TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics, Mechanics, and Engineering
Vol. 64, Issue I, March, 2021

INFLUENCE OF THE BRAKE DISC FRICTION SURFACE DIMENSION ON THE THERMAL REGIME OF THE DISC BRAKE OF HIGH-SPEED RAILWAY VEHICLES

Liviu Sevastian BOCÎI

Abstract: *One of the most important criteria for the use of a brake in the mechanical brake system on high-speed rail vehicles is the temperature of the brake friction elements (chok - wheel or brake disc - friction pad). The disc brake eliminates the action of the brake blocks on the wheel tread during the braking process and provides the possibility of greater energy dissipations during braking due to use of materials to make friction coupler components with optimum physical-mechanical characteristics. This work presents an analysis of the thermal regime (thermal flow and temperature) of the components of the disc brake friction coupler. Also to show the influence of the brake disc friction surface size on the thermal regime were considered four sizes of the brake disc friction surface (Sd = 0,2 m^2; Sd = 0,25 m^2, Sd = 0,3 m^2, Sd = 0,35 m^2). Thus, it was concluded that the use of unventilated brake disc leads to a lighter thermal regime (thermal flow and, consequently, lower temperature of the disc brake friction coupler components) regardless of the type of friction lining used.*

Key words: *brake disc; friction lining; thermal flow; temperature*

1. INTRODUCTION

The braking of railway vehicles from high speeds is by means of braking systems which are *dependant* (disc brake, rotary eddy current brake, etc.) on or *independant* of wheel adhesion — the rail (electromagnetic rail brake, linear eddy current brake, etc.). In order to make best use of the coefficient of adhesion, high-speed rail vehicles have been fitted with a combination of these braking systems, usually an adhesion-dependant brake (disc brake) and a brake whose action is independant of adhesion (electromagnetic brake on rail).

Regardless of the type of braking of the railway vehicle, its kinetic energy is dissipated by the braking system components (mechanical, electrical, fluid, etc.). In the case of high-speed vehicles where the main brake in most cases is part of the mechanical braking system, the conversion of their kinetic energy shall be carried out by means of the elements of the friction clutches as shown in the diagram in the following figure (1) [2]:

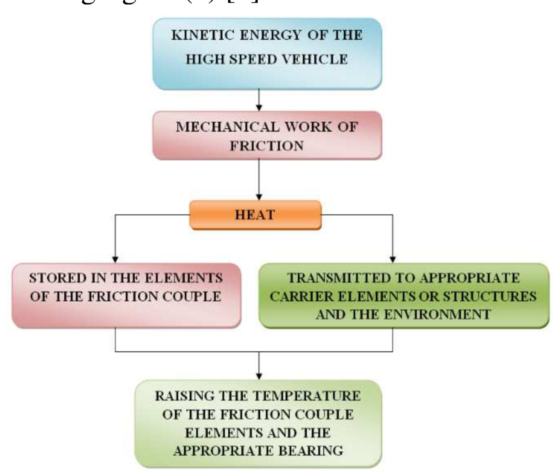

Fig. 1. The mode of kinetic energy dissipation of the railway vehicle

On a microscopic analysis of the surfaces of the components of the friction coupling, which come into direct contact, it has been found that they consist of "*micro-mountains*" (protrusions) and "*micro-valleys*" (recesses) (Figure 2):

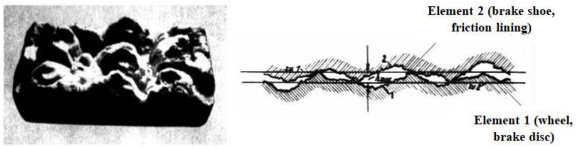

Fig. 2. The contact surface of the components of the friction coupling of the disc brake (microscopic analysis)

It follows that the kinetic energy of the vehicle is converted primarily into mechanical deformation work (elastic and plastic) of these lugs and finally into heat, thus increasing the temperature of the friction coupler components. It is also noted that the effective contact surface of both components is different from the nominal surface, i.e. only a part of it determined by the sum of the surfaces of the projections in contact (1):

$$S_e = S_1 + S_2 + ... + S_n = \sum_{i=1}^{n} S_i \quad (1)$$

Since high speed driving involves kinetic energy of such large vehicles, it is very important to determine the heat regime of the friction coupler components which come into direct contact and which are storage components and heat dissipators.

In the following figure (3) the contact area of two different solid bodies was shown on an enlarged scale as material highlighting the thermal flow lines.

It is noted that the effective contact surface is significantly lower than the nominal surface [13] [16-18], the heat transfer being almost entirely carried out through the projections of the two surfaces. The interstitions between contact areas have more or less regular forms, in which air is found in this case.

