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Research concerning thermal stress in case of stop and duration braking of electric locomotives EA-060 type

Mircea A. DUNGAN, Ioana IONEL, Luisa I. DUNGAN

Abstract - Generally, kinetic energy in the locomotive braking is converted into heat that is transmitted to the bandage. Because of overloading of the bandaged wheels the phenomena of bandages weakening combined with tread defects appears. This danger exists especially when train brakes on long slopes and with high gradients. The decisive effect on the frictional and wear characteristics of the brake system is exerted by temperature generated at friction.

The aim of the paper - Determine by calculation the maximum temperature incurred during braking on friction members, both for new and for used bandages for the case of stop braking and for the case of duration braking.

Keywords - braking process, thermal stress, friction coefficient, wheels with bandages.

I. INTRODUCTION

In the nineteenth century, following the development of industrial production, needs raw materials and called for the creation of vehicles with higher capacity, powerful, reliable in operation and providing increased speed.

As in any other field, safety comes first, whether it is passenger or, equally, on freight transportation.

From the early days of rail transport has significant emphasis on improving quality, and, last but not least, the safety.

Being in fierce competition, continuous and long-term with other means of transport, modern rail transport has an important share in most countries worldwide with an increasing trend in traffic speeds, hauled tonnages and offered competitive costs.

Continuous growth of velocity on the railway imposed special security issues regarding guiding rail safety in general. **2.1**

One of the problems in the operation of railways **2.2**
vehicles equipped with wheels with bandages and brake blocks is the appearance of bandages rotating on disk [1].

To determine the causes leading to these spins there was examined the aspect of the heating through thermal calculation for the bandages in case of stopping braking.

To ensure necessary braking space becomes a problem more difficult with increasing speed of movement, which is explained by the fact that with the axle speed increase, the coefficient of friction of cast iron brake blocks suddenly shrinks and pressure on the blocks is limited by the potential wheel lock. Also, on increasing the normal force push, it is increased the wear degree and the danger of turning blocks of bandages on the centre of the wheel will become bigger.

II. BRAKING WITH BRAKE SHOES FOR RAIL VEHICLES

Rail vehicles are characterized by the movement on the railways through wheels auto guided by contact forces between wheels and rails.

Braking facility holds the most important role in ensuring the movement of railway vehicles in safety conditions for the traffic.

During braking friction, the largest share of energy is dissipated as heat absorbed by friction members which are thus subjected to an thermomechanical intense. In the field of railway braking for example, brakes disk are brought considerable energy to dissipate in a few tens of seconds, several megajoules for a single disc, or even tens of megajoules in the case of high speed.

The severity of such stresses leads to localization phenomena thermal interface, which manifest themselves in different forms, such as the hot bands migrants, or macroscopic hot spots, depending on the severity of braking, temperature reaching locally extreme values.

The impact of these phenomena on durability of the braking is of course crucial, and studies show that thermo mechanical approaches alone are not sufficient to describe these phenomena, it is necessary to consider the phenomena of friction and wear at the interface.

A General information

Brake shoe is best used on rolling stock materials. Wheel is required first by the driving forces and secondly by braking forces [2]. For the second category of claim was necessary to set temperature limits and wear. At automatic brake shoes, force which manifest to the brake cylinder piston rod is amplified by the wheelhouse of brake components and sent to break blocks which press the wheel tread.

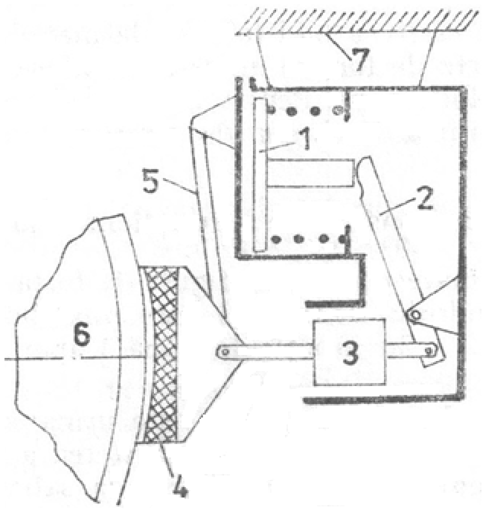

Fig.1 Braking shoes unit's scheme:
1-piston; 2- lever amplifiers; 3- automatic regulator wheelhouse; 4- brake shoes; 5- padlock; 6- wheel; 7- bogie frame [1]

Brake shoe is the main body of which depends largely braking effect.

