MATHEMATICAL MODELING OF DISC BRAKE FRICTION LINING HEAT AND WEAR

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Abstract. The author analyzes experimental research for needs of the joint-stock company “Jelgava engineering plant” (A/S “Jelgavas masinbuves rupnica”; they produce disc and drum brake) regarding heat and wear of disc brake friction linings. The disc brake DBE-1 was tested in the laboratory of “Jelgava engineering plant” on inertia stand in a short-time repeating braking regime. The article analyzes the main impact factors on the braking process and interrelation between these factors: temperature, friction coefficient and wear depending on friction lining material, specific surface pressure, braking frequency and slipping velocity of the friction surface. To improve the disc brake design the author develops calculation algorithm of friction pair heat and wear – Russian scientist’s (A. B. Чичинадзе) mathematical model regarding heat and wear was complemented and adopted. Thereby, all main braking process impact factors are linked up in a unite equation system. Accuracy of calculation is approximately 10 %. There are recommendations regarding the selection of the optimal friction lining material for disc brakes given in this article.

Key words: disc brake, temperature, wear, mathematical model, friction lining, braking process.

Introduction

Brakes are used to stop the motion or reduce the speed of cargo, lorry or crain. During all life-time brake have to work quietly, without an accident. Desireble dimensions of brake is small, that is why brake is mounted on motor shaft with the smallest turning moment [1].

Brakes absorb the kinetic energy of moving masses by help of friction forces. The dynamic of braking process depends on friction pair materials, loading and temperature, as well as impact of environment [2-5]. These factors bring impact on friction coefficient, too.

The aim of this research is to develop and to test the mathematical model to prognosticate the temperature, wear and braking time of single disc brake friction pair in short time repeating braking regime depending on all main braking process impact factors: friction coefficient, friction pair design and material, specific surface pressure, braking frequency, slipping velocity of the friction surface, friction work and other.

Sliping velocity, loading, braking moment, temperature and wear is linked and depends on friction pair friction, mechanical and thermophysical properties, as well as design and regime of exploitation. [6]

The results of the research allow prognosticating the service time of friction linings already in design stage and allow to choose the most suitable friction lining material for manufactural needs.

Matherials and methods

The experimental research was carried out in the laboratory of “Jelgava engineering plant” on disc brake inertia stand (Fig. 1) in a short-time repeating braking regime. Four kind of friction linings were mounted on disc brake DBE-1: TWG (Spain), Cosid 516 (Germany), Dafmi (Ukraine) and ЭМ-1 (Russia). Friction pair consisted of friction lining and steel disc (Ст 45).

The experimental work consisted of a number of experiments to find the coherences between different braking process impact factors. The duration of each experiment was 6 hours. The experimental work was carried out in short time repeating regime – braking process and pause (cooling time) repeated with definite intervals. The pause was relatively short and that is why the temperature of friction pair increases in every next braking time till temperature of thermal balance was established.

In every experiment one of these parameters was provided: one kind of friction lining material, specific surface pressure p, disc rotational speed n at the beginning of braking and braking frequency R. the inertia stand was designed so that electromotor (P = 45kW) through transmission and coupling turn flywheel till necessary turning speed (n = 200, 400, 600, 800, 1000 rpm). When the adjusting turning speed of flywheel and disc was achieved the coupling turned off, but the braking was started at this moment and lasted till flywheel stoped (n = 0 rpm) in every braking time. The braking frequency (R = 1, 2, 3 times/min) was adjusted before the experiment and was constant during the experiment. The specific surface pressure (p = 0.36, 0.62, 0.89, 1.15 MPa) was adjusted before the experiment changing strain of brake spring. The inaccuracy of spring force measurement ±1 %.