Thus it can be claimed that heat transfer between two solid bodies of different materials ($\lambda_1 \neq \lambda_2$) takes place through a combined mechanism, namely:

- conducting through actual contact areas;
- conduction through fluid (air) in interstitions;
- convection and radiation through this fluid.

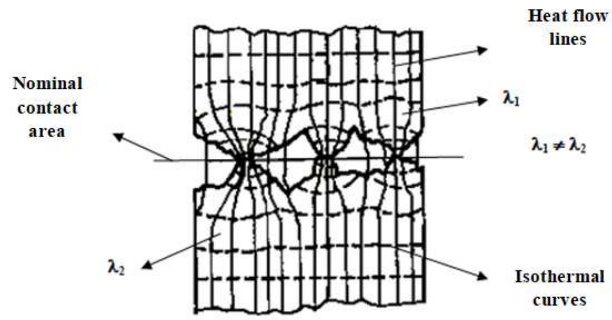

Fig. 3. The contact area of two solid bodies different in material

Due to the small size of the interstitions and the high contact pressure, the effect of convection in the air in them can be neglected. Also, the existence of a small temperature difference between the two surfaces causes a decrease in the share of thermal radiation. As a result, heat transfer between two solid contact bodies occurs mainly through conduction both through the effective contact areas and through the fluid in the interstitions.

The temperature of the friction elements of a brake is one of the important criteria for using this brake on high-speed rail vehicles.

Current trends in the design and construction of high-speed vehicles converge toward:

- The reduction of the vehicle mass as much as possible;
- An aerodynamic shape of the vehicle that does not require high power consumption for traction;
- Use of materials with mechanical and physico-chemical characteristics that ensure maximum operational safety and high reliability.

As the main functions of the drive wheel are *support* and *guidance* and because of the relatively high temperatures reached when braking at speeds above 160 km/h, the shoe brake is not suitable for use on high-speed rail vehicles as ***a main brake*** but only as ***an additional*** *(parking) brake.*

Reduction in the temperature of brake *disc friction coupler components — lining* can be achieved by:

- Increase the number of discs per axle;

- The use of high-configuration (non-ventilated) discs and low-density (aluminum-based alloys) or ceramic materials (see the figure 4);
- Use of the disc brake with a brake whose action is independent of the wheel-rail adhesion.

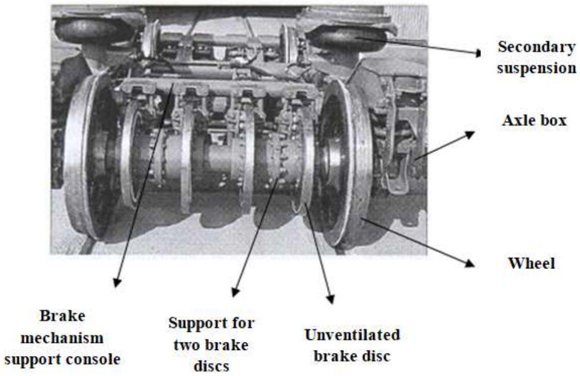

Fig. 4. *Location of unventilated brake discs on the a axle non-electric high-speed rail vehicle*

By increasing the number of self-centered cast iron discs placed on the axle, the unsuspended mass of the vehicle is increased with negative effects on the service life of the wheelset and indirectly on the reliability of the vehicle as a whole. By using unventilated disks (see figure 4) made of high elasticity alloy steel, lower temperatures are achieved compared to self-ovated discs and the considerable reduction in aerodynamic resistance due to the rotational movement of self-oventylated brake discs in traction mode (this resistance leads to a consumption of approximately 3 % of the installed power for traction, [2], [4], [9-11]).

2. DETERMINATION OF THE THERMAL REGIME OF THE DISC BRAKE

When determining the friction surface temperature of the brake disc/lining friction coupling elements, the following assumptions were taken into account:

- The vehicle speed varies linearly with the time when a stop is applied;
- The stopping times considered for the calculation are: tb = 30; 40; 50 and 60 seconds;
- The variation in brake power as a function of speed is determined by the peak of the diagram V= f(t);
- neglect of the influence of radiation and heat convection at stopping braking due to its short duration;
- Coefficient of friction between the brake disc and the lining as determined by Karwatsky's formula [12] for three lining contact forces: FB1 = 20 KN , FB2 = 45 KN , FB3 = 50 KN (force F_{b2} is the contact force in the experimental tests, [2]).

