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# MODELLING OF THERMOELASTIC PROCESSES AND TEMPERATURE DISTRIBUTION IN METAL TRIBOPAIRS DURING DRY FRICTION

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***Abstract:*** *In this paper, a model for calculation of the temperature distribution in the metal tribopair bar - plane in the conditions of dry friction is developed. A model for solving of the problem of thermoelasticity for a metal bar under uniaxial tension-compression is also proposed. A simulation of the interaction between the bar and the plane was carried out taking into consideration various conditions, such as ambient temperature and linear dimensions of the contacting pairs. The simulation results showed that the distribution of the thermal field nonlinearly depends on the linear dimensions of the body, but the frictional heating in the first counter-body can be compensated by small changes of the linear dimensions.*

*When calculating the stress-strain state of the bar, a non-linear dependence of the stress in the bar was obtained, which is due to the presence of contact pressure.*

*The modelling results were proved experimentally with the use of the friction machine MTU-1.*

*The results of the work can help us predict frictional heating for metal tribopairs and decrease its negative consequences.*

***Keywords:*** *frictional heating, thermoelastic processes, distribution of the thermal field, stress-strain state of tribopair elements, uniaxial tension-compression.*

## 1. INTRODUCTION

In the process of friction, a sufficiently large part of the energy transforms to heat losses [1]. Energy dissipation processes, as a rule, lead to negative consequences, such as catastrophic wear, ignition of lubricant, etc. The theoretical description of the processes occurring in this case is extremely difficult, which is due to the nonlinearity of the processes, as well as taking into account a large number of factors affecting both the friction process and the energy transformation [1].

The thermal problem of friction was analyzed in [1-3]. The friction heat between sliding bodies can cause thermoelastic deformation that changes the contact pressure distribution. The sliding velocity that is higher than the critical velocity can cause the appearance of local high temperature areas called hot spots. These hot spots with high local stresses may cause material degradation, eventual failure and undesirable frictional vibrations [4-6]. Thermoelastic processes in composite materials were investigated in [7]. The results of these researches made it possible to achieve significant success in engineering calculations of non-stationary friction modes in brakes and friction devices with complex thermal conditions [8-12].

The development of theories based on simplified fundamental models that take into account the influence of wear, elasticity, phase transitions, etc. is of great importance.

In this paper, we propose a model for distribution of thermal fields in the volume of a solid body in dry friction conditions and a model for thermoelastic processes in a metal bar under uniaxial tension-compression during dry friction.

## 2. DISTRIBUTION OF THERMAL FIELDS

The basis of the simulation is the classical thermal conductivity equation:

$$-\nabla\cdot \mathrm{h}+\mathrm{b}=\mathrm{c}\dot{\mathrm{T}},\mathrm{h}=-\mathrm{k}\nabla\mathrm{T},-\nabla\cdot\mathrm{h}=\mathrm{k}\Delta\mathrm{T}', \quad (1)$$

where h is the heat flux vector, b is the volumetric heat release, *c* is the volumetric heat capacity, k is the thermal conductivity coefficient. It must be taken into account that heat will be concentrated on the contact surface.

The initial conditions for equation (1) are the following:

$$\mathrm{T}\big|_{O_1}=\mathrm{T_e},\mathrm{k}\partial_\mathrm{n}\mathrm{T}\big|_{O_2}=\beta=\lambda(\mathrm{T_e}-T); \quad (2)$$
$$\mathrm{T}\big|_{t=0}=\mathrm{T_0}.$$

On the contactless surface of the body $\mathrm{O}_1$, the ambient temperature is $\mathrm{T_e}$. The heat flux $\beta$ is specified on the contact part of the surface $\mathrm{O}_2$. However, it can be expressed in terms of temperature difference and heat transfer coefficient $\lambda$.

To simplify the model, we assume that the volumetric heat capacity and thermal conductivity coefficient are constant.

The differential equation in partial derivatives with given boundary and initial conditions can be solved numerically by a variational method with an approximation by a system of coordinate functions. The method is based on a variation equation:

$$\int_V(\mathrm{k}\Delta\mathrm{T}+\mathrm{b}-cT)\delta T dV+\int_O(\beta-\mathrm{k}\partial_\mathrm{n}T)\delta T dO=0, \quad (3)$$

where $\delta\mathrm{T}$ is an arbitrary temperature increment.

