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THERMAL ANALYSIS OF THE INDUSTRIAL SHOE BRAKES TO REDUCE THE RISK OF FAILURE DURING BRAKING

The aim of the study was to develop an assessment methodology for the temperature of the surface of the friction pair during the braking for mine hoists. During the braking process, the work of friction is transformed into heat at the level of friction surfaces, and in case high temperatures are reached, the friction coefficient is influenced negatively, thus the risk of braking failure exists. In the first part of the study we measured the temperature of the friction surfaces for a particular case of hoist in real braking conditions. In the second part of the study is presented a theoretical model for the calculation of the temperatures resulted in the braking process for the hoist equipped with shoe brakes. The theoretical model for calculation was simulated numerically for a particular case in real braking conditions. Based on the conclusions resulted after the study, a series of hypotheses and recommendations for adjusting the control of the process parameters have been given out, in order to avoid the excessive heating of the brakes of the hoists and, respectively, their improved safety, maintenance and availability.

Keywords: temperature, braking, friction surfaces, risk of failure, mining hoist

1. Introduction

A series of aspects referring to the operation of the braking systems, such as the wear of the brake linings, the vibrations and the heating of the brakes are of interest from the point of view of reliability and maintenance (Ambikaprasad & Abhijeet, 2015; Dragomir, 2014; Wolny, 2016; Legutko, 2010). From these, the most difficult to control is the heat flux density produced by heating the braking surfaces during the service braking or during the emergency braking (Ścieszka, 2013; Tudor, 2005; Zhen, 2013).

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The accumulated heat can also have a negative influence on the physical and mechanical properties of the materials of the friction pair (Sethupathi, 2015; Lan, 2012; Puncioiu, 2015; Trzepieciński, 2015).

The heating size of the braking surfaces of the hoists brakes depends on the physical properties of the materials of the friction pair, on the construction of the subassembly of the brake and on the process operational parameters. About 95% of the heat produced on the friction surfaces is taken over by the material of the drum brake and only 5% is taken by the material of the brake lining (Banciu, 1993; Belobrov, 1981).

In case of using the service braking, the brake heating is not produced continuously up to a maximum value, but is interpolated with break periods, namely cooling periods, corresponding to the period till the next braking (Belobrov, 1981; Zhu, 2013).

2. Measuring the temperature of the friction surface

The temperature of the friction surface of the drum brake was measured in real working conditions. The measurements have been done with the thermometer TMTL 1400K, at the mining hoist 2T3,5X1,7A, located at Mining Enterprise from Baia Sprie, for the maximum unbalanced load and for four running velocities, in two conditions: in case of service braking and in case of emergency braking. The main technical characteristics of the mining hoist 2T3,5X1,7A are presented in Table 1. The methodology for collecting and processing of experimental data complied with the basic principles: repeating experiments, implicitly measuring at least 7 times, eliminating aberrant data, statistical processing of data etc. The location of the measuring points was at the periphery of the brake drum.