2.1 Determination of the unit thermal flow

The unit thermal flow (q) on the friction surface of the friction elements at linear velocity change with time (duration of braking) shall be determined with the relationship, [2-4]:

$$q = \frac{P_{mfd}}{S_d} \qquad [W/m^2] \tag{2}$$

where:
P_{mfd} – braking power from a brake disc, [W];
S_d – total brake disc friction surface, [m^2].
Considering that a stopping brake is carried out using only the disc brake, the braking power on the disc (P_{mfd}) is given by the following relationship (3) [2]:

$$P_{mf} = n_d \cdot P_{mfd} \Rightarrow P_{mfd} = \frac{(1+\gamma)\cdot m \cdot \left(\frac{V}{3,6}\right)^2}{2\cdot t_b \cdot n_d} - \frac{10}{t_b \cdot n_d}\cdot\left[\frac{3,93\cdot(1+\gamma)\cdot V^2}{g\cdot\mu_s\cdot\delta + r_t} + \frac{V\cdot t_u}{7,2}\right]\cdot \left(a + b\cdot V + c\cdot V^2\right) \quad [W] \tag{3}$$

By replacing the relationship (3) in (2), the following expression (3) is obtained for the unit heat flow q, [11]:

$$q = \frac{(1+\gamma)\cdot m\cdot\left(\frac{V(t)}{3,6}\right)^2}{2\cdot t_b\cdot n_d\cdot S_d} - \frac{10}{t_b\cdot n_d\cdot S_d}\cdot\left[\frac{3,93\cdot(1+\gamma)\cdot V^2(t)}{g\cdot\mu_s(t)\cdot\delta + r_t} + \frac{V(t)\cdot t_u}{7,2}\right]\cdot\left(a + b\cdot V(t) + c\cdot V^2(t)\right) \qquad [W/m^2] \tag{4}$$

where:
m – mass of the vehicle, [t];
t_b – braking time, [s];

n_d – number of brake discs (in calculation of n_d = 1; 2; 3 and 4);

$a + b \cdot V(t) + c \cdot V^2(t)$ – General expression of the running resistance of the railway vehicle whose coefficients are experimentally determined, [5-6].

By means of the relationship (4), the unit heat flow q [W/m^2] has been determined for the proposed brake disc surfaces (Sd = 0.2 m^2; Sd = 0.25 m^2, Sd = 0.3 m^2, Sd = 0.35 m^2) the values of which are given in Table 1 and figures 5, 6, 7 and 8 represent its variation with the initial brake speed.

It is noted that at the same friction surface of the brake disc (Sd), the thermal flow decreases as the braking duration increases, thus increasing the braking space. For example, with an initial brake speed of 200 km/h taking a brake duration t_b = 30 s and the brake disc friction surface Sd = 0,3 m^2, a unit thermal flow q = 1.583106· W/m^2 is obtained. At the same speed and the brake disc friction surface at a braking time tb = 60 s, a lower unit thermal flux is obtained q = 0.3959106· W/m^2.

This variation in the unit heat flow with the friction surface of the brake disc and with the duration of the braking may be explained by the fact that with the increase in the brake lining force on the disc, the friction coefficient and the effective heat transmission area to the two friction coupler components (brake disc and lining) shall be subtracted.

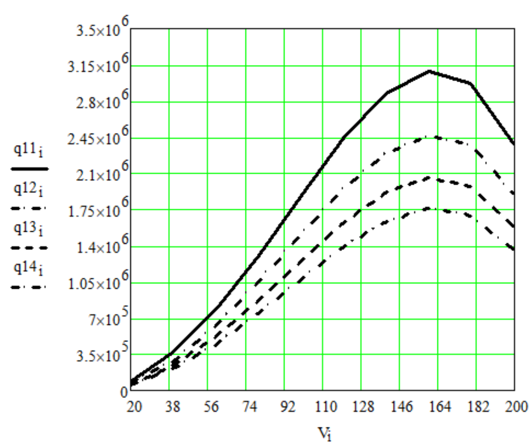

Fig. 5. Variation of thermal flow at speed at different brake disc friction surfaces ($q_{11} \rightarrow Sd_1 = 0{,}2$ m^2; t_b = 30 s, nd_1 =1; $q_{12} \rightarrow Sd_2 = 0{,}25$ m^2; t_b = 30 s, nd_1 =1; $q_{13} \rightarrow Sd_3 = 0{,}3$ m^2; t_b = 30 s, nd_1 =1; $q_{14} \rightarrow Sd_4 = 0{,}35$ m^2; t_b = 30 s, nd_1 =1)