Approximation of the solution with given coordinate functions:

$$T=\sum_{i=1}^{N}\theta_\mathrm{i}(\mathrm{t})\varphi_\mathrm{i}(\mathrm{r})=\theta^\mathrm{T}\varphi=\varphi^\mathrm{T}\theta, \quad (4)$$
$$\delta\mathrm{T}=\delta\theta\varphi$$

We use matrix notation. Functions $\phi_i$ are set by us. Variable functions $\theta_\mathrm{i}$ should be defined.

Substituting (4) into (3) and taking into account arbitrariness of, we obtain the following:

$$\mathrm{C}\dot{\Theta}+\mathrm{K}\Theta=\mathrm{B}(t), \quad (5)$$

where

$$\mathrm{C}=\int_V \mathrm{c}\varphi\varphi^T dV;$$
$$\mathrm{K}=-\int_\mathrm{V}\mathrm{k}\varphi\Delta\varphi^\mathrm{T}dV+\int_\mathrm{O}\mathrm{k}\varphi\partial_\mathrm{n}\varphi^\mathrm{T}dO; \quad (6)$$
$$\mathrm{B}=\int_V \mathrm{b}\varphi dV+\int_O \beta\varphi dO.$$

In equation 6, C is the heat capacity matrix, K is the thermal conductivity matrix, B is the column of thermal loads.

When calculating the intensity of heat generation during friction contact, we use a simplified contact scheme (Fig. 1).

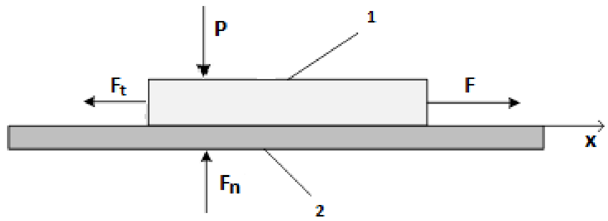


**Figure 1.** Bar on a rough plane

The bar moves on a plane under the action of force F, overcoming the force of dry friction $\mathrm{F_t}$. The pressing force is equal to P, the normal reaction is $\mathrm{F_n}$.

According to Coulomb's law:

$$\mathrm{F}=\mu\mathrm{F}_n, \quad (7)$$

where $\mu$ is friction coefficient. Since mechanical energy remains constant, all mechanical power is equal to heat generation. Thermal power per unit area could be defined as following:

$$\mathrm{b}=\tau\mathrm{v}, \quad (8)$$

where $\tau$ is the tangential stress during friction, $v$ is the velocity of the body.

In the contact of two bodies of the same material, the heat release is distributed approximately equally between both bodies.

The temperature distribution in the volume of the bar will be obtained by solving equation (5).

We take steel C22 as counter-bodies for initial data.

The number of functions N is 7. For steel C22: k = 45.4, W / (m·K), c = 460 J / m$^3$. Heat transfer coefficient between steel and air $\lambda = 30$ W / (m$^2$ · K). Bar dimensions are the following: L = 0,1 m, $a_1$ = 0,1 m, $a_2$ = 0,03 m. The pressing force P is 150 N, friction coefficient is 0,55, the speed $v$ = 1 m / s.

In Fig. 2, temperature versus time at the center T (L, t) and at the edges T (0, t) of the bar is shown.

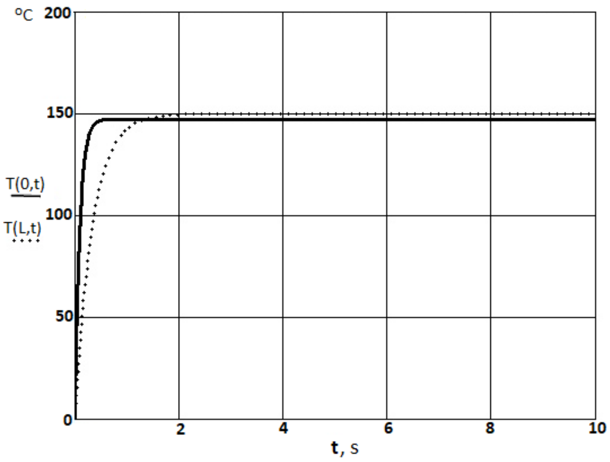


**Figure 2.** Temperature in the center of the bar (dotted line) and at the edges of the bar (solid line) versus time for steel C22

The graph shows that the temperature change is maximum at the beginning of the simulation, and the temperature graphs have approximately the same shape both in the center and at the edges of the bar.

## 3. EXPERIMENTAL RESEARCH

A set of experiments with samples made of steel C20E2C was carried out for the verification of the simulation.

