RESEARCH IN IMPACT OF CARGO VEHICLE LOAD WEIGHT ON BRAKING SYSTEM ELEMENT HEATING PROCESS IN SINGLE EMERGENCY STOPPING

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Abstract. The paper presents the results of the simulation tests of the impact of the total vehicle weight (with load) on heating discs and brake pads. It was necessary to develop a mathematical model describing the braking process of the truck to perform the simulation. The necessary material features, such as: coefficient of friction, thermal conductivity, density and thermal capacity at constant pressure were determined experimentally using laboratory equipment. Vehicle technical data were obtained from the manufacturer. The tests were carried out for constant initial speed - 90 $km \cdot h^{-1}$. For the purposes of this paper, a CAD model with the working elements of the braking system (disc and pad) was developed. The geometry of the system has been simplified relative to real parts. The simplification involved removal of unnecessary complicated geometry that does not have a significant impact on the final result of the tests, but may cause unnecessary compaction of the mesh. The finite element method (FEM) was used for the calculations. Appropriate equipment was also needed. In the case under consideration, a computer equipped with Comsol Multiphysics software was used. This is the software commonly used to calculate the distribution of temperature fields. As the result of calculations, this software allows to measure the temperature at any time and any point of the tested system. In this case, the temperature was measured at two points: inside the brake disc and inside the brake pad. In both cases at a distance of 0.05mm from the friction surface. The results clearly show that the mass of the vehicle has a direct effect on the maximum temperature of the friction linings. With overloaded vehicles, it reaches dangerously high values. Overheating the brakes may result in the fading phenomenon and loss of the stopping ability.

Keywords: simulation study, brakes, delivery vehicles, friction, wear.

Introduction

Nowadays, road transport is very extensive. The lifestyle of many people dictates the need to transport many types of goods [1]. A significant part of the whole of this industry is transport by small vehicles, which mass does not exceed 3.5t [2]. The advantage of using this category of vehicles is the possibility of driving them by persons authorized to drive passenger vehicles, lower fuel consumption and better manoeuvrability compared to a typical truck.

Locally used delivery vehicles usually have large (in volume) cargo spaces. This fact may encourage users to fully load, which often leads to exceeding the permissible total weight of the vehicle. This procedure results from the desire to save financial resources and time (transporting a larger load at one time can generate savings on fuel costs and reduce the delivery time, as the driver can make delivery to several pickup points during one trip). However, this may have dangerous consequences. The greater mass means that the moving vehicle has a higher kinetic energy [3]. Its emergency stop will therefore be much more difficult. The braking distance will be longer and there may be a risk of overheating the friction pair [4]. This, in turn, may lead to the occurrence of the so-called fading phenomena [5]. This phenomenon occurs, when the matrix of the friction material is sublimated causing formation of a gas cushion between the friction surfaces. The value of the coefficient of friction drops almost to zero then, preventing therefore the vehicle from stopping effectively [6; 7].

In delivery vehicles, the most commonly used brake system is the friction disc brake type. The working elements in this solution are: discs rotating together with the wheel, and brake pads located in the callipers [8]. When the driver decides to turn on the brakes, the contact pressure is increased, which translates into a friction torque. Because of that brakes can be considered as a kind of energy converter (there is a change of kinetic energy into heat energy [9; 10]). The energy in this form is discharged into the atmosphere and into the immediate surroundings (brake fluid, rim, tire, suspension components, etc.), which, unfortunately, can also lead to internal thermal and mechanical stress and damage as a result in extreme cases [11; 12].

Having the above in mind, the aim of the study was to develop a mathematical model of the delivery vehicle braking process and then check how the amount of transported load affects the heating process of the braking system working elements.

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Materials and methods

The object of the research was the CAD model of the pad and disc of a popular delivery vehicle in Poland. The CAD model was made on the basis of technical documentation gathered from the manufacturer, and then simplified (Fig. 1) (elements that do not have a significant impact on the test results (e.g. holes for pins), but could introduce unnecessary mesh density, and thus, among others, extension of the test time, were removed). It was also assumed that the nominal weight (ready to run) of the vehicle is 1975 kg.

