

FINITE ELEMENT SIMULATION OF HEAT GENERATION IN THE BRAKE UNIT OF HEAVY-DUTY TRIBOSYSTEMS

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In the presented research, the heat distribution on the contact surfaces of the brake system of a drilling rig was studied using thermal simulation. The friction temperature distribution model on contact surfaces was obtained using the finite element method in COMSOL Multiphysics 5.5. As a prototype, the mobile drilling rig designed for drilling wells with a depth of up to 3000 m was included in the model. The brake systems of drilling rigs usually use a steel disc and a pad made of a composite material. In this study, a pair of steel and a new asbestos-free organic composite was selected. Based on the selected parameters, the thermal model was used to determine the temperature distribution mechanism along the contact surface of the disc and the brake pad. The highest friction temperature was observed during the first 2 – 4 sec of the braking cycle.

Keywords: finite element method, thermal model, drilling rig, friction composites, temperature distribution

Friction heat is one of the main factors determining the tribological characteristics of two-body sliding surfaces. The energy (95 – 100%) generated on the contact surface is converted into heat and affects the properties of the contacting materials. One of the devices operated under high-duty conditions is the friction unit of a drawworks. During the process of drilling wells, the drawworks performs several important tasks. These include operations such as adjusting the speed when lowering and withdrawing drill pipes [1]. During friction, the heat generated on the contact surfaces, as a result of heat transfer and radiation, is distributed between the pad and the disc and disseminates to the environment. Overheating of the disc and pad causes a sharp deterioration in tribological properties; periodic heating that occurs when lowering pipes into the well leads to a complete failure of the braking function [2]. Thus, simulation of heat release and distribution is important for a better analysis of the braking process.

A new asbestos-free organic composite material and a steel disc were chosen to develop a thermal model of the drilling rig's brake system. To simulate heat transfer in solids using the finite element method, COMSOL Multiphysics 5.5 software was used. For the thermal model, a brake pad made of a non-asbestos phenolic resin composite material and a brake disc made of steel 35 were used. The brake pad

material was made using barite, phenolic resin, wollastonite, tin, lead, silicon dioxide, copper-graphite particles, aluminium oxide, magnesium oxide, molybdenum disulphide and brass chippings (CuZn30) based on powder metallurgy methods. The composition of the developed composites is presented in the Table below, wt.%:

Barite	25
Phenol formaldehyde	25
Copper-graphite particles, wollastonite, magnesium oxide	14
Aluminium oxide, silicon dioxide	12
Lead, tin	20
Molybdenum disulphide	2
Brass chippings	2

To improve the thermal conductivity properties of the samples, mainly rectangular brass chippings with dimensions of 150 – 200 μm were used; the particle size of the remaining components to form a stable contact surface was 50 μm (Fig. 1).

The properties of the braking elements for thermal simulation are presented in Table 1. Properties of the air used (27°C) are as follows: density of 1.170 kg/m^3 , specific heat capacity of 1100.0 $\text{J}/(\text{kg}\cdot^\circ\text{C})$, thermal conductivity of 0.026 $\text{W}/(\text{m}\cdot^\circ\text{C})$. The prototype used was a ZJ30 mobile drilling rig, which is used for short-term cyclic drilling. With the use of drill pipes with a diameter of 127 mm on the ZJ30 facility, the nominal drilling depth is 2000 – 3000 m.