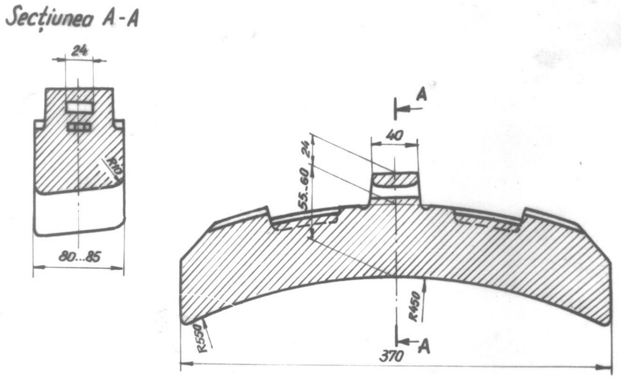

Fig.2 Brake shoe [4]

On experimental basis has been established that the optimal length of break shoes are 320 mm for easy mounting in shoe-wear and 250 mm when fitted to double shoe-wear. Width of the break shoes in all cases is 80 mm.

Quality brake characterized by the length of braking road depends among other things by the shape, sizes and quality of the material being manufactured brake shoe. The influence of these factors is contained in the amount of friction's coefficient between shoe and wheel. Figure 3 shows schematically the formation of the braking force on the action of the brake shoe pressure, where P_s is the pressure on the shoe, Q is the load on wheel rail, F_a normal contact force on the point of contact, μ_s representing the coefficient of friction for the case of wheel and brake blocks, and R radius of the wheel. After executing a braking action, the kinetic energy of the locomotive (and possibly potential energy - flow on the slope) is consumed by the work of the drag forces and braking forces, leading to slowdown. If cast iron brake shoes classic (Fig. 3) can be regarded as the work of the entire braking forces are fully transformed into heat, which is practically part of the wheel over, another part by shoes and some (negligible) is collected by metal particles which are removed from the contact surfaces [6].

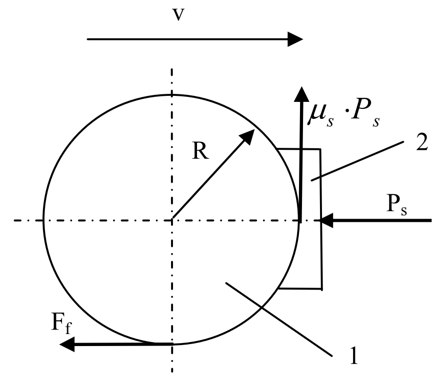

Fig. 3 Braking with brake shoes: 1 – wheel; 2- brake shoe

$$F_f = \mu_s \cdot P_s$$
$$F_{f\,\max} = F_a = \mu_a \cdot Q \qquad (1)$$

The distribution of heat between wheel and brake pads is directly dependent thermophysical properties of materials that are built [5]. It was found both in the theoretical calculations and experimental determination that most of the developed heat is taken from the wheel so, in the wheel can occur significant increases of temperature. Both temperature values reached in the wheel and their distribution during braking is of particular importance because it was found that high temperatures that develop thermal stresses rise to very high. In extreme cases (very dangerous situations in terms of traffic safety) can lead to weakening or spins of the bandage's wheel or wheel on the wheel axle.

As is known, braking system is necessary for:

• stopping the train (rail vehicle) within the limits of braking space;
• partial reduction of speed;
• lowering maintenance train on the slopes;
• train (railway vehicle) immobilization after stop.

In the first two cases, inside the process of breaking it is dissipated the kinetic energy stored in the train's speed. In the third case, the potential energy is dissipated, which is stored on boarding ramps and in the latter case the role of the brake is related with preventing movements that could be caused by external factors.

One of the chief factors conditioning stress state of the frictional material in a brake lining is the friction temperature gradient [7].