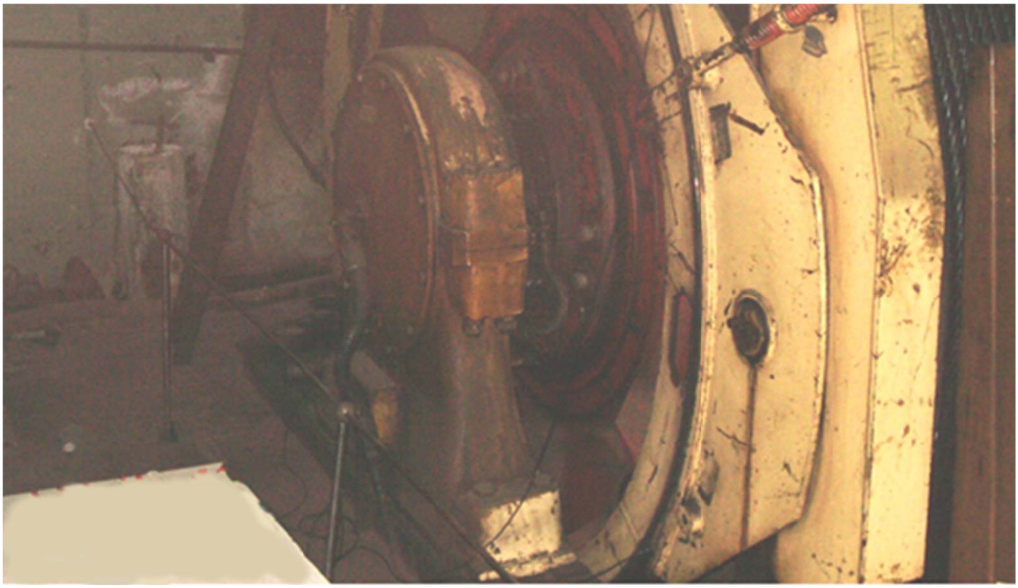

Fig. 1. Brake shoe of a mining hoist 2T3,5X1,7A

TABLE 1

Technical data (Banciu, 1993)

Max. static tension F_{st} [N]	Max. differential static tension $\Delta F_{st.}$ [N]	Conveyances	Coiling Width [m]	The maximum hoisting speed [m/s]	Type of Winder	Type of Brake
150000	125000	2 Cages	500	12	2 Double Drums	Double brake shoe system

The brake of the mining hoist 2T3,5X1,7A is of type with two exterior shoes with pneumatic operation (Fig. 1). The drum brakes are manufactured from steel and have a diameter of 2.9 meters. The shoes are lined on the working surface with brake linings type SFGM (Banciu, 1993).

The results of measuring the temperature (average value) of the breaking surface for service braking and for emergency braking are presented in tables 2 and 3.

TABLE 2

Service braking

No.	Operational parameters			Temperature [°C]
	$\Delta F_{st.}$ [N]	v_0 [m/s]	t_f [s]	
1	125000	10.5	40	38
2	125000	10	40	37
3	125000	9.5	38	34
4	125000	8.5	42	31
5	125000	8	40	30
6	125000	5.6	39	28

TABLE 3

Emergency braking

No.	Operational parameters			Temperature [°C]
	$\Delta F_{st.}$ [N]	v_0 [m/s]	t_f [s]	
1	125000	10.5	2.6	105
2	125000	10	2.5	99
3	125000	9.5	2.4	62
4	125000	8.5	1.8	55
5	125000	8	2	46
6	125000	5.6	2.4	41

Notations used:

$\Delta F_{st.}$ — is the difference between the static tensions of the two ropes;

v_0 — velocity at the beginning of the braking;

t_f — duration of the braking.

Based on the measurements carried out we have built the graphs for the temperature variation depending on the braking time and depending on the initial velocity presented in the figures 2 and 3.

