



## THE INFLUENCE OF BRAKING TIME ON HEAT FLOW THROUGH THE FRICTION SURFACES OF THE FRICTION ELEMENTS OF DISK BRAKES FOR RAILWAY VEHICLES

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**Abstract.** Disk brakes were first introduced on the suburban and urban coaches and high speed freight cars ( $V > 120$  km/h) for the following reasons: shoe brakes reached their potential limits, especially at high speeds; the maintenance of disc brakes is cheaper; greater comfort; the friction coefficient of disk brake elements (friction pad and brake disc) presents lower variation in velocity and pressure; disk brakes present lower temperatures of friction surfaces. The determination of uniform heat flow through the friction elements of disk brakes is very important since the early designing stage. Such way one avoids overheating both disk brakes and friction pads that may result when the physical properties (chemical and mechanical) of materials are not correctly matched and have negative effects on braking efficiency (small coefficient of friction, large braking distance etc.) This work studies the influence of braking time (30 s, 40 s, 50 s, 60 s) on the uniform heat flow of the friction element surface. Note that in the same area of disk brake friction  $0.4 \text{ m}^2$ , heat flow decreases with an increased duration of braking, leading to longer braking space. At the same time, if pressure on the pad of disk brake friction decreases, heat flow increases.

**Keywords:** braking time, uniform heat flow, friction coupling, brake disc, brake stopping.

### 1. Introduction

A disk brake is used as the main braking device of railway vehicles because of the following factors (Bocîi 1997, 1999, 2006; Guiheu 1982; Panagin 1990):

- the friction coefficient of friction coupling elements has lower variations in speed, pressure, temperature and humidity;
- reduced complexity;
- braking distance and braking decelerations obtained at low and very high speeds;
- greater safety and high reliability in operation;
- eliminate the noises of shoe brakes;
- reduced operating forces;
- better thermal regime (low temperature) friction coupling elements;
- the weight of braking equipment (disc brake friction linings, brake wheelhouse etc.) is much lower than that of shoe brakes.

### 2. Determining the Flow Uniformity of Friction Surfaces

The factors that depend on uniform heat flow  $q$  [ $\text{W}/\text{m}^2$ ] and the friction surface of friction coupling components of disk brakes (friction pad and brake disc), in addition

to temperature variation in the physical properties of materials of such elements, are (Bocîi 1997, 1999):

- braking power is disc brake  $P_{mfd}$  [W];
- the total area of disc brake friction  $S_d$  [ $\text{m}^2$ ];
- braking time  $t_b$  [s].

Considering these factors and changes to influence on the physical properties (specific heat  $c_d$ ,  $c_g$  and thermal conductivity  $\lambda_d$  and  $\lambda_g$ ) of disc and seal friction, regarding temperature (Bocîi, 1997, 1999, 2006), the expression of heat flow through surface friction is:

$$q = \frac{P_{mfd}}{S_d} \text{ [W}/\text{m}^2], \quad (1)$$

where:  $P_{mfd}$  – braking power which is a disc brake [W];  $S_d$  – the total area of disc brake friction [ $\text{m}^2$ ].

During the movement of a train on a rail profile, thrust is consumed both for overcoming drag resistance (resistance to total submission) and for printing accelerated movement. Once the interruption of the process for transmitting torque to the engine thrust of axles is cancelled, the train drags acting alone. During the action of braking, friction pads 'click' on two flat surfaces of a disc brake causing braking force, thus, we can write the

equality between kinetic energy stored in the work table and train resistance forces:

$$(1+\gamma) \cdot \frac{m}{2} \cdot (v_2^2 - v_1^2) = S_f \cdot (F_b \cdot \mu_s + R_t \pm i \cdot G_t), \quad (2)$$

where:  $(1+\gamma)$  – a factor taking into account rotating masses;  $m$  – train mass, [kg];  $v_2, v_1$  – starting speed and ending speed of braking (brake off  $v_1 = 0$ ) [m/s];  $S_f$  – braking area [m];  $F_b$  – pressure on the friction pad of a disk brake [daN];  $\mu_s$  – the coefficient of friction between the friction pad and a disk brake;  $R_t$  – total resistance to advance the train, [daN];  $G_t$  – train weight [daN];  $i$  – gradients [%].

Taking into account an ending brake, in the alignment and landing, relation 2 becomes:

$$(1+\gamma) \cdot \frac{m}{2} \cdot (v_2^2 - v_1^2) = S_f \cdot (F_b \cdot \mu_s + R_t \pm i \cdot G_t). \quad (3)$$

Brake off and the kinetic energy of the train the vehicles of which are equipped with disk brakes, become resistant to mechanical forces and temperature raises the heat of friction coupling components, including a disk brake and friction pad.