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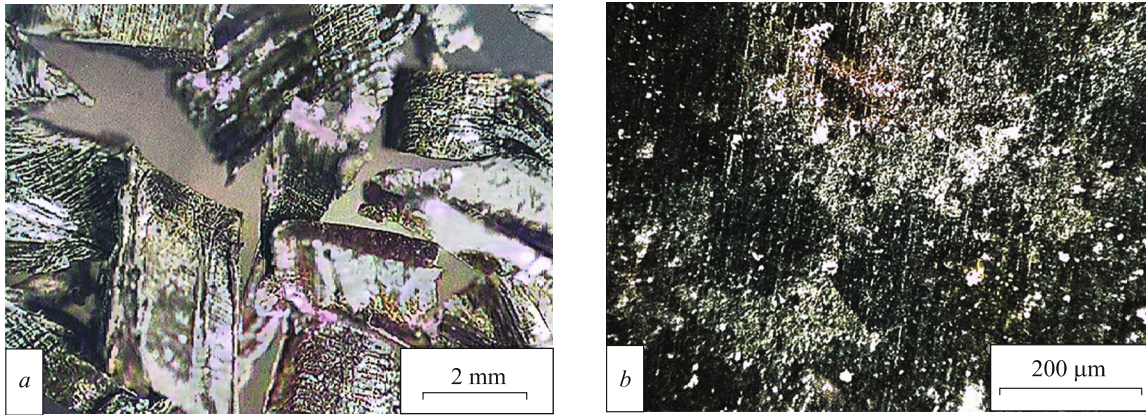


Fig. 1. Optical images of brass chippings (a) and contact surface of the brake pad composite (b).

TABLE 1. Properties of Braking Element Materials

Braking element	Density, kg/m ³	Specific heat capacity, J/(kg·°C)	Thermal conductivity, W/(m·°C)	Surface emissivity
Disc (steel 35)	7840.0	502.416	42	0.28
Pad (phenol formaldehyde)	2247.0	850.0	0.08	0.8

To reduce the simulation time, when designing the model, the scale was taken to be approximately 3.5 times smaller than the actual dimensions, but general proportionality was kept. In order not to reduce accuracy when the scale is reduced, the grid sensitivity parameter was increased and predominantly triangular shaped elements were used. Thus, the grid volume of the disc is 0.001578 m³, and that of the brake pad is 5.776×10^{-5} m³. The grid model and general information about simulation parameters are as follows: minimum element quality of 0.5048, average element quality of 0.8241, prism of 24800, triangle of 8046, quadrilateral of 1770, edge element of 808, vertex element of 40, spatial dimension of 3, number of domains of 5, number of boundaries of 30, number of edges of 60, number of vertices of 40.

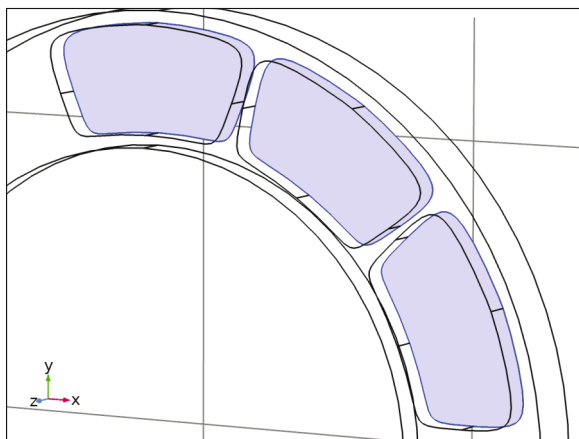


Fig. 2. 3D model of contact surfaces of brake pairs.

The calculation time was 59 min 40 sec, the time step was 0.015 sec. The main brake system of the ZJ30 mobile drilling rig consists of a disc and three pads. The contact model of the brake disc and pads is presented in Fig. 2. The calculated force on the device cable is 1500 kN, the maximum force is 1800 kN. The maximum disc rotation speed is 450 rpm. During the heating phase, the inner surface of the brake disc is exposed to heat flow. In the free parts of the surface and in the internal parts, according to Newton's law, heat exchange with the environment occurs [3]. At that, the ambient temperature is also taken into account.

The friction coefficient is set as 0.36. At the stage of frictional sliding of the brake at a speed of 10 m/s, the nature of the temperature change of the disc and pad with heat release was analysed. Analytical data showing this process are as follows: expression $d(v(t), t)$, argument t (sec function m/sec).