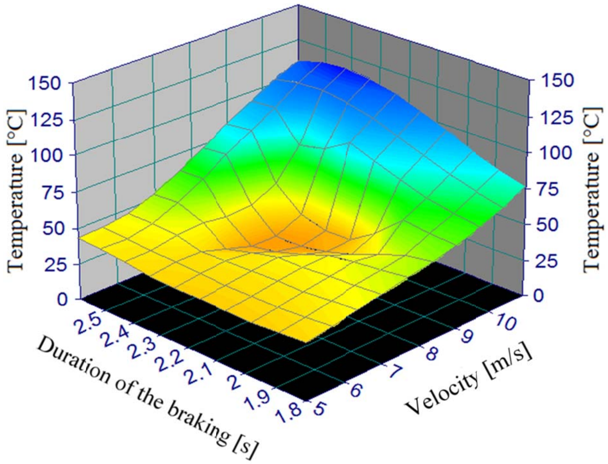

Fig. 2. Temperature variation at the friction surface for the emergency braking

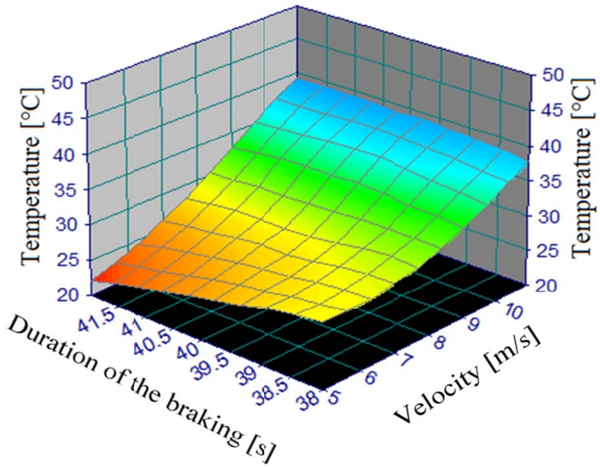

Fig. 3. Temperature variation at the friction surface for the service braking

From the analysis of the graphs for the temperature variation depending on the initial velocity of the cage and the braking duration, one can notice the fact that the temperature is directly proportional to the velocity and inversely proportional to the braking duration. One can also notice the fact that the temperature resulted over the braking duration does not reach high speeds during the service braking to influence the structure of the material or the stability of the friction coefficient (Sethupathi, 2015; Belobrov, 1981; Monkova, 2017). But one can notice that the temperature reaches high values that influence the stability of the friction coefficient in case of emergency braking (Ungureanu, 2005).

Based on these conclusions the study further focused on the analysis of the thermal condition in case of emergency braking.

3. Establishing a theoretical model for temperature field analysis

The following hypotheses are accepted for the research of the heating regime of the friction coupling:

- the mode of transmission of the heat resulted from friction in case of stopping braking is conduction (convention and radiation can be neglected);
- the resulted heat quantity is dissipated only in the brake friction pair (Bocîi, 2011);
- the transmission of the heat from the friction surface is done only in perpendicular direction on it (Puncioiu, 2015);
- the elements of the friction pair are bodies of finite dimensions (Tawanda, 2017);
- the transmission of the heat from the drum brake to the cable coiling component is not taken into account;
- the transmission of the heat from the brake lining to the metallic shoes is not taken into consideration;
- all the energy dissipated when braking is transformed into heat at the level of the friction surfaces;
- as a result of the processes of heat accumulation and transfer a domain of temperatures will be established in the friction pair.

3.1. Determination of the energy resulted at braking

The energy that turns into friction mechanical work and, respectively, into heat at a certain moment is the difference between the kinetic energy and the potential energy of the system (Fig. 4).

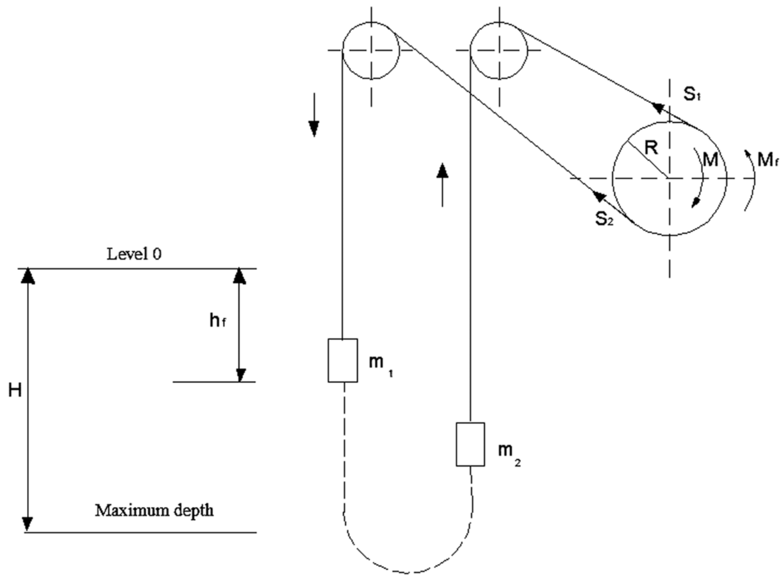

Fig. 4. Hoisting installation. Diagram for the determination of the spreader energy in the braking process