B Establishing the computational relations for brake shoes for braking stop

At brake stopping, of a thermal point of view, heat transmission occurs in non-stationary process, the amount of heat changing over time. In this case, heat transmission is calculated using Fourier's differential equation (2) and for simplicity it was considered that the heat is transmitted only in a direction perpendicular to the running surface:

$$\frac{v(\Delta v)}{v_t} = a \cdot \frac{v^2(\Delta v)}{v \cdot x^2} \ . \tag{2}$$

Starting from this relationship, for stop break it was obtained, for the raising of temperature Δv, the relationship:

$$\Delta v = \frac{q}{\sqrt{\pi} \cdot \sqrt{\lambda \rho g c} \cdot \sqrt{t_b}} \cdot e^{\frac{-x^2}{4at_b}} , \tag{3}$$

where:

λ (coefficient of heat transmission), (specific mass in kg / m), c (specific heat in J / kg degree C) and a (temperature index) are quantities that depend on the material it is made pad and bandage [3], [4], [5];

q - thermal load;

t_b - breaking time;

x – distance from the braking surface to the interior of the bandage.

By integrating (3) as a function of t and x it results relation (4) for determining the temperature increase, for braking stop:

$$\Delta v(x,t) = \frac{2q_a\sqrt{t} \cdot e^{\frac{-x^2}{4at}}}{\sqrt{\pi} \cdot \sqrt{\lambda \rho g c}} \cdot \left[1 - \frac{2}{3} \cdot \frac{t}{t_b}\left(1 + \frac{x^2}{4at}\right)\right] - \frac{q_a x}{\lambda} \cdot \left[1 - \Phi\left(\frac{x}{\sqrt{4at}}\right)\right] \cdot \left[1 - \frac{t}{t_b}\left(1 + \frac{2x^2}{12at}\right)\right] \tag{4}$$

To get rolling surface temperature rise in (4), substitute x = 0, resulting equation 5.

$$\Delta v(0,t) = \frac{2q_a\sqrt{t}}{\sqrt{\pi} \cdot \sqrt{\lambda \cdot \gamma \cdot c}} \cdot \left[1 - \frac{2}{3} \cdot \frac{t}{t_b}\right] \tag{5}$$

By differentiating this equation we obtained the time during the rolling surface temperature is maximum. This time is:

$$t = \frac{t_b}{2} \tag{6}$$

Replacing this time in (5) result temperature at the surface of the running, namely:

$$\Delta v(0, \max) = \frac{\sqrt{8} \cdot q_a \cdot \sqrt{t_b}}{3 \cdot \sqrt{\pi} \cdot \sqrt{\lambda \cdot \gamma \cdot c}} \tag{7}$$

From the above it seems that the maximum temperature is achieved not at the end of braking process but at the mid range.

The temperature at the end of braking will be:

$$\Delta v(0,e) = \frac{2 \cdot q_a \cdot \sqrt{t_b}}{3 \cdot \sqrt{\pi} \cdot \sqrt{\lambda \cdot \gamma \cdot c}} \tag{8}$$

III. TEMPERATURE VARIATION CALCULATIONS INSIDE THE BANDAGE FOR BRAKING STOP

One of the cases frequently encountered in practice is the braking stop and the most disadvantageous type of braking is rapid braking from maximum speed to zero. Braking deceleration on the road range varies with speed because of friction shoe-binding and binding-track.

To simplify calculations it is considered that the deceleration is uniform throughout the brake. In the calculations were admitted the following:

- for deceleration: $\dot{v}$=1.3 m/s^2 ;
- for wheel diameter: d_R=1250 m (new bandage) and d_R=1170 m (maximum used bandage);
- brake shoe width: δ_R =8 cm;
- brake shoe thick: b_k=1 cm;
- coefficient of heat transmission: λ_R =37;
- specific gravity: γ_R =4850 Kp/m^3 ;
- specific heat: C_R=0.11;
- coefficient of heat transmission: λ_k =45;
- specific gravity: γ_k =7250 Kp/m^3 ;
- specific heat: C_k=0.13;
- distance from the road surface: x_R=0 cm;
- environment temperature: 20°C;
- braking time (t): from 0 to 60 s.;
- circulating speed : v=120 km/h;
- mass factor: ξ =1.2;
- wheel load: G_R=10 MPa.