To solve the thermal problem using the finite element method in COMSOL Multiphysics 5.5 software, the following steps were used: physical processes, such as heat transfer inside a solid body, heat flow, heat transfer during thermal contact, and heat radiation from surfaces contacting with the body surface. Environment was included in the preliminary analysis. Accordingly, the material was chosen, the model grid and 3D model were built, and the load and speed parameters were introduced. At the final stage, the validity of the solution was verified and compared.

To analyse the distribution of heat released on the friction surface, the differential heat transfer equation for a cylindrical coordinate system was used:

$$\frac{\partial T_1}{\partial t} = a \left(\frac{\partial^2 T_1}{\partial r^2} + \frac{1}{r} \frac{\partial T_1}{\partial r} \right) \quad 0 \leq t \leq t_t \quad (1)$$

where T_1 is the initial temperature, °C; a is the temperature diffusivity, m/sec²; r is the radius from the disc centre to the friction path, m; t_t is the braking time, sec.

Heat transfer inside solid bodies is solved using the following formula:

$$\rho C_p (\partial T / \partial t + u_{\text{trans}} \nabla T) + \nabla \cdot (q + q_r) = Q + Q_{\text{ted}}, \quad (2)$$

where ρ is the density, kg/m³; C_p is the specific heat capacity, J/(kg·°C); T is the absolute temperature, °C; u_{trans} is the velocity vector in the moving motion, m/s; q is the heat flow due to conductivity, W/m²; q_r is the heat flow due to radiation, W/m²; Q is the additional heat source, W/m³; Q_{ted} is the thermoelastic effects in solid bodies, W/m³.

The heat flow vector is related to the temperature field by Fourier law

$$q = -k \nabla T, \quad (3)$$

where k is the thermal conductivity of the material, W/(m·°C); ∇T is the temperature gradient, °C/m.

Expressions for the heat flow arising from frictional contact can be written as follows:

$$\begin{aligned} -nq_d &= -h(T_u - T_d) + rQ_b, \\ -nq_u &= -h(T_d - T_u) + (1 - r)Q_b \end{aligned} \quad (4)$$

where n is the normal vector on the boundary; q_d is the downward heat flow, W/m²; T_u is the temperature of the upper part, °C; T_d is the temperature of the lower part, °C; Q_b is the boundary heat source, W/m²; q_u is the upward heat flow, W/m².

Thermal radiation from contact surfaces into the environment is determined by the following formula:

$$-nq = \varepsilon \sigma (T_{\text{amb}}^4 - T^4) \quad (5)$$

where ε is the surface emissivity coefficient; σ is the Stefan – Boltzmann constant, W/(m²·°C⁴); T_{amb} is the ambient temperature, °C.

Heat generation during the drilling process mainly occurs after the first drill pipes are lowered and the main brake is activated. The brake is then used to adjust the speed at which the pipe is lowered into the well. The brake mechanism is introduced to the model in the form of a function (Fig. 3).

The number of degrees of freedom (DOF) for the thermal model was calculated as 110866 (plus 53846 internal degrees of freedom). The result of constructing a thermal model for the specified mechanism in 1.85 sec is visualized and presented in Fig. 4, in which one can observe the regularity of the friction heat distribution over the contact surface of the

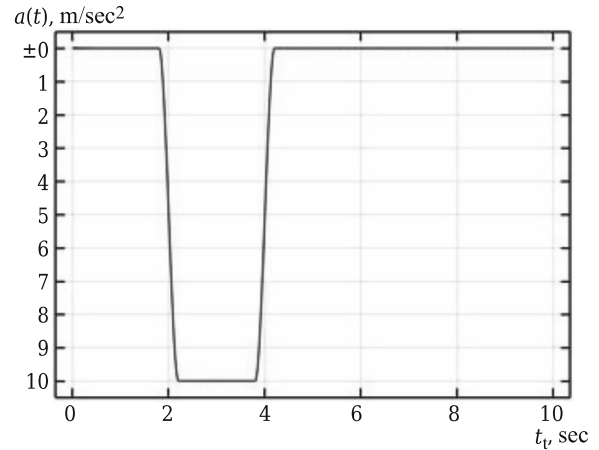


Fig. 3. Functional description of the braking process in the model.