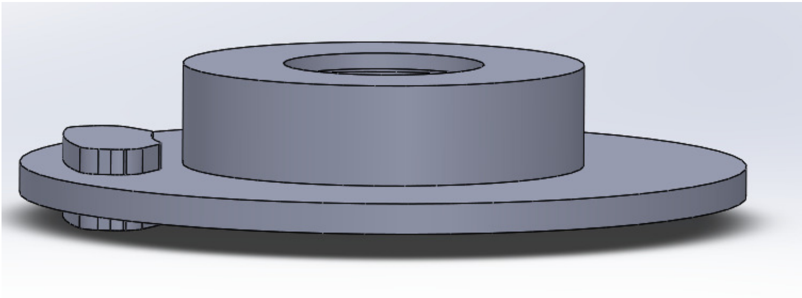

Fig. 1. **CAD model of the disc and brake pads (simplified)**

It was assumed that the disc material is cast iron. The data used in the study are summarized in Table 1. The pad is a composite material. Its main components are: metal fibers (Cu, Steel and Al) ≈13 %, organic fibers ≈18 %, matrix (resin) ≈21 %, solid lubricants (graphite, $MoS2$) ≈8 %, abrasive ($ZrSiO4$) ≈15 % and fillers ($BaSO4$, $Ca(OH)2$) ≈25 %. The whole creates a friction material, of which the most important characteristics determined experimentally are presented in Table 2.

Table 1

Selected properties of the brake disc material

Property	Value
Thermal conductivity	47 $W \cdot (m \cdot K)^{-1}$
Density	7870 $kg \cdot m^{-3}$
Thermal capacity at constant pressure	498 $J \cdot (kg \cdot K)^{-1}$

Table 2

Selected properties of the brake pad material

Property	Value
Thermal conductivity	146 $W \cdot (m \cdot K)^{-1}$
Density	2845 $kg \cdot m^{-3}$
Thermal capacity at constant pressure	1034 $J \cdot (kg \cdot K)^{-1}$

There are many available research methods [13]. In this work it was decided to perform simulation tests. The reason for the decision was the numerous of benefits, e.g. low cost of research and short time of obtaining results [14, 15]. Unfortunately, this method also has its drawbacks. It is necessary to develop a mathematical model that accurately reflects the actual physical phenomenon; otherwise the results may be subject to an error affecting the final results [16-18].

The COMSOL Multiphysics software was used in this study. It is software, which is using FEM (fine element method). The mesh imposed on the developed model consisted of about 7000 elements, mostly triangular shaped, which gave nearly 38000 DOFs (degrees of freedom).

It was decided that simulation tests will be performed for the emergency braking process from the initial speeds of 90 $km \cdot h^{-1}$ and 120 $km \cdot h^{-1}$ (maximum allowed speed on interstate and highway). The braking result in any case is complete stop of the vehicle. It was also assumed that the coefficient of friction of the tire to the road is 1.0, which as a result gives a constant braking delay of 9.81 $m \cdot s^{-2}$. For tires size 225/65R16C, the wheel dynamic radius equals 301mm. The lab experiment (on pin-on-disc test stand) allowed to determine the coefficient of friction between the pad and the disc. The research has shown that its value is 0.44. The determined value was used for simulation. Also, the ambient temperature of air was set on 20 ºC. The tests were carried out for different loading levels. The total weight of the vehicle was assumed: 3000 kg, 3500 kg and 4000 kg (for a partially loaded vehicle, at full load and overload). The following assumptions were also made: constant coefficients of friction, contact pressure for both pads is stable and homogeneous, the brake pad material is homogeneous and its contact with the disc is by full surface, constant braking deceleration and no external factors causing deceleration (such as road inconsistencies, air resistance).

The negative derivative of the vehicle kinetic energy can describe the brakes' retardation power [19]:

$$P = -\frac{d}{dt} \cdot \left(\frac{mv^2}{2} \right); 0 \le t \le t_s \tag{1}$$

or

$$P = -mR^2 \omega(t) \alpha; 0 \le t \le t_s \tag{2}$$

where m – vehicle mass, kg;
v – speed of the vehicle, m·s^{-1};
R – dynamic radius of the road wheel, m;
ω – angular velocity of the wheel, RPM
t – time, s;
t_s – braking time, s;
α – angular deceleration, rad·s^{-2}.