Temperature variation in bandage, brake off, was calculated in the following situations:

- new bandage, used brake shoes, deceleration of 1.3 m/s^2
- used bandage, used brake shoes, deceleration of 1.3 m/s^2

Given that all of the heat is distributed between the wheel and block, the amount of heat will received at the wheel will be:

$$q_{aR} = \frac{q_a}{1 + \frac{1}{f_{0k}} \cdot \frac{\sqrt{\lambda_k \rho_k c_k}}{\sqrt{\lambda_R \rho_R c_R}}} \ . \tag{9}$$

A. The case: new bandage, used brake shoes, deceleration of 1.3 m/s^2

The necessary preliminary values in (4) are given in Table I.

Table I. Preliminary values

Crt. no.	Name	Notation	Calculating relation	Value
1	Braking time	t_b [s]	$t_b = \frac{v}{v^{\bullet}}$	25.6
2	Wheel braking surface	F_{OR} [m^2]	$F_{OR}=\pi \cdot d_R \cdot \delta_k$	0.31415
3	Thermal loading	q_a [kJ/m^{2}h]	$q_a = \frac{\xi_M \cdot G_R \cdot v^2}{q \cdot F_{OR} \cdot t_b}$	5962972
4	Fourier coefficient for the lock	F_{OK}	$F_{OK} = \frac{a \cdot t_b}{b^2}$	3.4
5	Correction factor	f_{FOK}	-	1.6

Variation of temperature during braking bandage according to the distance from the braking surface toward bandage interior is given in Figures 4 and 5.

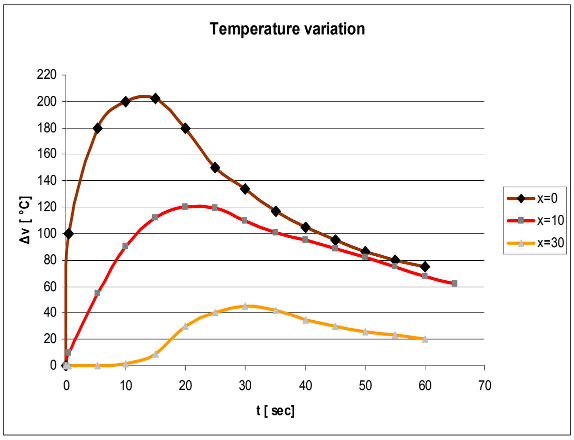

Fig. 4Temperature variation for the bandage as a function of braking time

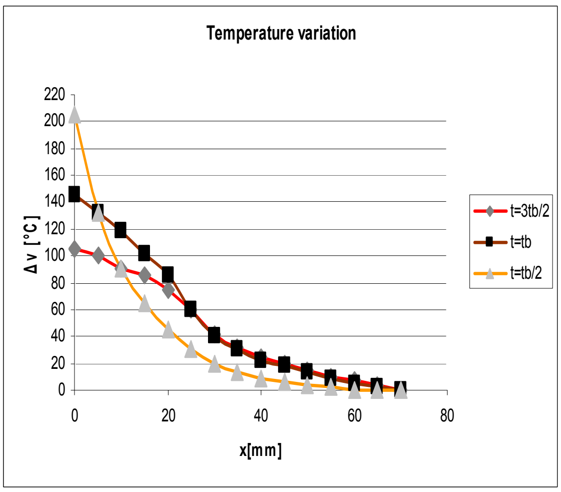

Fig. 5 Temperature variation for the bandage as a function of the distance from the braking surface toward bandage interior

B. The case: maximum used bandage, used brake shoes, deceleration of 1.3 m/s^2

Preliminary values needed in (4) are all those from Table I, only the thermal load has a value of q_a = 561 736 kJ/m^{2}h.

Temperature variation inside the bandage as a function of the duration of braking and distance from the braking surface toward bandage interior is given in Figures 6 and 7.