Table 1. *The unit heat flow values for the different friction surfaces of the brake disc*

THE UNIT HEAT FLOW, [W/m²]												
SPEED, [km/h]		0	20	40	60	80	100	120	140	160	180	200
$t_b = 30$ s	Sd = 0,2	0	92460	365400	788200	1317000	1892000	2441000	2875000	3090000	2968000	2375000
	Sd = 0,25	0	73970	292300	630600	1053000	1514000	1953000	2300000	2472000	2374000	1900000
	Sd = 0,3	0	61640	243600	525500	877800	1261000	1627000	1916000	2060000	1979000	1583000
	Sd = 0,35	0	52830	208800	450400	752400	1081000	1395000	1643000	1766000	1696000	1357000
$t_b = 40$ s	Sd = 0,2	0	46230	182700	394100	658400	946100	1220000	1437000	1545000	1484000	1188000
	Sd = 0,25	0	36980	146100	315300	526700	756900	976400	1150000	1236000	1187000	950100
	Sd = 0,3	0	30820	121800	262700	438900	630700	813700	958200	1030000	989300	719700
	Sd = 0,35	0	26420	104400	225200	376200	540600	697400	821300	882800	848000	678600
$t_b = 50$ s	Sd = 0,2	0	30820	121800	262700	438900	630700	813700	958200	1030000	989300	791700
	Sd = 0,25	0	24660	97430	210200	351100	504600	650900	766600	824000	791500	633400
	Sd = 0,3		20550	81190	175200	292600	420500	542400	638800	686600	659500	527800
	Sd = 0,35	0	17610	69590	150100	250800	360400	464900	547600	588600	565300	452400
$t_b = 60$ s	Sd = 0,2	0	23110	91340	197100	329200	473000	610200	718700	772500	742000	503800
	Sd = 0,25	0	18490	73070	157600	263400	378400	488200	574900	618000	593600	475000
	Sd = 0,3		15410	60890	131400	219500	315400	406800	479100	515000	494700	395900
	Sd = 0,35	0	13210	52190	112600	188100	270300	348700	410700	441400	424000	339300

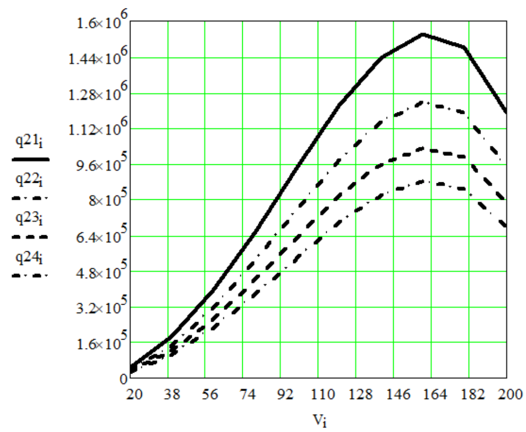

Fig. 6. Variation of thermal flow at speed at different brake disc friction surfaces ($q_{21} \rightarrow Sd_1 = 0{,}2$ m^2; t_b = 40 s, nd_2 =2; $q_{22} \rightarrow Sd_2 = 0{,}25$ m^2; t_b = 40 s, nd_2 =2; $q_{23} \rightarrow Sd_3 = 0{,}3$ m^2; t_b = 40 s, nd_2 = 2; $q_{24} \rightarrow Sd_4 = 0{,}35$ m^2; t_b = 40 s, nd_2 =2)

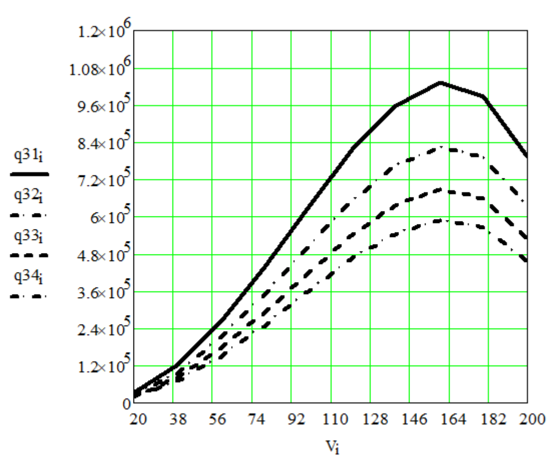

Fig. 7. Variation of thermal flow at speed at different brake disc friction surfaces ($q_{31} \rightarrow Sd_1 = 0{,}2$ m^2; t_b = 50 s, nd_3 =3; $q_{32} \rightarrow Sd_2 = 0{,}25$ m^2; t_b = 50 s, nd_3 =3; $q_{33} \rightarrow Sd_3 = 0{,}3$ m^2; t_b = 50 s, nd_3 = 3; $q_{34} \rightarrow Sd_4 = 0{,}35$ m^2; t_b = 50 s, nd_3 =3)