The experiment was carried out with the use of a friction machine MTU-1 [13, 14]. The scheme of the experiment was plate-on-plate, the rotational speed was 150 rpm, the loading force was 150 N. Friction coefficient obtained experimentally is 0.55.

The obtained results were used for the simulation. The simulation results are presented in Fig. 3.

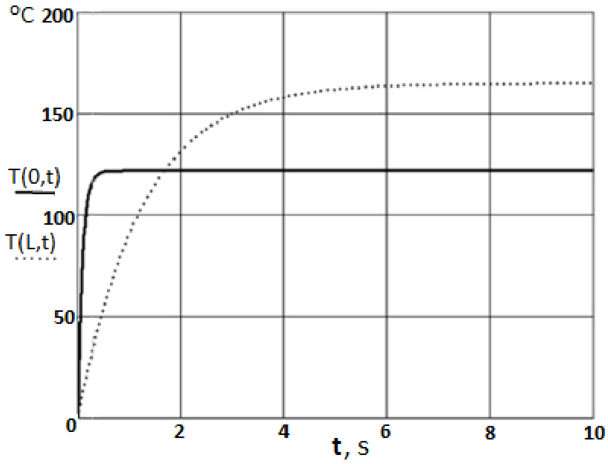


**Figure 3.** Temperature in the center of the bar (dotted line) and at the edges of the bar (solid line) versus time for steel C20E2C

In the center of the bar there is a real contact area where the interaction occurs with minimal loss of energy. The graph in figure 3 shows that the temperature changes in that area smoothly, which is probably due to energy dissipation to mechanical surface destruction, as well as to the occurrence and destruction of molecular bridges in tribopairs of similar materials.

The graph of temperature at the edges of the sample has a sharp jump at the beginning of the experiment, which is explained by the initial stage of running-in. There is no temperature flash in the contact zone. The absence of extremum is due to the fact that the model offers not a point distribution, but an integral distribution of the temperature field.

## 4. THERMOELASTIC PROCESSES

When moving the bar along the base (Fig. 1), we assume that the stress state of the bar is uniaxial, which means that the bar works for compression and tension. The equation for a straight rod with a combined thermomechanical action under tension-compression is well known [15, 16]:

$$N' + f = \rho \cdot \ddot{u}; \quad N = E_1(u' - \alpha T); \tag{9}$$

$$N(-L) = 0; N(L) = f, \tag{10}$$

where N is the force directed along the x axis, is the transverse load, u (*x, t*) is deflection, is the density of the bar, is the stiffness of the bar for tension, is the coefficient of linear thermal expansion, is the temperature field distribution.

Since the load changes slowly and smoothly, the inertia forces can be neglected. In this regard, we obtain a quasistatic solution of the equation (9):

$$N(x) = -\int_L^x f dx; \quad u = -\int_x^L \left( \frac{N}{E_1} + \alpha T \right) dx. \tag{11}$$

To obtain the results of simulation, we substitute to the equation (11) the following data: tension stiffness $E_1 = 6 \cdot 10^8$ N, linear thermal expansion coefficient $\alpha = 1.1 \cdot 10^{-5}$ $K^{-1}$. The distribution of the temperature field T (*x*) = const = 90 $^0$C (Fig. 2, 3).

The simulation results are presented in Fig. 4.

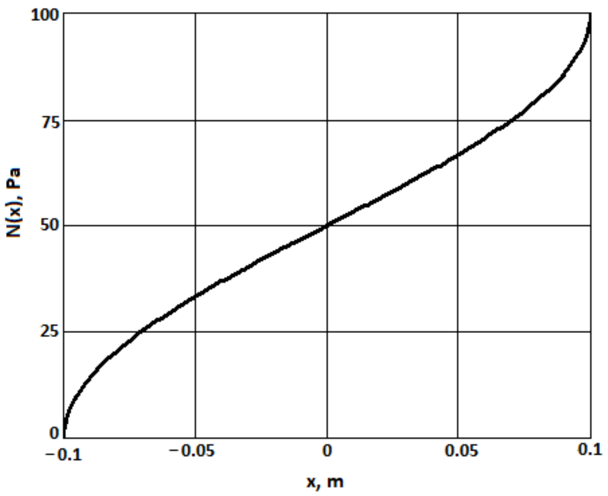


**Figure 4.** Stress distribution in the bar

It can be noticed from the graph in Fig. 4 that the deviation from the linear distribution is observed at the ends of the bar due to the presence of excessive contact pressure and, probably, elastic-plastic deformation at the edges of the bar.