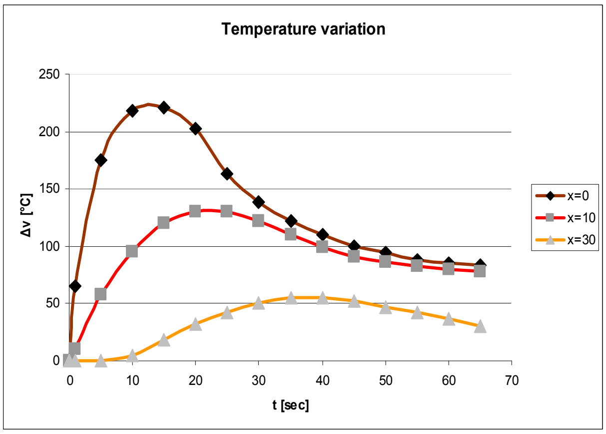

Fig. 6 Temperature variation for the bandage as a function of braking time

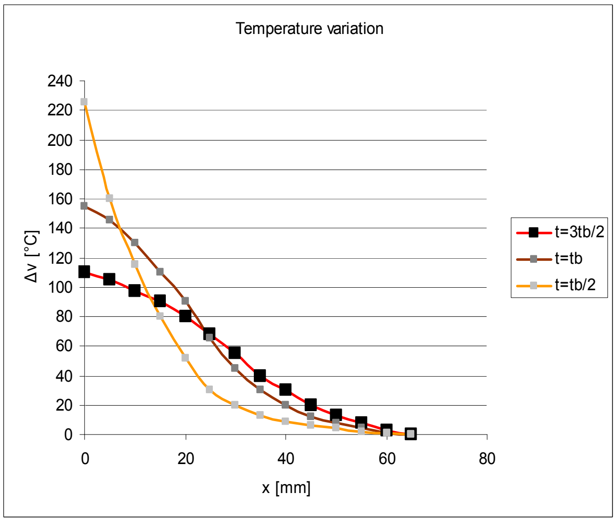

Fig. 7 Temperature variation for the bandage as a function of the distance from the braking surface toward bandage interior

From assessment roll surface temperature variation is observed that the maximum temperatures do not get to the end but to the middle of braking, that at time $t = \frac{t_b}{2}$.

From calculation obtained that the maximum temperatures in this case is 244 °C, and according to what is stated in the literature these values are lower temperature than temperature production of surface defects on rolling.

IV. TEMPERATURE VARIATION CALCULATIONS INSIDE THE BANDAGE FOR DURATION BRAKING

While in operation locomotive EA-060 type, besides stop braking occurs very frequently the case of duration braking on long slopes with different declivities at different times. In case of these braking can not be neglected the heat transfer.

To determine the relationship between temperature and power, we consider some simplifications, namely:

- brake has the same temperature everywhere Δv;
- material constants and the heat transmission is considered invariant throughout the temperature range.

Braking duration has decisive influence on the ability of brakes, because in a short braking time can store a larger quantity of heat than in case with a larger duration.

In the unit time, in the brake storing the CγVd(dv) heat quantity and by convection and radiation is released αFΔv heat quantity, total heat quantity will be:

$$Q = C\gamma v \frac{d(dv)}{dt} + \alpha F \Delta v \tag{10}$$

This equation, written as the differential equation and solved for boundary conditions, has the following form for increase of the maximum temperature:

$$\Delta v_{\max} = \frac{Q}{\alpha F}\left(1 - e^{\frac{-\alpha F t_b}{C\gamma v}}\right) \tag{11}$$

In reality, for the practical use of this equation must take into account the temperature in different parts of the wheel.

In this respect, it is sufficient to divide the wheel into temperature areas to reach values that correspond to those measured in practice. We divided the wheel in six surface elements and six volume corresponding elements.

To assess the distribution of temperature on the wheel is inserted temperature factor " f " defined as the ratio of average temperatures at the depth "x" and average temperature on the friction surface (x=0).

$$f = \frac{\Delta v_{ijm}}{\Delta v_{\mathrm{Im}}} \tag{12}$$

Influence the transmission of heat by convection and radiation at the surface of friction is taken into consideration in that they assumed that the friction surface area per unit quantity of heat $q_R + \alpha_I \Delta v_{omR}$ arises and is transmitted outside $\alpha_I \Delta v_{omR}$.