Kinetic energy that has to be cancelled by braking is given by Bocii (1997, 1999, 2006):

$$E_{cf} = \frac{(1+\gamma) \cdot m \cdot \left(\frac{V}{3.6}\right)^2}{2} - 10 \cdot S_f \cdot R_t. \quad (4)$$

The kinetic energy of the train, given by 4, corresponds to an average braking power the expression of which is as follows:

$$P_{mf} = \frac{E_{cf}}{t_b} \quad [\text{W}]. \quad (5)$$

Substituting kinetic energy, relationship 5 (relation 4) is obtained:

$$P_{mf} = \frac{(1+\gamma) \cdot m \cdot \left(\frac{V}{3.6}\right)^2}{2 \cdot t_b} - 10 \cdot \frac{S_f \cdot R_t}{t_b} \quad [\text{W}], \quad (6)$$

where:  $V$  – initial speed of braking [km/h].

Braking space in relation to (6) has the following expression (Bocii 1997, 1999; Karwaski 1950):

$$S_f = \frac{3.93 \cdot (1+\gamma) \cdot V^2}{g \cdot \mu_s \cdot \delta + r_t} + \frac{V \cdot t_u}{7.2} \quad [\text{m}] \quad (7)$$

and considering, for example, the known high-speed train (TGV-PSE), the total drag resistance of which was determined experimentally (Guiheu 1982) for a total mass of 390 tons and a composition  $L + 8V + L$ , relationship (6) becomes:

$$P_{mf} = \frac{(1+\gamma) \cdot m \cdot \left(\frac{V}{3.6}\right)^2}{2 \cdot t_b} - \frac{10}{t_b} \times \left[ \frac{3.93 \cdot (1+\gamma) \cdot V^2}{g \cdot \mu_s \cdot \delta + r_t} + \frac{V \cdot t_u}{7.2} \right] \times (a + b \cdot V + c \cdot V^2) \quad [\text{W}]. \quad (8)$$

If a braking stop uses only disc brakes, the power of brake disks is obtained using equation (9):

$$P_{mf} = n_d \cdot P_{mfd} \Rightarrow P_{mfd} = \frac{(1+\gamma) \cdot m \cdot \left(\frac{V}{3.6}\right)^2}{2 \cdot t_b \cdot n_d} - \frac{10}{t_b \cdot n_d} \cdot \left[ \frac{3.93 \cdot (1+\gamma) \cdot V^2}{g \cdot \mu_s \cdot \delta + r_t} + \frac{V \cdot t_u}{7.2} \right] \times (a + b \cdot V + c \cdot V^2) \quad [\text{W}]. \quad (9)$$

If relationship is replaced by an expression of the power of brake disks, it looks as follows:

$$q = \frac{(1+\gamma) \cdot m \cdot \left(\frac{V}{3.6}\right)^2}{2 \cdot t_b \cdot n_d \cdot S_d} - \frac{10}{t_b \cdot n_d \cdot S_d} \times \left[ \frac{3.93 \cdot (1+\gamma) \cdot V^2}{g \cdot \mu_s \cdot \delta + r_t} + \frac{V \cdot t_u}{7.2} \right] \times (a + b \cdot V + c \cdot V^2) \quad [\text{W/m}^2]. \quad (10)$$

Relation unit (10) defines the heat flow of surface friction elements into coupling brake disk (friction pad and brake disk), (Bocii 1997, 1999, 2006; Ehlers 1969; Hasselgruber 1954, 1959; Kirschstein 1970; Saumweber 1974). When calculating sizes in relation (10) it has been established that:

- the total mass of the train view (TGV-PSE)  $m = 390$  t;
- the number of disks per axles  $n_d = 42$ ;
- the total area of disk brake friction  $S_d = 0.4$  m<sup>2</sup>;
- specific resistance to advance  $r_t = 20$  N/kN;
- the coefficient of braking  $\delta = 0.35$ ;
- the time of filling the brake cylinder with compressed air  $t_u = 4$  s;
- experimentally determined drag resistance is given by (Guiheu 1982):  $R_t = 250 + 3.256 \cdot V + 0.0572 \cdot V^2$ .

### 3. The Influence of Braking Time on Uniform Heat Flow in Coupling Brake Disk

Uniform heat flow ( $q$ ) on surface friction, the friction torque components of brake disk and linear variation in velocity considering time (during braking) are determined referring to relationship (10) where the total drag resistance is determined experimentally (coefficient expressions  $a, b$  and  $c$  are determined by experimental tests in wind tunnels or the current line).

This relationship assisted in determining uniform heat flows on the surface of the friction elements of the friction torque of disc brakes (disc brake friction lining), as shown in Table.