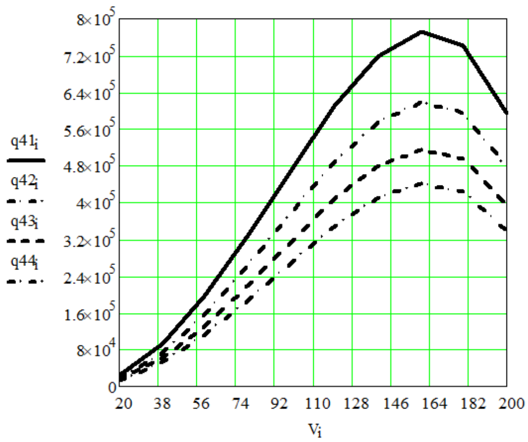

Fig. 8. Variation of thermal flow at speed at different brake disc friction surfaces
($q_{41} \rightarrow Sd_1 = 0{,}2\ m^2$; $t_b = 60$ s, $nd_4 = 4$; $q_{42} \rightarrow Sd_2 = 0{,}25\ m^2$; $t_b = 60$ s, $nd_4 = 4$; $q_{43} \rightarrow Sd_3 = 0{,}3\ m^2$; $t_b = 60$ s, $nd_4 = 4$; $q_{44} \rightarrow Sd_4 = 0{,}35\ m^2$; $t_b = 60$ s, $nd_4 = 4$)

2.2. Determination of the temperature

Once the unit heat flow of the surface is known, the average temperature of the friction surface (5) can be determined:

$$T_m = const \cdot f(T) \cdot f_m \cdot q(T) \qquad \left[{}^{o}C\right] \tag{5}$$

Using the expressions of the quantities in relation (5) and taking into account their variation with temperature, the following expression of the medium temperature of the brake disc friction surface (6) is obtained:

$$T_m = const \cdot f(T) \cdot \frac{2}{1 + \dfrac{\sqrt{\rho_g \cdot c_g \cdot \lambda_g}}{\sqrt{\rho_d \cdot c_d \cdot \lambda_d}}} \cdot \left\{ \frac{(1+\gamma) \cdot m \cdot \left(\dfrac{V(t)}{3{,}6}\right)^2}{2 \cdot t_b \cdot n_d \cdot S_d} - \frac{10}{t_b \cdot n_d \cdot S_d} \cdot \left[\frac{3{,}93 \cdot (1+\gamma) \cdot V^2(t)}{g \cdot \mu_s(t) \cdot \delta + r_t} + \frac{V(t) \cdot t_u}{7{,}2} \right] \cdot \left(a + b \cdot V(t) + c \cdot V^2(t)\right) \right\} \qquad [{}^{o}C] \tag{6}$$

For sizes in the relationship (6) the following values were taken as the actual calculation of the mean brake disc friction surface temperature:

- brake cylinder compressed-air filling time, $t_u = 4$ seconds;

Table 2. *The values of the f (T) function for braking times (t_b = 30 s; 40 s; 50 s; 60 s)*

Temperature [°C]	f(T) (t_b =30 s)	f(T) (t_b =40 s)	f(T) (t_b =50 s)	f(T) (t_b =60 s)
0	0	0	0	0
20	7,0805817340	8,1759515403	9,1409917124	10,013454717
40	2,7061299695	3,1247697327	3,4935987682	3, 8270457044
60	1,7677242501	2,0411921433	2,2821221938	2,4999396090
80	1,3617481003	1,5724112646	1,7580092388	1,9258026320
100	1,0966455677	1,2662972274	1,4157633401	1,5508911035
120	0,8863786770	1,0235019356	1,1443099515	1,2535287464
140	0,7394477629	0,8538407299	0,9546229570	1,0457370550
160	0,6279494939	0,7250936187	0,8106793107	0,888054690
180	0,5509594279	0,6361931480	0,7112855629	0,7791742953
200	0,4907731805	0,5666960557	0.6335854516	0,6940580879
220	0,4479153827	0,5172081335	0,5782562725	0,6334480090
240	0,4123332174	0,4761213882	0,5323198947	0,5831272283
260	0,3852003792	0,4447910814	0,4972915469	0,5447555958
280	0,3659195272	0,4225274751	0,4724000783	0,5174883581
300	0,3544427698	0,4092752571	0,4575836482	0,5012577721
320	0,3527177578	0,4072833848	0,4553566673	0,4988182368
340	0,3682345936	0,4252006835	0,4753888162	0,5207623564
350	0,4079264320	0,4710328706	0,5266307592	0,5768950926

