{33}^{(i)} + \lambda_i(\varepsilon_{11}^{(i)} + \varepsilon_{22}^{(i)}) - \gamma_i\theta^{(i)}, \sigma_{31}^{(i)} = 2\mu_i\varepsilon_{31}^{(i)}, i = 1,2
\end{aligned} \tag{2}$$

here $\gamma_i = (3\lambda_i + 2\mu_i)\alpha_i$, α_i - thermal expansion coefficient, Λ_i - coefficient of thermal conductivity, C_i - heat capacity, t_{0i} - the absolute temperature of the initial state of the body, ρ_i - density, λ_i, μ_i - Lame coefficients, $u^{(i)}$ - displacements vector, $\theta^{(i)}$ - temperature, $\sigma_{jk}^{(i)}$, $\varepsilon_{jk}^{(i)}$ - components of the stresses and strain tensors.

The boundary and initial conditions have the form:

$$\begin{aligned}
& u_r^{(2)} = u_z^{(2)} = 0, \ \mathbf{x} \in S^{ext}, \\
& \int_{S_1 \cup S_2} \sigma_{rr}^{(1)} dS = P, \ \mathbf{x} \in S_1 \cup S_2, \\
& \tau_{r\phi}^{cont} = k\sigma_{rr}^{cont}, \ \mathbf{x} \in S^{cont}, \\
& q_{cont} = k_{cont}(\theta^{(1)} - \theta^{(3)}), \ \mathbf{x} \in S^{cont}, \\
& q^{(1)} = a(\theta_{env} - \theta^{(1)}), \ \mathbf{x} \in S_1 \cup S_2 \cup S_-^{shaft} \cup S_+^{shaft}, \\
& q^{(2)} = a(\theta_{env} - \theta^{(2)}), \ \mathbf{x} \in S_{2\,layer}^{ext} \cup S^{ext}, \\
& q^{(3)} = a(\theta_{env} - \theta^{(3)}), \ \mathbf{x} \in S_{1\,layer}^{ext}, \\
& \dot{u}_\phi^{(2)}(t) = \omega, \ t \in [0, T^{\text{int}}], \\
& \theta^{(i)}(0) = 0, \ i = 1,2,3,
\end{aligned} \tag{3}$$

here $\tau_{r\varphi}^{cont}, \sigma_{rr}^{cont}$ - contact stresses, S^{cont} - is the contact surface between the shaft and the bearing, q_{cont} - heat flow in the contact region, k_{cont} - contact thermal conductivity coefficient, $S_{2\,layer}^{ext}$ ={ $0 \le \varphi \le 2\pi$, $R_1 \le r \le R_2$, $z = \pm l/2$ } – lateral bearing surfaces, θ_{env} - ambient temperature, $q^{(i)}$ - heat flows.

3. PROBLEM STUDY

To solve this problem the finite element method and the special software ABAQUS were used. To improve the convergence of the algorithm for solving the problem and reducing calculation time, the solution was carried out in two stages. The first stage is static contact problem of elasticity for the shaft pressing on the surface of the bearing, the second is coupled transient thermoelastic contact problem of the shaft rotation.

The finite element mesh is built using 8-node coupled thermoelastic C3D8T element. The resulting finite element model of the bearing system is shown in Figure 2. For the simulation of contact interaction between the inner surface of the bearing and the outer surface of the shaft, both were covered by the contact pairs of elements. To solve the non-stationary problem, minimal and maximal time steps are set, and the settings that allow the package to choose the optimal time step during the calculations are defined.

4. RESULTS

In the numerical experiments, the following values of the geometrical and material parameters were set as in Tab. 1.

Table 1

Geometrical parameters for problem

Parameter	Inner radius for bearing R_1, m	Outer radius for bearing R_2, m	Clearance between the shaft and the bearing Δ, m	Bearing length $l = 0.03$ m	Shaft protrusion d, m
Value	0.023	0.03	$9 \cdot 10^{-5}$	0.03	0.005

The bearing is assumed to be made of bronze and the inserts contacting with the shaft of PTFE-4. During the calculations shaft is assumed to be made of steel. Corresponding material properties for the materials mentioned could be found in Tabs. 1-2.