The resulted braking energy is:

$$\Delta E = \frac{\Delta Fst.}{g} \cdot \frac{v^2}{2} + m_{rot.red.} \cdot \frac{v^2}{2} - \Delta Fst. \cdot h_f - \Delta F_r \cdot h_f \tag{1}$$

where:

- $\Delta F_{st.}$ — is the difference between the static tensions of the two ropes [N];
- ΔF_r — is the difference of the resistance force at the movement of the cages along the shaft [N];
- g — acceleration of gravity [m/s^2];
- v — velocity at the beginning of braking [m/s];
- $m_{rot.red.}$ — reduced mass at the drum periphery of all rotating elements [kg];
- h_f — the distance covered by the traveling of the cages over braking duration [m].

The difference between the static tensions is calculated with equation (2) for mining hoists not equipped with balance rope and with equation (3) for mining hoists that use the balance rope (Belobrov, 1981).

$$\Delta F_{st} = \left[Q_u + q \cdot \left(H - 2h_f\right)\right] \cdot g \tag{2}$$

$$\Delta F_{st} = \left[\cdot Q_u + \left(q - q_e\right) \cdot \left(H - 2h_f\right)\right] \cdot g \tag{3}$$

where:

- Q_u — is the mass of the vehicle corresponding to the maximum load [kg];
- q — is the mass of the rope on linear meter [kg/m];
- q_e — the mass of the equilibration rope on linear meter [kg/m];
- H — maximum traveling distance of the cages along the shaft [m];
- h_f — the distance covered by the cages over the braking duration [m];
- g — acceleration of gravity [m/s^2].

In these conditions the heat flux density is:

$$q_0 = \frac{\Delta E}{S \cdot t_f} \tag{4}$$

where S is the contact surface between the braking shoe and the drum brake and t_f is the braking duration.

The heat flux density increases with the growth of the difference of the static loads.

3.2. The temperature of friction surfaces for the pair brake lining – drum brake

In the hypothesis that over the contact surface the two elements of the friction pair have the same temperature, the temperature of the friction surface in relation to the temperature of the environment over the braking duration t_f(Bocîi, 2011):

$$\Delta T\left(t \prec t_f, 0\right) = \frac{2 \cdot \dfrac{\Delta E}{S \cdot t_f} \cdot \sqrt{t} \cdot \left(1 - \dfrac{2}{3} \cdot \dfrac{t}{t_f}\right)}{\sqrt{\pi} \cdot \sqrt{\lambda_j \cdot \rho_j \cdot c_j} \cdot \left(1 + \dfrac{1}{f_{F_0}} \cdot \dfrac{\sqrt{\lambda_s \cdot \rho_s \cdot c_s}}{\sqrt{\lambda_j \cdot \rho_j \cdot c_j}}\right)} \tag{5}$$

where:

ΔE	—	the energy resulted at braking [J];
t_f	—	braking duration [s];
S	—	surface of brake lining [mm^2];
f_{F_0}	—	correction factor depending on the Fourier factor F_0;
λ_j	—	thermal conductivity of the drum brakes, [W·m^{-1}·°C^{-1}]
λ_s	—	thermal conductivity of the brake lining, [W·m^{-1}·°C^{-1}]
ρ_j	—	density of the drum brakes, [kg·m^{-3}]
ρ_s	—	density of the brake lining [kg·m^{-3}]
c_j	—	specific heat capacity of the drum brake, [J·kg^{-1}·°C^{-1}]
c_s	—	specific heat capacity of the brake lining, [J·kg^{-1}·°C^{-1}]

In the moment when $t = t_f$, the process of producing the heat finishes and the process of cooling begins.