- total brake disc friction surface, S_d = 0,2; 0,25; 0,3; 0,35 m^2;
- specific running resistance, r_t = 20 N/kN;
- brake disc friction lining force, F_b = 20 kN, F_b = 45 kN și F_b = 50 kN;
- braking time: t_b = 30 s; t_b = 40 s; t_b = 50 s; t_b = 60 s;
- the experimentally determined running resistance is: $R_t = 250 + 3{,}256 \cdot V(t) + 0{,}0572 \cdot V^2(t)$, [5-6].
- the physical properties (density ρ , c_d- specific heat and thermal conductivity λ) of the materials of which the friction elements are made were considered to be constant with temperature (in the range 0 to 350° C they vary with temperature), the following values were used in the calculations, [14-15]:
  - ✓ for non-ventilated alloy steel brake disc 30MoCrNi20; ρ_{d2} = 7840 kg / m³; c_{d2} = 465 J / kg·°C; λ_{d2} = 49,8 W / m·°C;
  - ✓ for the friction lining: ρ_g = (1000 – 2800) kg / m³; c_g = (628,05 - 2093,5) J / kg·°C; λ_g = (0,16282 - 1,01181) W / m·°C.

The average frictional surface temperature values of the disc brake friction elements calculated using the MathCad utility are shown in table 3 and figure 9 shows the variation of temperature as a function of time (braking time) for the different friction surfaces of the brake disc.

Table 3. *Temperature values during fanfare for different brake disc friction surfaces and braking times*

AVERAGE TEMPERATURE [°C]								
TIME (BRAKING DURATION), [s]		0	10	20	30	40	50	60
$t_b = 30$ s	$Sd = 0,2\ m^2$	0	260,178	427,166	499,524	541,871	575,085	624,757
	$Sd = 0,25\ m^2$	0	257,460	418,629	481,535	507,860	508,673	466,013
	$Sd = 0,3\ m^2$	0	256,982	417,130	478,376	501,992	497,014	438,147
$t_b = 40$ s	$Sd = 0,2\ m^2$	0	225,321	369,936	432,601	468,767	498,038	541,055
	$Sd = 0,25\ m^2$	0	222,967	362,543	417,021	439,820	440,523	403,579
	$Sd = 0,3\ m^2$	0	222,553	361,245	414,286	434,738	430,427	379,446
$t_b = 50$ s	$Sd = 0,2\ m^2$	0	201,533	330,881	386,930	419,278	445,458	483,935
	$Sd = 0,25\ m^2$	0	199,427	324,268	372,995	393,387	394,016	360,972
	$Sd = 0,3\ m^2$	0	199,058	323,107	370,548	388,841	384,985	339,387
$t_b = 60$ s	$Sd = 0,2\ m^2$	0	183,974	302,052	353,217	382,746	406,696	441,770
	$Sd = 0,25\ m^2$	0	182,051	296,015	340,496	359,111	359,686	329,521
	$Sd = 0,3\ m^2$	0	181,714	294,955	338,263	354,962	351,442	309,816

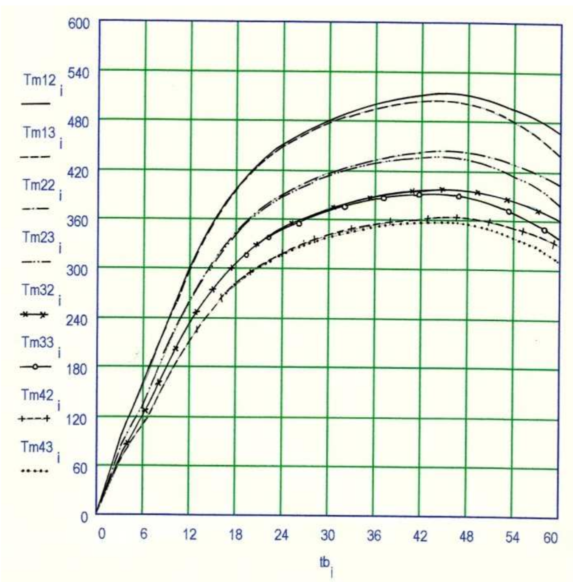

Fig. 9. Variation of the average friction surface temperature during braking for different brake disc friction surfaces
($T_{m12} \rightarrow q_{12}$ (nd = 1; Sd = 0,25 m^2); $T_{m13} \rightarrow q_{13}$ (nd = 1; Sd = 0,3 m^2); $T_{m22} \rightarrow q_{22}$ (nd = 2; Sd = 0,25 m^2); $T_{m23} \rightarrow q_{23}$ (nd = 2; Sd = 0,3 m^2); $T_{m32} \rightarrow q_{32}$ (nd = 3; Sd = 0,25 m^2); $T_{m33} \rightarrow q_{33}$ (nd = 3; Sd = 0,3 m^2); $T_{m42} \rightarrow q_{42}$ (nd = 4; Sd = 0,25 m^2); $T_{m43} \rightarrow q_{43}$(nd = 4; Sd = 0,3 m^2))