In the following Figs 1–4 variations in the uniform heat flow rate at different pressure of the friction pad on disc brakes are presented ( $F_b = 20$  kN,  $F_b = 45$  kN,  $F_b = 50$  kN);  $q_{11} \rightarrow t_b = 30$  s and  $F_b = 20$  kN;  $q_{12} \rightarrow t_b = 30$  s and

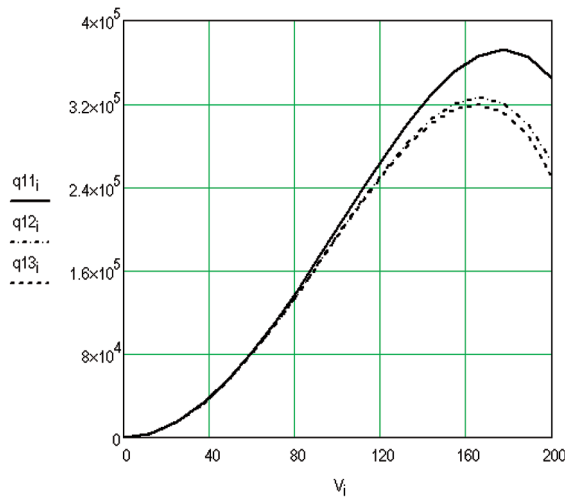


Fig. 1. Variation in uniform heat flow depending on the speed of walking when braking duration  $t_b = 30$  s

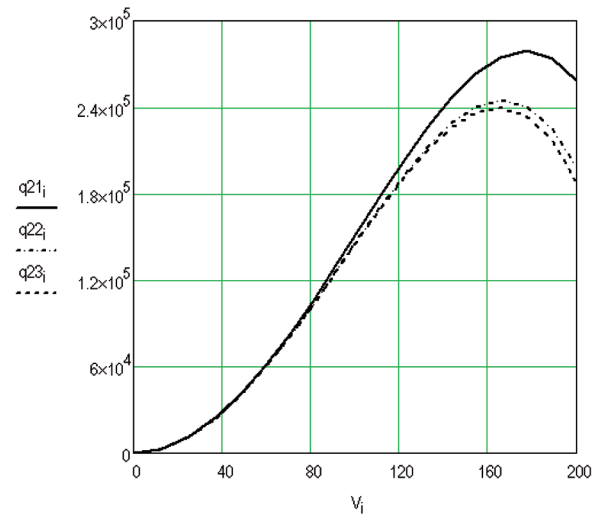


Fig. 2. Variation in uniform heat flow depending on the speed of walking when braking duration  $t_b = 40$  s

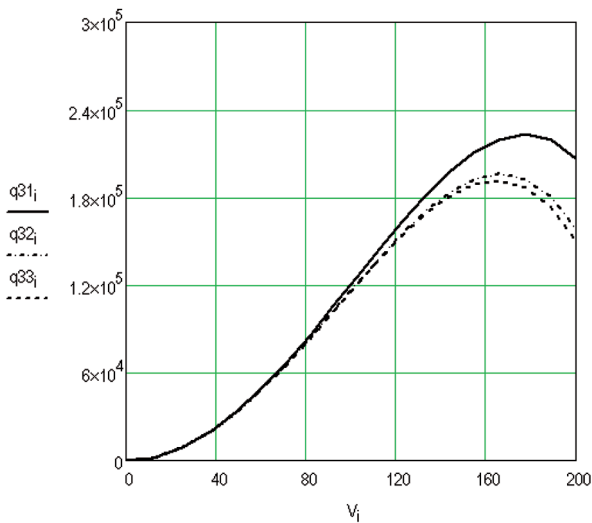


Fig. 3. Variation in uniform heat flow depending on the speed of walking when braking duration  $t_b = 50$  s

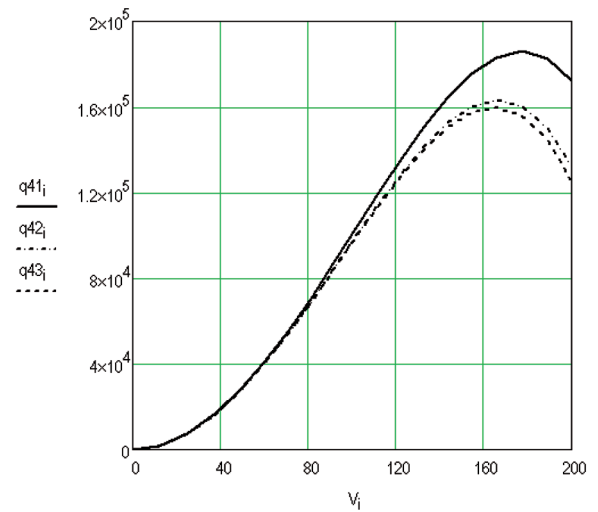


Fig. 4. Variation in uniform heat flow depending on the speed of walking when braking duration  $t_b = 60$  s