Temperature on friction surface can be calculated and for calculation of the "in depth" temperature (x>0), correction factor is used.

$$f_{korr} = \frac{\Delta v_{korr}}{\Delta v} = \frac{q_R + \alpha_I}{q_R} \tag{13}$$

where α_I has two components:

- convection component α_k
- radiation component α_s

Calculating the convective component, for surface friction and the displacement of the hub, will be made by the laws of heat transfer in a rotating cylinder and for the sides of the wheel with disc in rotation relationships. Laws of heat transfer by radiation does not depend on the geometrical shape of bodies.

Total heat transmission coefficient results as the sum of the convective part and the radiation part(for each surface element of the wheel) and has values given in Table II.

Table II. Values of total heat transmission coefficient

	New bandage	Used bandage
$\alpha_{kI} + \alpha_{sI}$	21.9	22.3
$\alpha_{kII} + \alpha_{sII}$	29	30.4
$\alpha_{kIII} + \alpha_{sIII}$	27.2	24.8
$\alpha_{kIV} + \alpha_{sIV}$	16.6	17.8
$\alpha_{kV} + \alpha_{sV}$	9.5	9.8
$\alpha_{kVI} + \alpha_{sVI}$	9.5	9.8

Due to the fact that on duration braking heat further into the wheel more than on stop braking cooling time will be considerably longer.

Heat transfer coefficients depend on how the vehicle moves and are particularly low if the vehicle is stationary and cooling only occurs naturally.

Like in stop braking case, in duration braking case heating of bandages calculation involves choosing of a running system in the slopes of some declivities and some time.

In our country, we meet the highest slopes on the route Predeal-Brasov and Balta-Dr. Turnu Severin.

For calculation, gradient descent is allowed on the route Balta-Dr.Turnu Severin, with slope average of 25‰ on a length of 8 km. Maximum allowable traffic speed is 50 km/h.

From the above data result duration of the descent of the slope as:

$$t = \frac{s}{v} = 0.16 \quad [\mathrm{h}] \tag{14}$$

For the study we admit more disadvantaged case lowering the locomotive on the slopes, namely: slope of 25‰; slope length 20 km which lowers with 50 km/h resulting the time t=0.4 h.

The preliminary values necessary in (15) are given in Table III.

Table III. Preliminary values

Crt. no.	Name	Notation	Value		
			New bandage		Used bandage
0	1	2	3		4
1	Braking time	t_b [h]	A	0.16	0.16
			B	0.4	0.4

0	1	2	3		4
2	Wheel braking surface	$\sqrt{\lambda_R \gamma_R C_R}$		178.74	178.74
		$\sqrt{\lambda_k \gamma_k C_k}$		205.94	205.94
3	Thermal loading	a_k [m^2/h]	0.0477		0.0477
4	Fourier coefficient for the lock	F_{OK}	A	76.32	76.32
			B	190.8	190.8
5	Correction factor	f_{FOK}	A	5.229	5.229
			B	8.204	8.204
6	Angular speed	ω [1/s]	22.22		23.74
7	Peripheral speed	u [m/s]	13.88		13.88
8	Brake shoe friction surface	[m^2]	0.088		0.088

To calculate the road surface temperature is used the following relationship :

$$\Delta v_{\mathrm{Im}axR} = \frac{Q \cdot \left(1 - e^{\frac{-\left(\sum f_i \alpha_i F_i\right) t_b}{\sum f_j \gamma_j C_j v_j}}\right)}{\left[\sum f_i \alpha_i F_i\right] \cdot \left[1 + \frac{1}{f_{Fok}} \frac{\sqrt{\lambda_k \gamma_k c_k}}{\sqrt{\lambda_R \gamma_R c_R}}\right]} \qquad (15)$$

where:
- Q is the equivalent thermal power brakes.
Between Q and power brakes exists the following relationship:

$$Q = 632 \cdot N ;$$

-N is the power brake of the locomotive which determine with relationship:

$$N = G_R \cdot \left(\sin \alpha' - \xi\right) \cdot V$$

and has the value: N=34.05 kW;
- $\sum f_i \alpha_i F_i$ si $\sum f_j \gamma_j C_j v_j$ are amounts of factors and have values given in Table IV.