Note that a lower thermal regime (lower average temperature values) is achieved in the case of a larger friction surface of the brake disc when braking with a longer duration (e.g. for a value **Sd = 0,25 m^2**, at **t_b = 30 s** an average temperature of **481.535 °C** is obtained and, with the same friction surfaces, at a brake time **t_b = 50 s** the temperature obtained is **394.016 °C.** The excessive increase in braking time (**t_b >60 s**) leads to a steady deceleration increase in braking space for high-speed trains with negative side effects on road safety. The best solution in this situation is to combine brakes, i.e. to use, in addition to the disc brake, another brake whose effect does not depend on wheel adhesion - rail: electric brake and/or electromagnetic brake on the rail.

3. CONCLUSIONS

With the increasing speed of trains running on high-speed rail networks in the world (currently running at commercial speeds of 350 km/h) the kinetic energy which has to be dissipated by braking systems is very high. Thus, for braking these trains (speed reduction, constant speed and stopping), both brakes are used, the effect of which depends on the adhesion between the wheel and the rail, and brakes independant of this adhesion.

The disc brake is used on both conventional and high-speed rail vehicles. The friction elements of this type of brake (friction linings and brake disc) are different in terms of both the materials from which they are made and the geometric configuration (shape, volume).

This work has presented, by means of calculations and diagrams, a proposal to decrease the heat regime to which the elements of the disc brake friction coupling are subject, i.e. to increase the friction surface of the brake disc, taking into account the limits imposed by the railway gauge.

From the analysis of the tables and diagrams presented, the following conclusions can be drawn:

- it is possible to increase the friction surface of the brake disc by increasing its actual diameter (self-ventilated brake discs with diameters of 590 and 640 mm are currently used, as well as non-ventilated discs with smaller diameters), [2-4];
- by increasing the friction surface of the brake disc, the heat rating of the friction coupling elements: *brake disc — lining* will be lower. Thus, for example, at an initial brake speed of 200 km/h, considering a brake disc friction surface Sd = 0,2 m^2 and

a braking time tb = 30 s, a unit thermal flow q = 2,375106· W/m^2 is obtained, And for a larger friction surface Sd = 0,3 m^2, at the same brake duration, q = 1,583106· W/m^2;

- the increase in the number of discs per axle (nd > 2), without increasing the unsuspended mass of the axle, is practically only feasible when non-ventilated alloy steel discs are used, which have a simpler configuration and a smaller mass than those made of perlitic gray cast iron (high density);
- increase the number of discs per axle from nd = 1 (case shown above) to nd = 3 non-ventilated disks, for Sd = 0,2 m^2 a unit heat flow q = 0.7917106· W/m^2 is obtained. In the case of a larger friction surface Sd = 0,3 m^2, the unit thermal flow shall be reduced to q = 0.5278106· W/m^2;
- temperature of the brake disc friction clutch components decreases with the increase of the disc friction surface and braking time (tb). Thus, at a brake disc surface Sd = 0,3 m^2 for a braking time tb = 40 s, a temperature of 434,738°C is obtained and for a braking time tb = 60 s the temperature is reduced to 354,962 °C.

4. REFERENCES

[1] J. Adamsen, ”Heating and cooling of Friction Brakes”, Seminar on Braking, London 1986;

[2] L.S. Bocîi, ”Contribuţii la frânarea vagoanelor de călători de mare viteză”, Teză de doctorat, Universitatea Politehnica Timişoara, 1997;

[3] L.S. Bocîi, ”Programa de calculo de la temperatura en ruedas de ferrocarril durante la frenada”, INGENIERIAS, ISSN:1405-0676, Universidad Autonoma de Nuevo Leon Mexico vol. VI, no. 21, 25 – 32, Octubre – Diciembre 2003;

[4] L.S. Bocîi, ”Determination of the friction surface temperature by the Hasselgruber method using brake disc with different physical properties”, revista Metalurgia Internaţional, vol. XVI, no. 8 2011, ISSN: 1582-2214, pag. 42 – 47.