Table 2

Material properties for components of system

Parameter	Density $\rho, kg/m^3$	Lame coefficient λ, GPa	Lame coefficient μ, GPa	Thermal conductivity $\Lambda, W/(m \cdot K)$	Thermal expansion coefficient α, K^{-1}	Specific heat coefficient $C, J/(kg \cdot K)$;
Value for bronze	8800	63	42	76	$1{,}8 \cdot 10^{-5}$	435
Value for PTFE-4	2200	230	154	0.25	See Tab. 3	1040
Value for steel	7800	121	810	50.2	$1.1 \cdot 10^{-5}$	460

Table 3

The dependence of the thermal expansion coefficient for PTFE on temperature

t, C	-10	20	50	110	120	200	210	280
$\alpha_2(t) \cdot 10^{-4}$, K^{-1}	0.8	2.5	1.1	1.1	1.5	1.5	2.1	2.1

The convective heat transfer coefficient for the metal-air pair on the surface of the shaft and bearing bordering with surroundings is indicated as a. It was chosen to be in the calculations equal to $20W/m^2$

The influence of different values of the coefficient of friction k and the load P on the maximum temperature $t_{\max}$ of the bearing (marked with index $bear$) and shaft (marked index $shaft$) for the problem after the rotation for $T^{\text{int}} = 5$ seconds was studied.

Figs. 3-4 show the distribution of temperature and contact pressure for the value of pressing force equals to $2 \cdot 10^4$ N and rotation speed of 10π rad/sec after 5 seconds. For all other stated cases, the picture of distribution for the temperature field and contact stresses are typically the same, the difference is only in numerical values. Taking this into account, the presentation of corresponding figures is not sufficient and only plots for contact stresses and temperatures with time will be given.

As expected, in the zones of PTFE inserts, the temperatures are less than those for other zones. It should be mentioned that temperature distribution is unsymmetrical.

Elements of a bearing system are significantly heated during rotation of the shaft, and the increase in the speed or rotation time leads to a rise in temperature. Figure 5 shows the character of the temperature change of the maximum values for the shaft and the bearing. Here and after all results correspond to point of shaft with coordinates ($r_1 = R_0$, $\phi = -\pi / 2$, $z = 0$). Clearance between the shaft and the bearing $\Delta = 9 \cdot 10^{-5} m$, the value of pressing force P equals to $3 \cdot 10^4$ N, the coefficient of friction $k = 0.04$, rotation speed $\omega = 10\pi$ rad/sec. The period of rotation is $T = 0.2\,\sec$. It can be seen that the bearing temperature is considerably lower than the temperature of the shaft, due to the greater heat transfer surface area. Ramp character for bearing temperature is due to the presence of the gap between the contacting bodies. Bearing point, entering the contact zone, receives a certain amount of heat due to friction, then moving in the area of the gap between the shaft and the bearing, some part of the heat is redistributed inside the bearing volume and is emitted into the environment, which leads to a small decrease in temperature for the corresponding point.

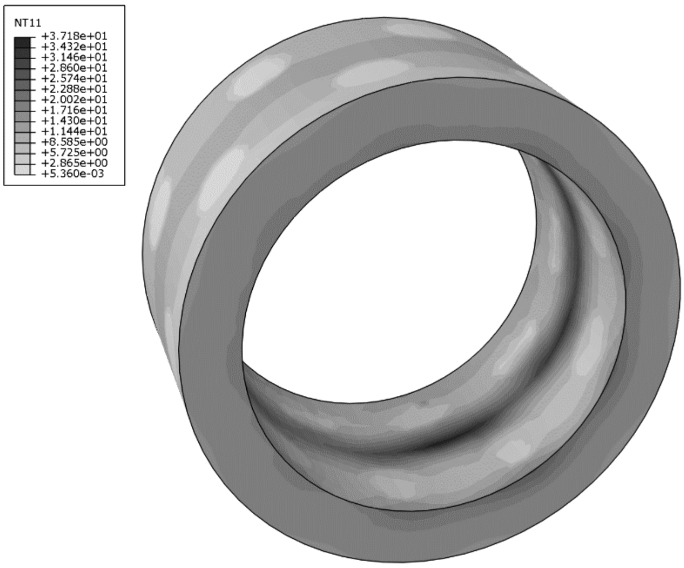

Fig. 3. Temperature distribution for bearing system
Рис. 3. Распределение температуры в подшипнике