In the hypothesis that in the layer limit of contact the brake lining and the drum brakes, have the same temperature for the case of cooling ($t \succ t_f$) the temperature variation of the braking surfaces after the end of the braking process is calculated with the equation:

$$\Delta T_r = \Delta T\left(t \succ t_f, 0\right) = \frac{2 \cdot \dfrac{\Delta E}{S \cdot t_f}\left[\sqrt{t} \cdot \left(1 - \dfrac{2}{3} \cdot \dfrac{t}{t_f}\right) - \dfrac{2}{3} \cdot \sqrt{t - t_f} \cdot \left(1 - \dfrac{t}{t_f}\right)\right]}{\cdot\sqrt{\pi} \cdot \sqrt{\lambda_J \cdot \rho_J \cdot c_J} \cdot \left(1 + \dfrac{1}{f_{F_{ok}}} \cdot \dfrac{\sqrt{\lambda_s \cdot \rho_s \cdot c_s}}{\sqrt{\lambda_J \cdot \rho_J \cdot c_J}}\right)} \tag{6}$$

It is important to know the time after which the coupling temperature reaches the environment temperature in order to avoid another braking in this period. If another braking takes place in this period, in this case the temperature of the couple is not calculated in relation to the environment temperature but in relation with the coupling temperature in the moment when the braking begins.

3.3. Method and numerical simulation

3.3.1. Description of the method

Using the software MathCAD we have determined the numeric values for surfaces temperatures of the braking coupling for different operating conditions. In a first stage the heat flux density is determined with the method presented in paragraph 3.1 and then the temperature of the shoe-drum brake contact surfaces is calculated with the formulas presented in paragraph 3.2. The basic data for brake are presented in table 4.

The numerical simulation was carried out for 5 speeds of the cages at the beginning of the braking ($v_1 = 8$ m/s, $v_2 = 9$ m/s, $v_3 = 10$ m/s, $v_4 = 11$ m/s, $v_5 = 12$ m/s) and for three durations of the braking. (t_f = 1,8 s, t_f = 2,1 s, t_f = 3 s). Therefore 5 variation curves resulted for the temperatures corresponding to the five velocities, they have been noted as follows: $T1(t)$ for v_1, $T2(t)$ for v_2, $T3(t)$ for v_3, $T4(t)$ for v_4, $T5(t)$ for v_5.

The temperatures corresponding to the cooling of the three operating velocities also resulted, as follows: $TR1(t)$ for v_1, $TR2(t)$ for v_2, $TR3(t)$ for v_3, $TR4(t)$ for v_4, $TR5(t)$ for v_5. In the hypothesis that the physical properties of the braking pair materials are constant with the variation of temperature.

TABLE 4

Basic data for brake

Parameter	Value
Mass of vehicle Q_u [kg]	6000
Reduced mass at the drum periphery of all rotating elements $m_{rot.red.}$ [kg]	52544
Maximum traveling distance of the cages along the shaft [m]	550
Distance covered by the cages during the braking [m]	8
Mass of rope on linear meter q [kg/m]	6.69
Thermal conductivity of the drum brakes, λ_j [W·m^{-1}·°C^{-1}]	49.8
Thermal conductivity of the brake lining, λ_s [W·m^{-1}·°C^{-1}]	0.561
Density of the drum brakes, ρ_j [kg·m^{-3}]	7840
Density of the brake lining, ρ_s [kg·m^{-3}]	1850
Specific heat capacity of the drum brake, c_j [J·kg^{-1}·°C^{-1}]	465
Specific heat capacity of the brake lining, c_s [J·kg^{-1}·°C^{-1}]	893

3.3.2. Numerical simulations

Numerical simulation for t_f = 1,8 s: after the simulation resulted the graphs represented in figures 5 and 6.

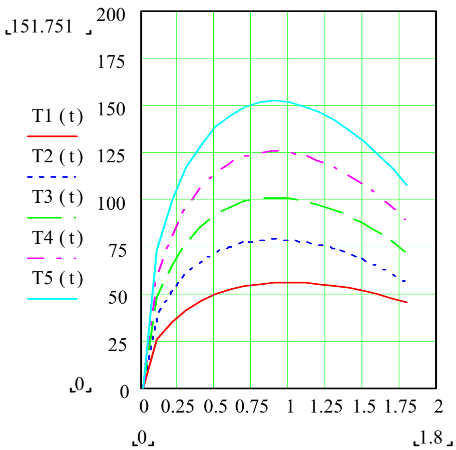

Fig. 5. Variation of temperatures T [°C] according to time t [s]