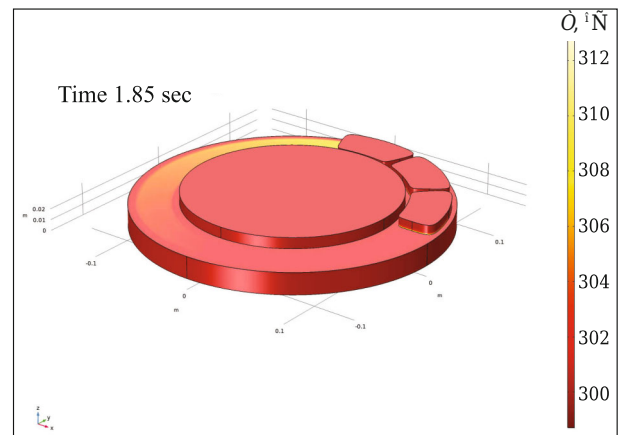


Fig. 4. Temperature distribution T on the contact surface at the initial stage of braking.

friction pair materials during braking depending on time. After approximately 1.70 sec, an increase in temperature along the disc surface was observed. The resulting model showed that heat is directed in the opposite direction from the intersection surface of both friction elements into the material (see Fig. 4).

The reason for the faster spread of heat inside the brake pad compared to the disc is due to the higher thermal properties of the disc material. After approximately 5 sec, the temperature on the pad surface began to change to approximately 720°C. In general, the highest temperature was detected in the first 2 – 4 sec. Thus, after 3.85 sec the temperature was 1250°C. After the braking process is completed, the disc surface temperature decreases, and after 10 sec it was 350°C. In addition, the temperature was lower in areas located near the central part of the pulley compared to the peripheral parts. In general, the temperature distribution in the thermal model can be divided into three phases: the initial

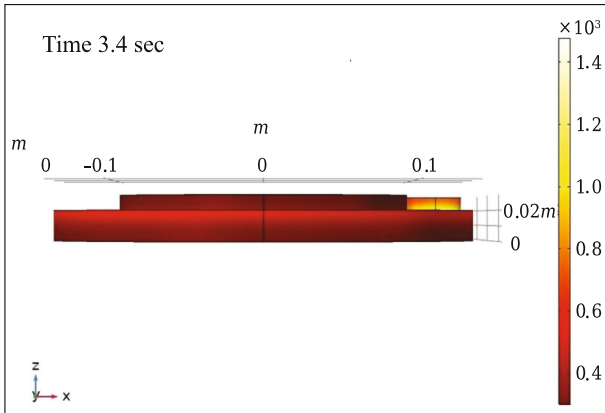


Fig. 5. Temperature distribution in different directions along the steel disc surface.

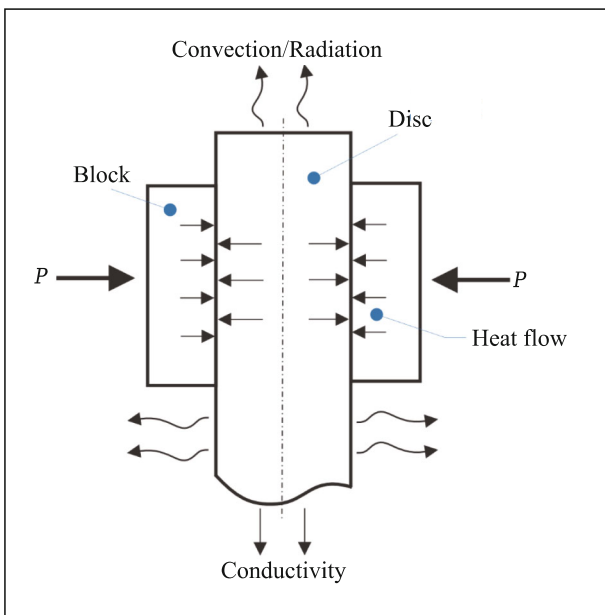


Fig. 6. Heat transfer mechanism in the disc/block brake system.

phase before braking, the braking phase, and the final phase after braking.