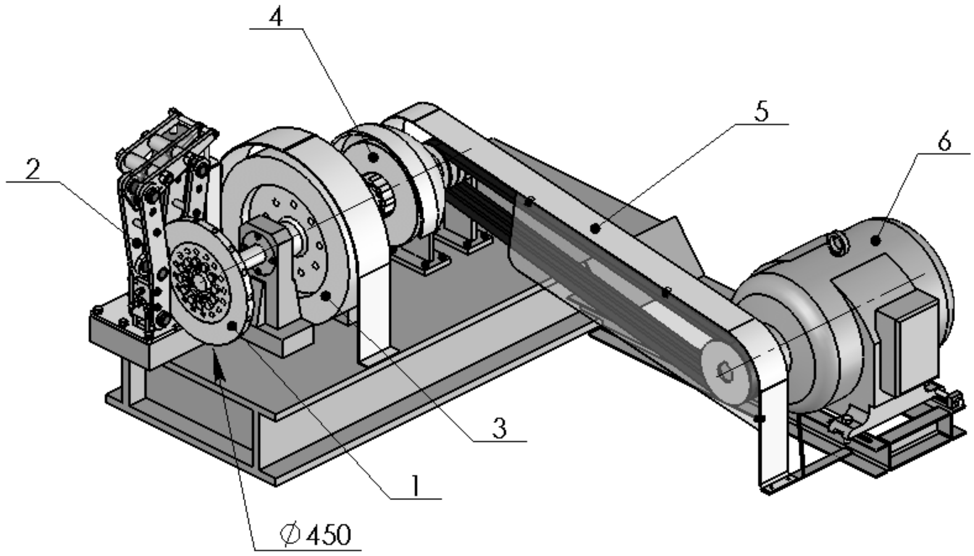

Fig. 1. **Inertia stand:** 1 – disc; 2 – disc brake DBE-1; 3 – flywheel;
4 – coupling (automatically unlink electromotor when necessary rotation speed is achieved);
5 – transmission with a cover; 6 – electromotor (P = 45 kW)

The temperature (elektromotive force) of friction lining was measure with three accurate thermo-couples. The thermo-couples were located 1 mm from friction surface on inner, middle and outside friction lining friction radius. Before and after the experiment the wear of the friction lining were measured with micrometer (value of section 0.001 mm) in nine points on friction surface. During the experiment the friction lining temperature and braking time were measured. Braking time in seconds was measured automatically with accuracy 0.01.

The experimental work correspond to the real exploitation conditions in easy, medium and heavy braking regime (braking frequency one, two and three times in minute). [1]

The friction coefficient was calculated by formula:

$$f = \frac{\left[\frac{J \cdot \pi \cdot n}{30 t_{br}} - M_g\right]}{2F \cdot r_{ef}}, \tag{1}$$

where J – inertia moment of turning masses, J = 14 kgm^2;
n – turning speed at the beginning of braking, rpm;
t_{br} – braking time, s;
M_g – friction moment in bearings, M_g = 4.02 Nm;
F – pressing force applied to the friction lining, N;
r_{ef} – effective radius of friction, r_{ef} = 0.185 m;
f – friction coefficient.

Mathematical model

The wear N of friction lining [7]:

$$N = \frac{N_V}{a \cdot b}, \tag{2}$$

where N – linear wear of friction lining, mm;

N_v – the wear of friction lining volume, mm^3;
a – the width of friction lining, a = 80 mm;
b – the length of friction lining, b = 112 mm.

After literature analyzing [7, 8] author conclude that in a short time repeating braking regime it is possible to calculate the wear of friction lining volume by formula:

$$N_V = \frac{I \cdot W \cdot s}{j}, \tag{3}$$

where I – intensity of wear, mm^3/J;
W – the friction pair work during one braking time, J;
s – number of braking time;
j – number of friction lining in single disc brake design, j = 2.

The intensity of wear, mainly, depends on temperature [9]. Experimentally four friction linings were tested and author revealed the coherence (Table 1) of wear intensity I depending on friction lining temperature T.

Friction work W of one braking time:

$$W = J \cdot w^2 = 14\left(\frac{\pi \cdot n}{30}\right)^2 = 0{,}153n^2, \tag{4}$$

where w – angular velocity of flywheel, s^{-1}.