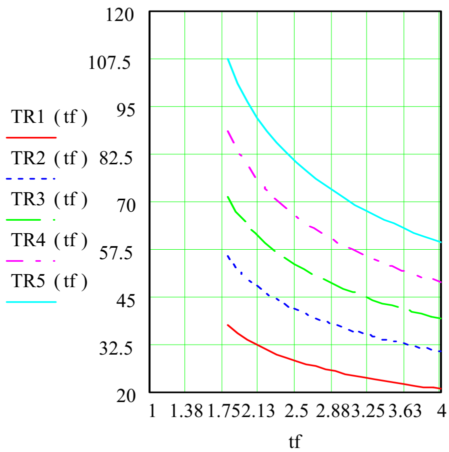

Fig. 6. Temperatures TR [°C] in case of cooling according to time t_f [s]

In figure 5 we observe that the temperature $T1(t)$ corresponding to the minimum speed v_1 reaches a maximum value of 52°C and the temperature $T5(t)$ corresponding to the minimum velocity v_5 reaches a maximum value of 151°C. During the cooling period (Fig. 6), after 4 seconds from finishing of braking at minimum speed v_1, the temperature drops to 20°C and at maximum speed v_5 the temperature decreases to 58°C.

Numerical simulation for t_f = 2,1 s: after the simulation resulted the graphs represented in figures 7 and 8.

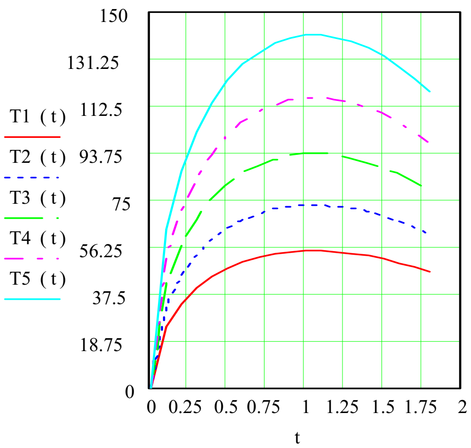

Fig. 7. Variation of temperatures T [°C] according to time t [s]

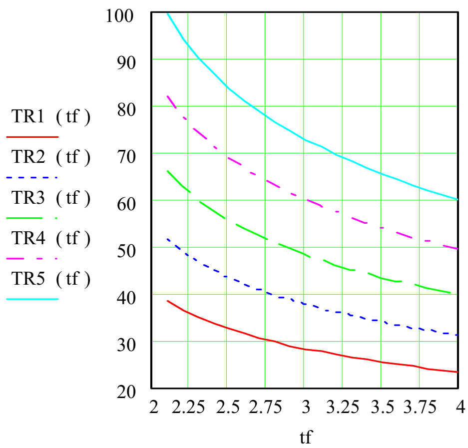

Fig. 8. Temperatures TR [°C] in case of cooling according to time t_f [s]

In figure 7 we observe that the temperature $T1(t)$ corresponding to the minimum speed v_1 reaches a maximum value of 56°C and the temperature $T5(t)$ corresponding to the minimum velocity v_5 reaches a maximum value of 140°C. During the cooling period (Fig. 8), after 4 seconds from finishing of braking at minimum speed v_1, the temperature drops to 24°C and at maximum speed v_5 the temperature decreases to 60°C.

Numerical simulation for t_f = 3 s: after the simulation resulted the graphs represented in figures 9 and 10.

In figure 9 we observe that the temperature $T1(t)$ corresponding to the minimum speed v_1 reaches a maximum value of 40°C and the temperature $T5(t)$ corresponding to the minimum velocity v_5 reaches a maximum value of 115°C. During the cooling period (Fig. 10), after 4 seconds from finishing of braking at minimum speed v_1, the temperature drops to 24°C and at maximum speed v_5 the temperature decreases to 64°C.