[5] M. Boiteux, ”Le probleme de l’adherence en freinage”, Revue Générale de Chemins de Fer, Februarie 1986;

[6] M. Boiteux, ”Influence de l’energie de glissement sur l’adherence exploitable en freinage”, Revue Générale de Chemins de Fer, Octombrie 1987;

[7] J.D. Chen, J.H. Chern Lin, ”Effect of humidity on the tribological behaviour of carbon-carbon composites”, Wear n° 193, 1996, pp. 38-47;

[8] F.Demilly, M. Trohel, ”Les disques de frein en acier forge des trains a grande vitesse”, Journees Europeennes du Freinage (JEF 2002), 13 et 14 Mars 2002, Lille France, pag. 103 – 110;

[9] H.R. Ehlers, ”Die thermishe Berechnung der Klotzbremse”, Sonderdruck aus Archiv für Eisenbahntechnik, Folge 18;

[10] H. Hasselgruber, ”Zur Berechnung der Wärmespannungen in der Bremstrommeln von Kraftfahrzeugen beim Haltbremsvorgang”, A. T. Z nr. 2 / 1954;

[11] H. Hasselgruber, ”Temperaturberechnungen für mecanische Reibkupplungen”, Wieweg 1959;

[12] B.L. Karwaski, ”Teoria generală a frânelor automate”, Oficiul de presă, Editură şi documentare CFR, 1950;

[13] O. Iordache, O. Smigelschi, ”Ecuaţiile fenomenelor de transfer de masă şi căldură”, Editura Tehnică Bucureşti;

[14] L. Sofroni, , I. Riposan, V. Brabie, M. Chişamera, ”Turnarea fontei”, Editura Didactică şi Pedagogică, Bucureşti 1985;

[15] L.Sofroni, V. Brabie, C. Bratu, ”Bazele teoretice ale turnării”, Editura Didactică şi Pedagogică, Bucureşti 1980.

[16] C. Baron Saiz, T. Ingrassia, V. Nigrelli, V. Ricotta, ”Thermal stress analysis of different full and ventilated disc brakes”, Frattura ed Integrità Strutturale, 34 (2015) 608-621; 2015, https://doi.org/10.3221/IGF-ESIS.34.67;

[17] N. Kudarauskas, ”Analysis of emergency braking of a vehicle”, Transport, 22:3,

154-159, 2007, https://doi.org/10.1080/16484142.2007.9638118;

[18] A. Adamowicz, P. Grzes. ”Influence of Convective Cooling on a Disc Brake Temperature Distribution During Repetitive Braking”, Applied Thermal Engineering 2011, https://doi.org/10.1016/j.applthermaleng.2011.05.016.

Influența dimensiunii suprafeței de frecare a discului de frână asupra regimului termic al frânei cu disc a vehiculelor feroviare de mare viteză

Rezumat: Unul dintre cele mai importante criterii pentru utilizarea unei frâne din sistemul mecanic de frânare în cazul vehiculelor feroviare de mare viteză este temperatura elementelor cuplei de frecare ale frânei (sabot - roată sau disc de frână - plăcuță de frecare). Frâna cu disc elimină acțiunea saboților de frână asupra suprafaței de rulare a roții, în timpul procesului de frânare și prezintă posibilitatea unei disipări mai mari a energiei pe durata frânării datorită utilizării materialelor pentru realizarea elementelor cuplei de frecare cu caracteristicile fizico-mecanice optime. În această lucrare se prezintă o analiză a regimului termic (flux termic și temperatură) al elementelor cuplei de frecare a frânei cu disc. De asemenea pentru a pune în evidență influența dimensiunii suprafeței de frecare a discului de frână asupra regimului termic s-au considerat patru mărimi ale suprafeței de frecare a discului de frână (Sd = 0,2 m^2; Sd = 0,25 m^2; Sd = 0,3 m^2; Sd = 0,35 m^2). Astfel, s-a ajuns la concluzia că utilizarea discului de frână neventilat conduce la un regim termic mai ușor (flux termic și, implicit, temperaturi mai scăzute ale elementelor cuplei de frecare ale frânei cu disc) indiferent de tipul garniturii de frecare utilizate.

Liviu Sevastian BOCÎI, Professor, PhD, ”Aurel Vlaicu” University of Arad; Department of Automation, Industrial Engineering, Textiles and Transport; no 2, Elena Drăgoi Street, 310330 Arad, Romania, tel/fax: +40 257 250389; bociis@yahoo.com