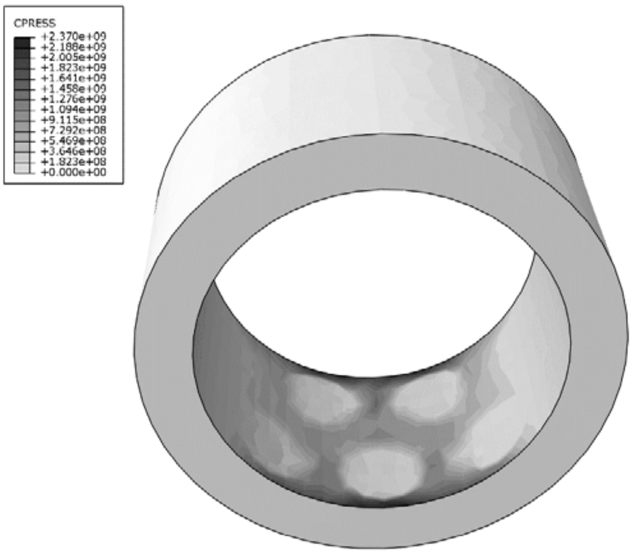

Fig. 4. Contact pressure distribution for bearing system
Рис. 4. Распределение контактного давления в подшипнике

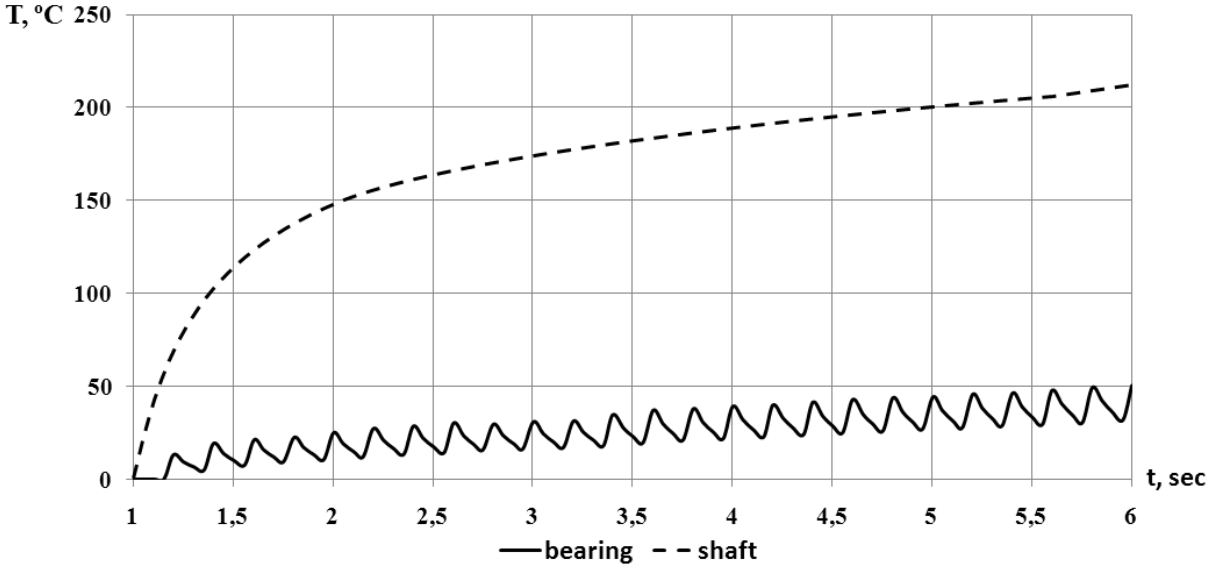

Fig. 5. Character of maximum temperature value change for the shaft and the bearing
Рис. 5. Характер изменения максимального значения температуры для вала и подшипника