Table IV

	$\sum f_i \alpha_i F_i$ [kJ/h*degree]	$\sum f_j \gamma_j C_j v_j$ [kJ/degree]
New bandage, braking time 0.16 h	118.44	93.66
New bandage, braking time 0.4h	132.25	110.07
Used bandage, braking time 0.16 h	117.85	48.13
Used bandage, braking time 0.4h	130.40	51.48

Using (15) has drawn curves of temperature variation depending on time for the following cases:

A. The case: new bandage, used brake shoes, braking time of 10 min

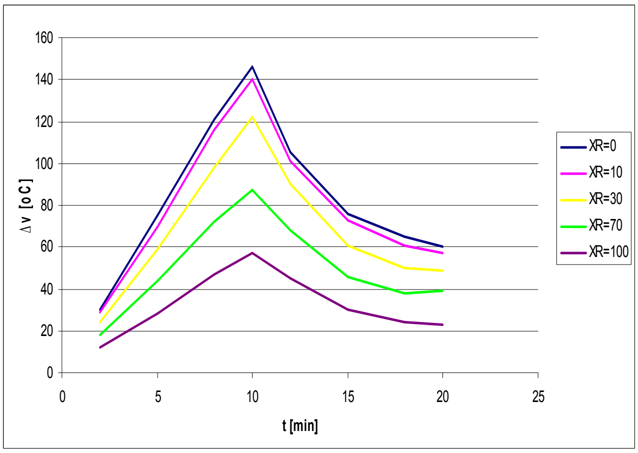

Fig. 8 Temperature variation for the new bandage, used brake shoes, braking time of 10 min

B. The case: maximum used bandage, used brake shoes, braking time of 10 min

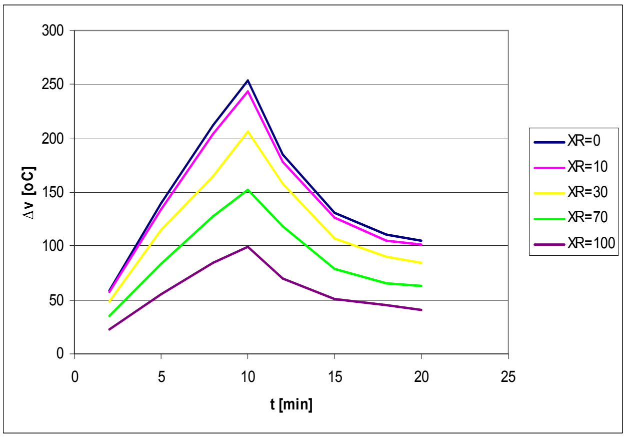

Fig. 9 Temperature variation for the maximum used bandage, used brake shoes, braking time of 10 min

C. The case: new bandage, new brake shoes, braking time of 24 min

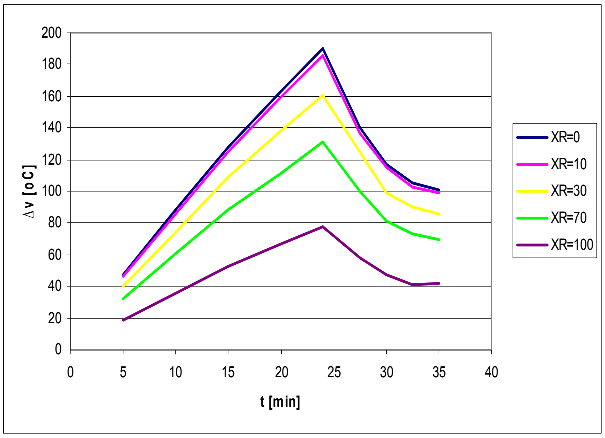

Fig.10 Temperature variation for the new bandage, new brake shoes, braking time of 24 min