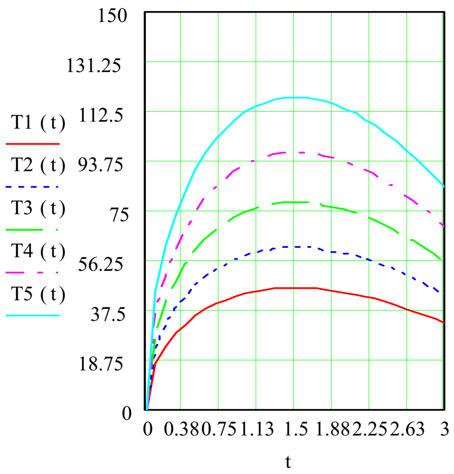

Fig. 9. Variation of temperatures T [ºC] according to time t [s]

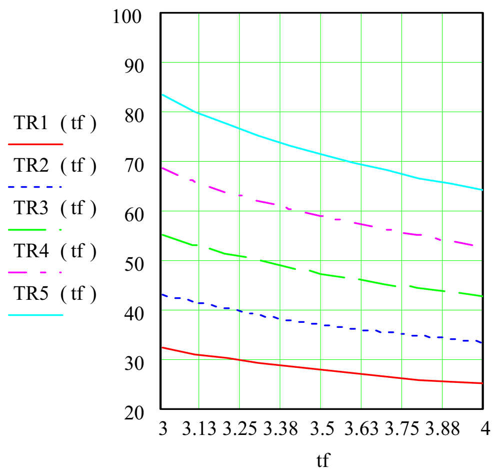

Fig. 10. Temperatures TR [ºC] in case of cooling according to time t_f [s]

4. Conclusion

The study of the thermal regime of the friction pair is important for the analysis of the safe operation of the braking system. If the maximum temperatures are known, it is possible to analyse to what extent the properties of the materials of the friction pair and the friction coefficient are affected. Following the numerical simulation of the thermal regime for the friction pair brake lining – drum brake of the mining hoist and following the study of the research carried out in the domain, the following conclusions resulted: by analysing the graphs presented in sub-chapter 3.3.2 one can point out the variation of temperature in time over the braking duration and after the termination of the braking process (the cooling period) according to different operational parameters in conditions of operation in industrial environment; one can notice that the maximum temperature lasts a short time and it is recorded halfway of the braking process; the temperature increases with the growth of the transport velocity; the growth of the braking duration determines the decrease of temperatures.

If we compare the measured temperatures to the theoretical ones, we observe that the values are closed. The measured temperatures are a bit smaller than the maximum calculated temperatures, which can be explained by the fact that the thermometer manifests thermic inertia and the braking process has a short braking duration in the case of the emergency brake, and the maximum temperatures are instant.

The resulted temperatures, singularly analyzed, do not modify the structure of the materials which form the friction pair, but taking into consideration the fact that the phenomena that are produced in the area of the friction pair during the braking period are complex demonstrates the need for this study. Researches in this domain, have highlighted the fact that the presence of the friction force, alongside of the thermal stress, lead to a negative consequences on the contact surface.

The braking process, respectively the mechanic and thermal contact is a dynamic process and with the aim to study the thermal process, a method of determination of the heat quantity has been put forward. A method whcih also includes the specific thermal flow, the area of temperatures for the friction pair, taking into consideration the entire duration of the braking and the variation of the main process parameters.

Recommendations

In order to avoid the safety break failure risk because of the decrease of the friction coefficient among with the growth of temperature it is recommend to analyze the pair of the friction break through the method presented in this paper. According to the operational parameters the temperature that results after the braking will be calculated. For this temperature, the value of the braking coefficient will be checked (value given by the producer of the brake linings). According to the friction coefficient, the moment of brake will be recomputed and if this value is below the necessary limit must be take some measures for decreasing the pair friction temperature.

From the point of view of application in practice we recommend to decrease the difference of the static loads by using the balance rope, to decrease the circulation velocity of the hoisting cages and to decrease the deceleration (within acceptable limits) in order to increase the braking duration, but in such a way that the maximum braking space admitted for the mine hoist should not be exceeded.

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