Two series of simulations were performed for the temperature distribution in the system at values of shaft speed $\omega = 10\pi$ rad / sec, the convective heat transfer coefficient $a = 20W/m^2$, and the coefficient of friction $k = 0.04$ [14] for various values of the pressing force $P = 10^4; 2\cdot10^4; 3\cdot10^4$ N. Maximum temperature and contact pressure values for shaft are shown in Figs. 6-7. The results demonstrate a proportional increase in the degree of warming for the system by increasing the pressing force.

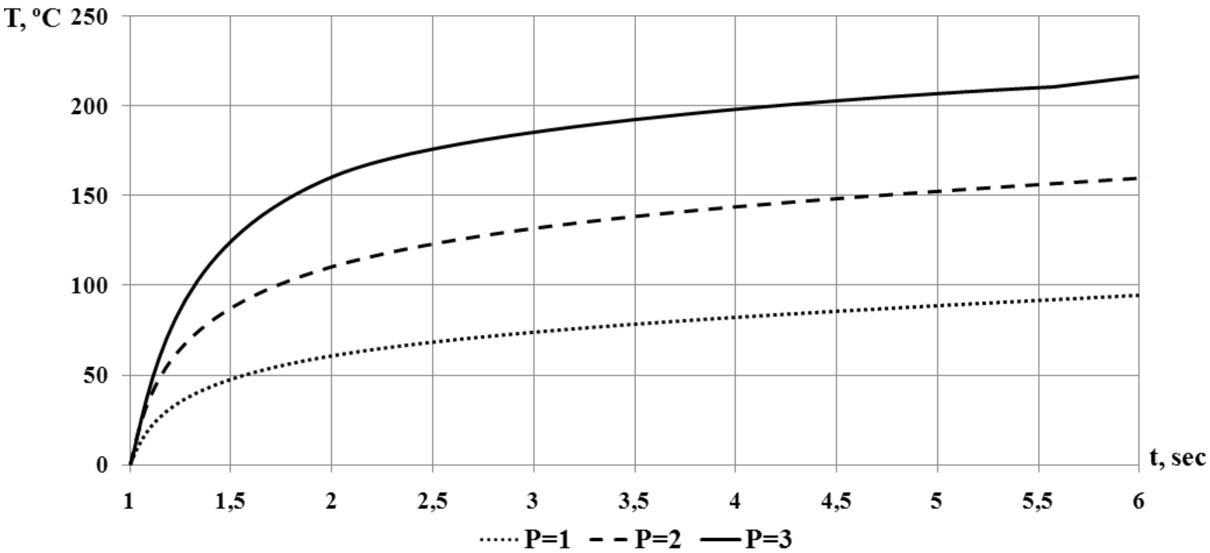

Fig. 6. The character of the maximum temperature change for the various pressing force values
Рис. 6. Характер изменения максимального значения температуры для различных значений силы вдавливания