D. The case: maximum used bandage, used brake shoes, braking time of 24 min

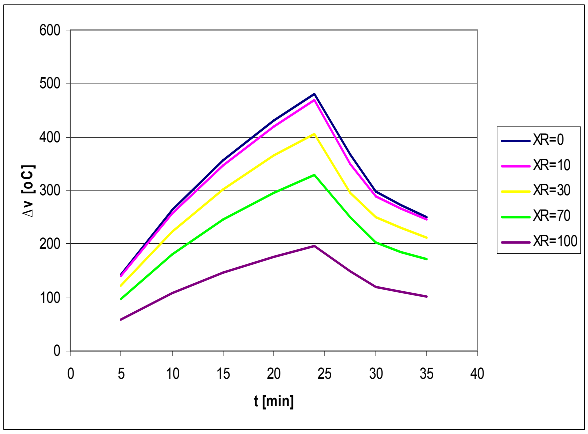

Fig.11 Temperature variation for the maximum used bandage, used brake shoes, braking time of 24 min

Using relation (15) has drawn curves of temperature variation depending on the distance to the bandage surface for the following cases:

A. The case: new bandage, used brake shoes

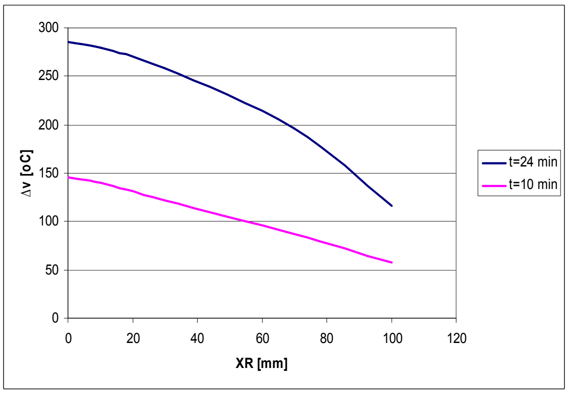

Fig.12 Temperature variation for the new bandage, used brake shoes

B. The case: maximum used bandage, used brake shoes

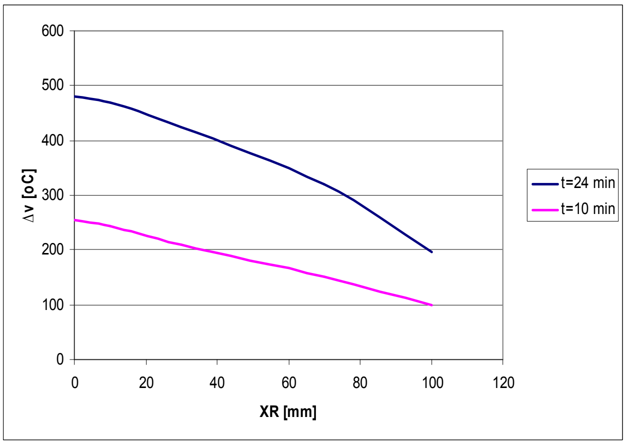

Fig.13 Temperature variation for the maximum used bandage, used brake shoes

V. CONCLUSIONS

Due to the short time acting braking power, temperature fluctuations occur at the beginning of the tread brake. Further, as approaching the stopping of the movement, braking power is reduced and, accordingly, variations in temperature are lowered.

In case of braking tread off at the bandage surface is obtained a maximum temperature of 244°C for second case- the worst case. This temperature creates a compression effort to tread of 48.5 daN/mm^2. Value of effort is far below the prescribed breaking effort for material bandages. Braking feature allows binding material to tread absolute temperature of 370 - 415°C without the appearance of defects on the tread.

During stop braking, in the disk-bandage joint, maximum temperature which is reached is that of the environment and so in this case there is no danger for bandage rotating on disk.

For duration braking, on the road surface, for new bandage and used brake shoes, maximum temperature is 166°C. In the disk-bandage joint area, for the same situation mention above, maximum temperature obtained is 108°C.

On the road surface, for used bandage and used brake shoes, maximum temperature is 274°C.

To descend a slope of 25‰ for 10 minutes with temperatures obtained is no danger for spin of the bandages. On the road surface, for new bandage and new brake shoes, maximum temperature is 210°C. In the disk-bandage joint area, for the same situation mention above, maximum temperature obtained is 151°C.

For new bandage and used brake shoes it was obtained the following temperatures:

- on the road surface 305°C;
- on the disk-bandage joint 216°C.

For maximum used bandage and used brake shoes it was obtained the following temperatures:

- on the road surface 506°C;
- on the disk-bandage joint 350°C.

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