One can see that the temperature in parts outside the contact surface was low and increased slowly. The temperature in these areas in the initial phase varied from 30 to 120°C. Figure 5 shows the heat distribution on the contact surface over a time of 3.45 sec. In this phase, the temperature has already increased to 1200°C. The temperature outside the contact surface increases during braking and gradually decreases at a later stage. After braking, the temperature of this part was close to 500°C. In the thermal model, the overall temperature increase started at 2 sec, reached a maximum within 5 sec, and then partially decreased. At this moment, the temperature increase continued to 1110°C.

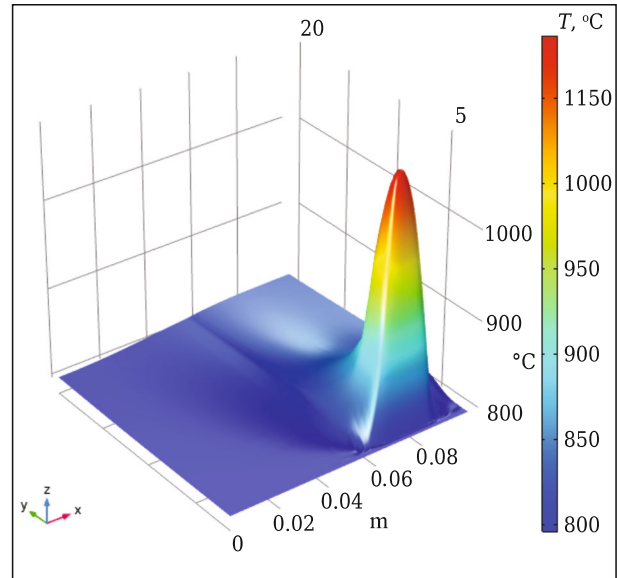


Fig. 7. Heating profile of the contact surface of the disc brake.

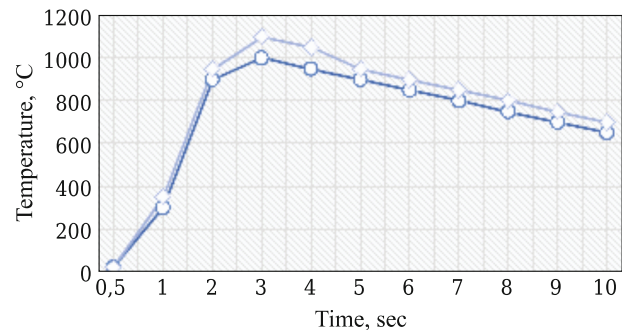


Fig. 8. Comparison of the results of measurements of the contact surface temperature obtained in this study (O) and given in [5, 6] (◊).

Disc/block brakes are widely used in heavy-duty tribological systems (drilling rigs and other moderately- or heavy-duty braking systems). Such brakes consist of levers and two or three pads located diametrically opposite to the disc. Braking occurs due to the friction force between the disc and the pad. When a moving disc is compressed on both sides according to Fourier's law [4] (Fig. 6), the processes of radiation, convection and conduction occur.

The resulting temperature profile describes the temperature distribution over the disc surface during and after braking (Fig. 7).

During braking, the temperature rose to 1250°C, and then gradually decreased. Verification of results is important in finite element method. To ensure that the temperature values are correct, the numerical results of thermal simulation were compared with the experimental ones presented in the scientific literature (Fig. 8). As one can see, the results of measuring the temperature of the contact surface are very

close to the literature data and, accordingly, confirm the accuracy of the thermal model.

CONCLUSION

Non-asbestos organic composite materials have been developed for heavy-duty friction units and the mechanisms of temperature distribution on contact surfaces have been studied using the finite element method. The results have shown that the temperature increased in 2 sec, the initial temperature reached a maximum in 5 sec, and then began to partially decrease. The maximum temperature on the contact surfaces was 1250°C for 3.85 sec. At the end of the simulation process, the temperature along the disc surface was 350°C.

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