## 5. CONCLUSION

When comparing fig. 2 and fig. 3, it can be concluded that the model can be used for simulation of the temperature distribution in the contact area during dry friction of metallic materials without taking into consideration the flash temperature rise. The difference between the results in figure 2 and figure 3 could be explained by the difference in the conditions of the experiments and initial data. The shape of the graphs proves that the model takes into consideration energy dissipation and temperature distribution in the contact area. The model uses characteristics of real metallic materials, such as heat capacity and thermal conductivity, and allow us to predict temperature distribution not only on the surface of the contacting bodies, but also in their volume.

The model for stress-strain state of the bar can be used for estimation of the influence of the frictional heating on the deformation of the elements of tribopairs during dry friction. The presence of elastic-plastic deformation at the edges of the bar can partially compensate the linear thermal expansion that occurs in the process of frictional heating during dry friction.

## REFERENCES


[1] F.E. Kennedy: Frictional Heating and Contact Temperatures, in: B. Bhushan (Ed.): *Modern Tribology Handbook, Volume One: Principles of Tribology*, CRC Press LLC, 1997.

[2] S. Boutechebak, A. Azeb Ahmed: Analysis of a Dynamic Frictional Contact Problem for Thermo-Elastic-Viscoplastic Materials with Damage, Gen. Math. Notes, Vol. 19, No. 1, pp.60-77, 2013.

[3] P. Zagrodzki, K. B. Lam, E. Al Bahkali, J. R. Barber: Nonlinear Transient Behavior of a Sliding System with Frictionally Excited Thermoelastic Instability, Journal of Tribology, Vol. 123, pp. 699-708, 2001.

[4] B. Mostefa, B. Ali: Thermoelastic Analysis of Disk Brakes Rotor, American Journal of

Mechanical Engineering, Vol. 2, No. 4, pp. 103-113, 2014.
[5] A.E. Tyurin, G.M. Ismailov, E.V. Beloenko, A.V. Baranov: Monitoring vibrations and microdisplacement for "pin on disc" tribology studies, Journal of Physics: Conference Series, Vol. 803, No. 1, paper 012167, 2017.
[6] G.M. Ismailov, A.E. Tyurin: Experimental study of frictional vibrations under dry friction conditions, Tribology in Industry, Vol. 40, No. 3, pp. 392-400, 2018.
[7] Z.F. Baczynski: Dynamic thermoelastic processes in microperiodic composites. Journal of Thermal Stresses, Vol. 26, pp. 55-66, 2003.
[8] S.S. Kang, S.K. Cho: Thermal deformation and stress analysis of disk brakes by finite element method, Journal of Mechanical Science and Technology, Vol. 26, No. 7, pp. 2133-2137, 2012.
[9] V. Dygalo, I. Zhukov: The thermal loading estimation of the friction pairs of a vehicle automated brake system, IOP Conf. Series: Materials Science and Engineering, Vol. 386, paper 012012, 2018.
[10] Y. Rozhdestvensky, E. Zadorozhnaya: A simulation of the thermal state of heavily loaded tribo-units and its evaluation, Bulletin of the South Ural State University, Series: Mathematical Modelling, Programming and Computer Software, Vol. 7 No. 4, pp. 51-64, 2014.
[11] N. Benhassine, A. Haiahem, B. Bou-Said: A comparative study of the transient thermomechanical behavior of friction of the ceramic brake discs: Temperature field effect, Journal of Mechanical Science and Technology, Vol. 33, No. 1, pp. 233–240, 2019.
[12] O.I. Abdullah, J. Schlattmann: Finite Element Analysis of Temperature Field in Automotive Dry Friction Clutch, Tribology in Industry, Vol. 34, No. 4, pp. 206-216, 2012.
[13] S. Perepelkina, P. Kovalenko, R. Pechenko: Investigation of Tribological Properties of Metallic Materials with the Use of the Universal Friction Machine “MTU-1", Procedia Engineering, Vol. 176, pp. 301-309, 2017.
[14] P. Kovalenko, S. Perepelkina, T. Korakhanov: Investigation of tribological properties of friction pairs duralumin – fluoropolymer used for design and manufacturing of biomechatronic devices, Tribology in Industry, Vol. 39 No. 2, pp. 192-197, 2017.
[15] V.V. Eliseev: *Mechanics of elastic bodies,* SPbPU, Saint Petersburg, 2003 (in Russian).
[16] V.V. Eliseev: *Mechanics of a solid deformable body*, SPbPU, Saint Petersburg, 2006 (in Russian).