When the thermal balance of friction pair is established, the disc and friction lining temperature of volume [2]:

$$T_i = T_0 + \frac{\alpha_{SPi} \cdot W_i}{m_i \cdot c_i} \cdot \left(\frac{1}{e^{k \cdot t_C} - 1}\right), \tag{5}$$

where i = index 1 (disc) or i = 2 (friction lining);
T_0 – temperature of environment, $T_0 = 20^\circ C$;
k – weighted average cooling coefficient, s^{-1};
m_i – effective masses of heat reception, kg;
c_i – coefficient of heat capacity, J/(kg·K);
t_c – cycle time, s.

The cooling coefficient k depends on disc design, temperature, turning speed. In this case during the rest $k_{st} = 7 \cdot 10^{-5} \cdot e^{0{,}0113 \cdot T_1}$, but $k_{rot} = 10^{-6} \cdot n + k_{st}$, if disc rotates.

The weighted awerage cooling coefficient k evaluate cooling coefficient in turning k_{rot} and rest k_{st} state, as well as disc turning and rest time intervals:

$$k = \frac{t_{rot} \cdot k_{rot} + t_{st} \cdot k_{st}}{t_c}; \tag{6}$$

where t_{rot} – turning time of disc, s;
t_{st} – disc rest time, s;
k_{rot} – cooling coefficient during disc turns, s^{-1};
k_{st} – cooling coefficient during disc is in the rest state, s^{-1};

$$t_c = t_{rot} + t_{st}. \tag{7}$$

The distribution of heat flow between the disc and the friction lining:

$$\alpha_{T.P.2} = \left\{1 + \frac{\psi_{V_2} \cdot b_2 \cdot \lambda_1}{\psi_{V_1} \cdot b_1 \cdot \lambda_2}\right\}^{-1}, \tag{8}$$

$$\alpha_{TP\,1} = 1 - \alpha_{TP\,2}\,, \tag{9}$$

where $\alpha_{SP\,i}$ – coefficient of heat flow;

b_i – thickness of disc (1) or friction lining (2), mm;

$\psi_{V\,i}$ – correction coefficient of effective heat absobtion volume;

λ_i – the coefficient of thermal conductivity, W/(m·K).

Braking time (1):

$$t_{br} = \frac{I \cdot \pi \cdot n}{30\left(2F \cdot r_{ef} \cdot f - M_g\right)}. \tag{10}$$

The friction coefficient depends, mainly, on the temperature (Table 1).

Table 1

Additional data for calculation

The material of friction pair	The coefficient of heat capacity c, $\frac{J}{kg \cdot K}$	The coefficien of thermal conductivity λ, $\frac{W}{m \cdot K}$	Density $\rho, \frac{kg}{m^3}$	Wear intensity depending on temperature *I=f(T)*	Friction coefficient depending on temperature *f=f(T)*
"TWG"	1420	0.44	2000	$I=T\cdot10^{-7}+6\cdot10^{-6}$	$f=5T^2\cdot10^{-6}-0.0017T+0.254$
"Cosid 516"	435	0.23	2700	$I=9T\cdot10^{-8}+2\cdot10^{-5}$	$f=0.0002T+0.256$
"Dafmi"	1530	1.25	2900	$I=T\cdot10^{-7}+3\cdot10^{-5}$	$f=-2T^2\cdot10^{-6}+7T10^{-4}+0.266$
"ЭМ-1"	1000	0.34	2000	$I=5T\cdot10^{-7}-10^{-5}$	$f=0.287e^{-0.0026T}$
Disc (Ст 45)	480	47.00	7850	-	-

Results

There are a comparation of between the results of experimental work and the mathematical modeling. Each point in the graphs correspond to one experiment (6 hours) or calculation results under experimental work conditions.