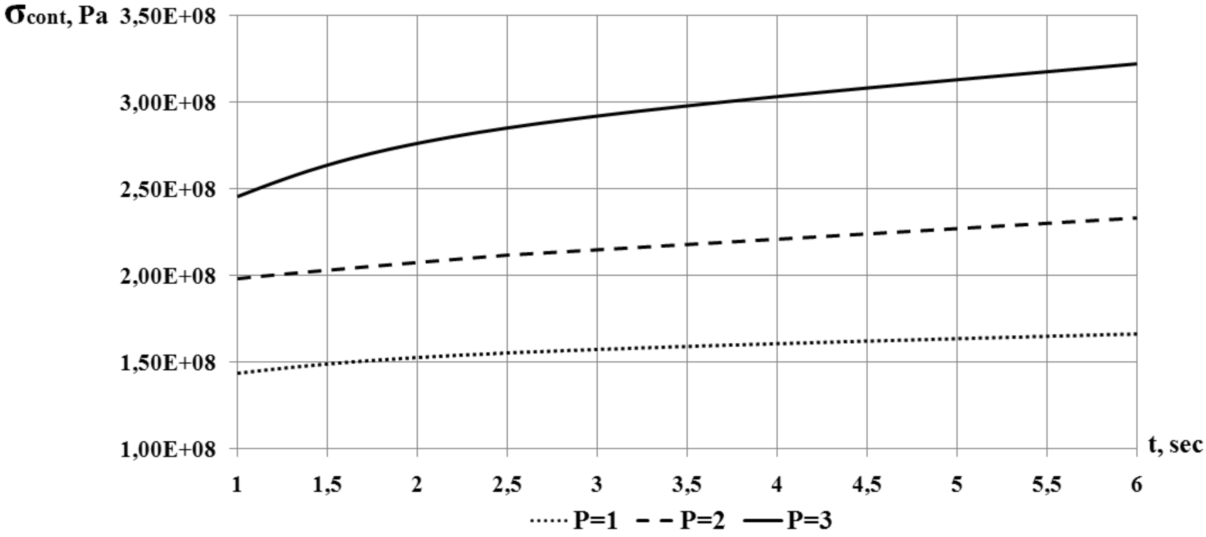

Fig. 7. The character of the maximum contact pressure change for the various pressing force values
Рис. 7. Характер изменения максимального значения контактного давления для различных значений силы вдавливания

5. CONCLUSIONS

Studies have shown that of great significance in achieving thermal balance for the demonstrated system is the value of convection for the shaft and bearing free surfaces, while increasing parameters such as load value, the coefficient of friction, the achieving of thermal balance in the bearing system is slowed down. It has been observed that during some initial period of time the temperature rises significantly, then the growth pattern shape becomes flatter. Note that the finite element method using Abaqus package for this problem turned out to be quite effective and allows one to explore the effectiveness of such problems for different values of the input geometrical and mechanical parameters.

Statement of the problem, experimental characterization of PTFE-4 polymer and investigation of the influence of parameters of thermoelastic contact interaction in binary bearing on its stress-deformed state and temperature fields are made by I.V. Kolesnikov and S.A. Danilchenko and supported by the Russian Science Foundation under grant 14-29-00116 and performed in Rostov State Transport University. Mathematical formulation of the problem, CAD geometry preparation and finite element modeling are made by M.I. Chebakov, A.A. Lyapin and E.M. Kolosova and supported by the Ministry of Education and Science of the Russian Federation under project no. 213.01 11/2014 28, the Russian Foundation for Basic Research under project no. 16-08-00852 and performed at the Southern Federal University.

References

1. Колосова, Е.М. Конечно-элементное моделирование контактного взаимодействия в комбинированном подшипнике скольжения с учетом тепловыделения от трения. *Известия Вузов Сев.-Кавказс. рег. техн. наук*. 2010. No. 5. P. 35-39. [In Russian: Kolosova, E.M. Finite-element simulation of contact interaction in combined sliding bearing by considering friction heat radiation. *Izv. Vyssh. Uchebn. Zaved. Sev.-Kavk. Reg., Ser. Tekhn. Nauk*].
2. Колосова, Е.М. & Ляпин, А.А. & Чебаков, М.И. Расчет термоупругого контактного взаимодействия в подшипнике скольжения с учетом трения. *Известия Вузов Сев.-Кавказс.*