Table. Uniform heat flow [W/m<sup>2</sup>] for different durations of braking and the top speed of brakes

Speed V [km/h]	0	20	40	60	80	100	120	140	160	180	200	
$t_b = 30$ s	$\mu_s (F_b = 20 \text{ kN})$	0	9507	37677	82006	138908	203621	270023	330880	377572	400447	388198
	$\mu_s (F_b = 45 \text{ kN})$	0	9441	37294	80781	135909	197371	258334	310724	344910	350109	313683
	$\mu_s (F_b = 50 \text{ kN})$	0	9429	37226	80565	135383	196273	256282	307185	339175	341270	300601
$t_b = 40$ s	$\mu_s (F_b = 20 \text{ kN})$	0	7130	28257	61504	104181	152715	202517	248160	283179	300335	291148
	$\mu_s (F_b = 45 \text{ kN})$	0	7081	27970	60585	101932	148028	193751	233043	258682	262581	235262
	$\mu_s (F_b = 50 \text{ kN})$	0	7072	27920	60424	101537	147205	192211	230389	254381	255953	225450
$t_b = 50$ s	$\mu_s (F_b = 20 \text{ kN})$	0	5704	22605	49203	83344	122172	162014	198528	226543	240268	232919
	$\mu_s (F_b = 45 \text{ kN})$	0	5664	22376	48468	81545	118422	155000	186434	206946	210065	188210
	$\mu_s (F_b = 50 \text{ kN})$	0	5658	22336	48340	81230	117764	153770	184311	203505	204763	180361
$t_b = 60$ s	$\mu_s (F_b = 20 \text{ kN})$	0	4754	18838	41003	69454	101810	135011	165440	188786	200224	194099
	$\mu_s (F_b = 45 \text{ kN})$	0	4720	18647	40391	67955	98686	129167	155362	172455	175055	156842
	$\mu_s (F_b = 50 \text{ kN})$	0	4715	18613	40283	67692	98137	128141	153593	169588	170635	150301

$F_b = 45$  kN;  $q_{13} \rightarrow t_b = 30$  s and  $F_b = 50$  kN;  $q_{21} \rightarrow t_b = 40$  s and  $F_b = 20$  kN;  $q_{22} \rightarrow t_b = 40$  s and  $F_b = 45$  kN;  $q_{23} \rightarrow t_b = 40$  s and  $F_b = 50$  kN;  $q_{31} \rightarrow t_b = 50$  s and  $F_b = 20$  kN;  $q_{32} \rightarrow t_b = 50$  s and  $F_b = 45$  kN;  $q_{33} \rightarrow t_b = 50$  s and  $F_b = 50$  kN;  $q_{41} \rightarrow t_b = 60$  s and  $F_b = 20$  kN;  $q_{42} \rightarrow t_b = 60$  s and  $F_b = 45$  kN;  $q_{43} \rightarrow t_b = 60$  s and  $F_b = 50$  kN).

The analysis of the diagrams presented in the previous figures show that at the same time of braking, heat flow decreases by increasing pressure on the friction pads of disc brakes. Thus,  $t_b = 50$  start of braking and speed  $V = 160$  km/h as well as thermal flow  $F_b = 45$  kN, has value  $q = 206946$  W/m<sup>2</sup>, and for  $F_b = 50$  kN, heat flow makes  $q = 203505$  W/m<sup>2</sup>.

#### 4. Conclusions

It can be noticed that on the friction surface ( $S_d = 0.4$  m<sup>2</sup>) of coupling brake disc, heat flow decreases with an increased duration of braking, encouraging the growth of an increasing braking distance with a decrease in flow pressure on the friction linings of brake disc. For example, when braking time is  $t_b = 30$  s and friction force is led to a push of  $F_b = 20$  kN, uniform heat flow  $q = 388198$  W/m<sup>2</sup> at a starting speed of braking 200 km/h is reached. At the same time, if an increase in braking force when pressing is  $t_b = 40$  s and  $V = 200$  km/h and the force of pressure –  $F_b = 20$  kN, uniform heat flow  $q = 291148$  W/m<sup>2</sup> is obtained.

Such variation in heat flow during a braking unit would explain the fact that with increased pressure for sealing the coefficient of disk friction, the effective area of the heat transmission of the two elements of friction coupling (friction pad and brake disc) decreases.

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