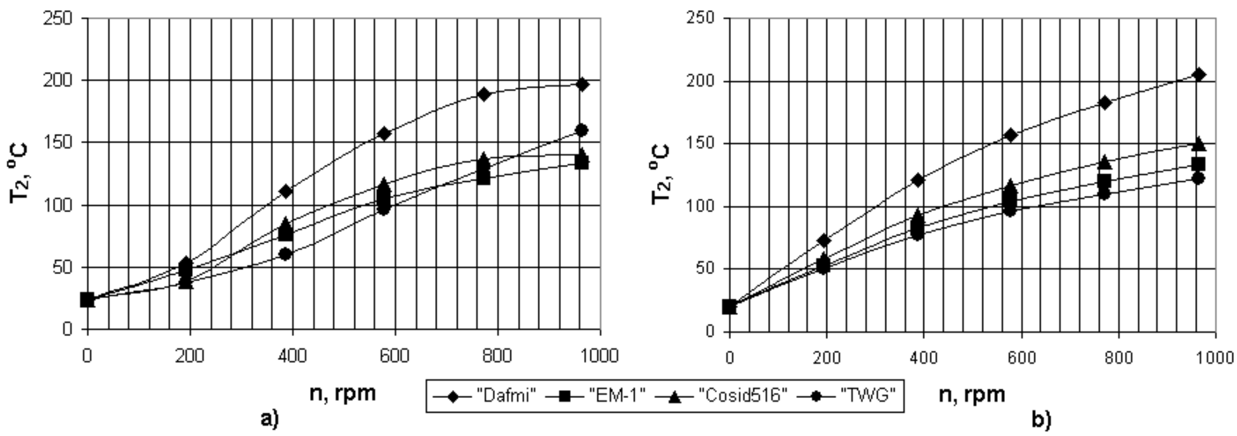

Fig. 2. **The friction lining thermal balance temperature T_2 depending on a disc revolutions n at the beginning of braking (p = 1.15 MPa, R = 2 time/min):**
a – results of experimental work; b – results of calculation

The mathematical model allows to calculate the temperature and the wear of friction lining with accuracy ± 10 % depending on all main braking process impact factors (Fig. 2, 3).

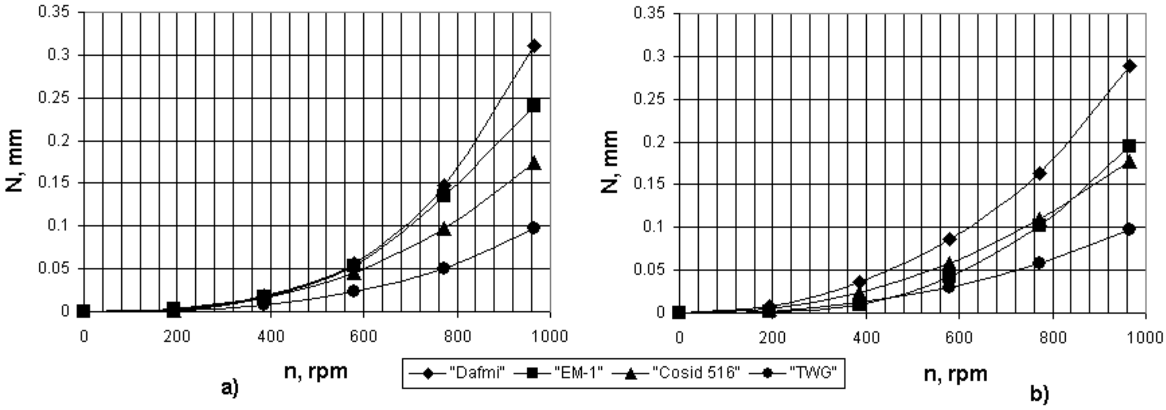

Fig. 3. **The wear of friction lining N depending on a disc revolutions n at the beginning of braking (p = 1.15 MPa, R = 2 time/min):**
a – results of experimental work; b – results of calculation