рег. Серия техн. наук. 2014. No. 6. P. 73-76. [In Russian: Kolosova, E.M., Lyapin, A.A., Chebakov, M.I. Calculation of thermoelastic contact interaction in sliding bearing considering the friction. *Radioelectronics North-Kavkazs. reg. Series tech. Sciences*].
3. Kolesnikov, V.I. & Kolosova, E.M. & Chebakov, M.I. Simulation of a Nonstationary Contact in a Sliding Bearing with Allowance for Frictional Heating and Convective Heat Transfer. *Journal of Machinery Manufacture and Reliability*. 2016. Vol. 45. No. 2. P. 156-162.
4. Александров, В.М. Определение контактных температур в цилиндрическом сочленении. *Известия Российской академии наук. Механика твердого тела*. 2010. No. 5. P. 86-88. [In Russian: Alexandrov, V.M. The way to detect contact temperatures in cylindrical jointing. *Izv. Russ. Akad. Nauk. Mekh. Tverd. Tela*].
5. Губарева, Е.А. & Мозжорина, Т.Ю. & Щетинин, А.Н. Моделирование взаимодействия цилиндрических и сферических тел с покрытиями при износе и тепловыделении. *Инженерный журнал: наука и инновации*. 2014. No. 3(27). P. 5. [In Russian: Gubareva, E.A. & Mozzhorina, T.Yu. & Schetinin, A.N. The way to simulate interactions between cylindrical and spherical bodies with coatings under wear and heat radiation. *Inzh. Zh.: Nauka Innov]*.
6. Дроздов, Ю.Н. & Надеин, В.А. & Пучков, В.Н. & Пучков, М.В. Трение и ресурс цилиндрических подшипников скольжения, работающих без смазки. *Проблемы машиностроения и надежности машин*. 2006. No 4. P. 72-78. [In Russian: Drozdov, Y.N. & Nadein, V.A. & Puchkov, V.N. & Puchkov, M.V. Friction and life time of cylindrical friction bearings operated without lubricants. *J. Mach. Manuf. Reliab*].
7. Tzanakis, I. & Conte, M. & Hadfield, M. & Stolarski, T.A. Experimental and analytical thermal study of PTFE composite sliding against high carbon steel as a function of the surface roughness, sliding velocity and applied load. *Wear*. 2013. No. 303. P. 154-168.
8. Liwen, M. & Yijun, S. & Xin, F. & Jiahua, Z. & Xiaohua, L. The effect of thermal conductivity and friction coefficient on the contact temperature of polyimide composites: Experimental and finite element simulation. *Tribology International*. 2012. No. 53. P. 45-52.
9. Ali, R. & Wouter, O. & Van Paepegem, W. & De Baets, P. & Degrieck, J. Experimental study and numerical simulation of the large-scale testing of polymeric composite journal bearings: Three-dimensional and dynamic modeling. *Wear*. 2011. No. 270. P. 431-438.
10. Imado, K. & Miura, A. & Kido, Y. Influence of testing method on the contact pressure distribution and its effect on the coefficient of friction in polymeric bearing. *Tribology International*. 2007. No. 40. P. 390-396.
11. Komanduri, R. & Bing Hou, Z. Thermal analysis of dry sleeve bearings - a comparison between analytical, numerical (finite element) and experimental results. *Tribology International*. 2001. No. 34. P. 145-160.
12. Колесников, В.И. *Теплофизические процессы в металлополимерных трибосистемах*. Москва: Наука. 2003. 279 p. [In Russian: Kolesnikov, V.I. *Thermal processes in metal tribosystems*. Moscow: Nauka].
13. Demirsi, M.T. & Duzcukoglu, H. Wear behaviors of Polytetrafluoroethylene and glass fiber reinforced Polyamide 66 journal bearings. *Materials and Design*. 2014. Vol. 57. P. 560-567.
14. Taylor, J.C. & Stanton, J.F. Friction coefficients for stainless steel/ptfe (teflon) bearings. *Final Research Report WisDOT Project ID 0092-08-13 Report No. WHRP 10-01*. Department of Civil Engineering, University of Washington. 2010. 126 p.
15. Новацкий, В. *Динамические задачи термоупругости*. Москва: Мир. 1970. 253 p. [In Russian: Nowacki, W. *Dynamic Problems of Thermoelasticity*. Moscow: Mir].

Received 03.03.2015; accepted in revised form 19.10.2016