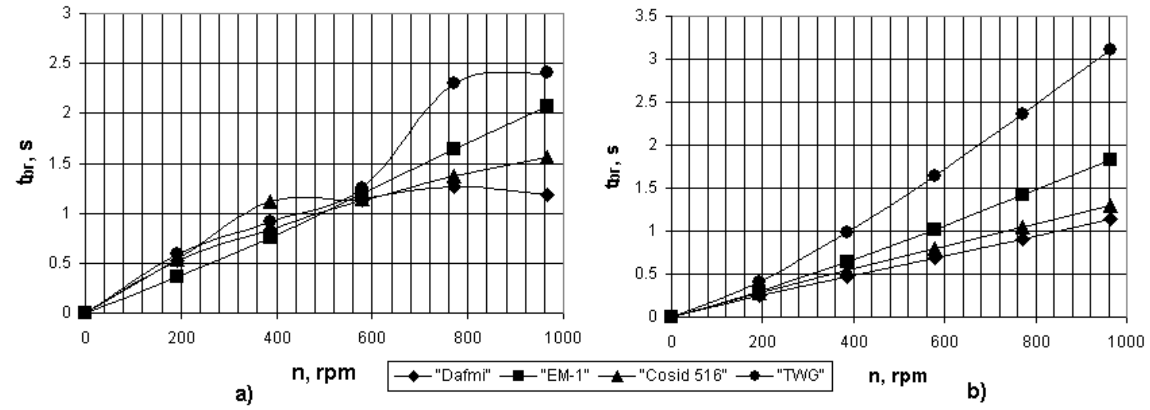

Fig. 4. **The braking time t_{br} depending on a disc revolutions n at the beginning of braking (p = 1.15 MPa, R = 2 time/min):**
a – results of experimental work; b – results of calculation

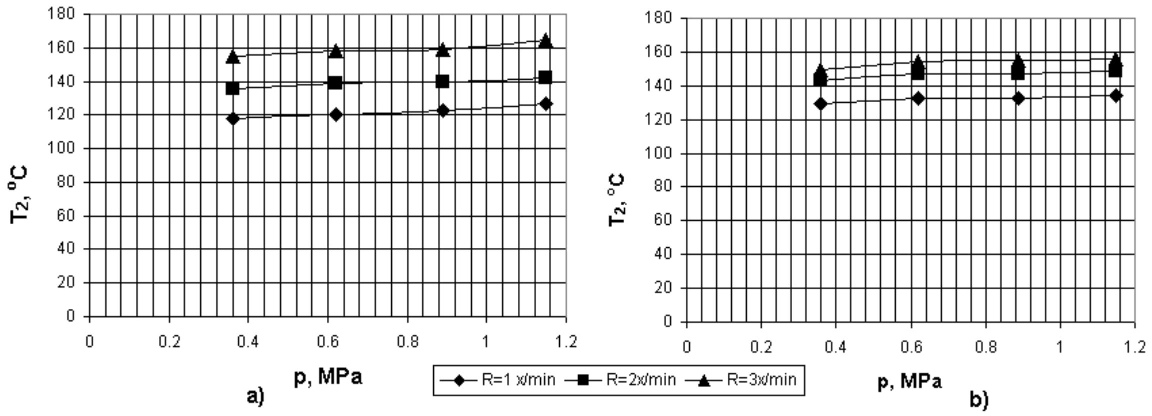

Fig. 5. **The friction lining ЭМ-1 thermal balance temperature T_2 depending on specific surface pressure p and braking intensity R (p = 1.15 MPa, R = 2 time/min):**
a – results of experimental work; b – results of calculation

The data from experimental work were approximate by equation and used in the mathematical model – it reduce the accuracy of calculation results, because the correlation between the experimental datas and generated equation is smaller than 1. One more reason of unaccuracy – the producers of friction linings give the thermophysical properties as constant number, but actually it changes a little bit depending on temperature.

Discussion

The calculation accuracy of braking time (Fig. 4) depends on accuracy of empirical equation describing friction coefficient depending on the temperature of friction lining (Table 1). The coefficient of correlation characterizes the accuracy of empirical equation.

The mathematical model allows calculating the temperature (Fig. 5) depending on the braking frequency R and the specific surface pressure p.

Although summary friction work is constant at each braking frequency R, the temperature T_2 increase if the specific surface pressure p increase, the explanation – disc stops faster if the specific surface pressure increase and rotate less, for that reason the disc cooling regime worsen and temperature increase (Fig. 5).

The mathematical model allows prognosticating successfully the temperature in the volume of disc or friction lining although during experimental work the temperature was measured 1 mm from the friction surface.

Conclusions

1. The mathematical model allows to calculate the braking time, temperature and the wear of friction lining in short time repeating braking regime with accuracy ± 10 % depending on all main braking process impact factors.
2. The most appropriate kind of friction lining for A/S „ JMR" disc brake production needs is "dafmi". These friction linings have the highest coefficient o friction f = 0.35 and the shortest braking time.

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