When the braking deceleration is constant:

$$\omega(t) = \omega_o + \alpha t; \; 0 \le t \le t_s \tag{3}$$

The vehicle stability analysis demonstrated that approximately 60 % of the total braking force comes from the front wheels (on the front axis roughly 15 % per each of the four brake pads). Therefore, the assumption was made that there is identical braking force on each of the brake pads. The relation between the complete vehicle braking force and the braking force of one wheel is as follows:

$$F_b = \frac{100\% \cdot F_{fb}}{2 \cdot 15\%} \approx 3.33 F_{fb} \tag{4}$$

where F_{fb} – braking force generated by one front wheel, N;
F_b – total braking force of the vehicle, N.

Heaving the above in mind, the retardation power can be described as:

$$P = -3.33 \iint f_f \cdot dA \cdot v_d \tag{5}$$

where f_f – frictional force per surface unit;
$v_d = \omega(t) \cdot r$ – brake disc linear speed at radius r, m·s^{-1};
A – disc and pad contact surface area, m^2.

The braking retardation power could be also expressed by the equation [20]:

$$P = f_f(t) \cdot \omega(t) \iint r_m \cdot dA; \; 0 \le t \le t_s \tag{6}$$

where r_m – distance from the pad centre of mass to the disc rotation axis, m.

When we compare these two equations, it is possible to determine the f_f coefficient:

$$f_f = -\frac{mR^2 \alpha}{3.33 r_m A} \tag{7}$$

Assuming that the vehicle's deceleration occurs only through the action of the brake disc and pad, the heat flux can be expressed using the equation [21]:

$$q(r,t) = -f_f \cdot v_d(r,t); \; 0 \le t \le t_s; \; r_0 \le r \le r_m \tag{8}$$

or

$$q(r,t) = -\frac{mR^2\alpha}{3.33 r_m A} r(\omega_o + \alpha t);\ 0 \le t \le t_s;\ r_0 \le r \le r_m \tag{9}$$

The contact pressure is possible to be determined from the Amonton-Coulomb friction law [22; 23], which in the analysed case is as follows:

$$p = \frac{P}{\mu \cdot v} \tag{10}$$

where μ – coefficient of friction between the disc and pad;
F_b – total braking force of the vehicle, N.

The study takes also in account the heat exchange between the disc and pad. It can be expressed by modifying the equation from [24]:

$$\rho \cdot C_p \frac{\partial T}{\partial t} + \nabla \cdot (-k \cdot \nabla T) = Q - \rho \cdot C_p \cdot u \cdot \nabla T \tag{11}$$

$$T(r,\beta,0^+,t) = T(r,\beta,0^-,t);\ \ (r,\beta) \in A \tag{12}$$

where k – thermal conductivity, W·(m·K)$^{-1}$;
C_p – thermal capacity, J·(kg·K)$^{-1}$;
u – heat flux rate, W·m^{-2};
Q – heating power per density unit;
ρ – density, kg·m^{-3};
T – temperature, ºK;
β – circumferential coordinate, rad.

During the action of the brakes (through convection and radiation) the following amount of heat is released:

$$q_d = -h(T - T_r) - \varepsilon\sigma(T^4 - T_r^{\,4}) \tag{13}$$

where $(T–T_r)$ – temperature difference between the friction material and ambient temperature, K;
h – convection coefficient, W·(m^2·K)$^{-1}$;
ε – emissivity of the material;
σ – Stefan-Boltzman constant.

The relation between the speed of the vehicle and the convection coefficient can be described as follows:

$$h = \frac{0.037k}{l}\left(\frac{\rho \cdot l \cdot v}{\mu_v}\right)^{0.8} \cdot \left(\frac{C_p \cdot \mu_v}{k}\right)^{0.33} \tag{14}$$

where μ_v – viscosity, Pa·s;
l – disc diameter, m.

Results and discussion

The direct result of the simulation tests is temperature data of the tested elements. The measuring points were set 0.1 mm below the contact surface in the geometrical center of the cooperation surface. The temperature profiles of the brake disc and pad are shown in Fig. 2 and Fig. 3. In addition, the amount of energy generated in the braking process was estimated for each case (Tab. 3).

The obtained data were compared with the results of similar works by other researchers [25-27]. Although the results were presented in a slightly different form, the values are proportional. This allowed to assess the correctness of the research.

The obtained results show how high temperatures are achieved in the braking system during emergency braking. The highest temperature reached by the brake pads (just over 734 ºK) was when overloaded vehicle (weighing 4 t) was stopping from an initial speed of 120 km·h^{-1}. The maximum was reached only 1.95 s from the start of braking. Brake discs in the same operating conditions reached a slightly lower temperature – 689 ºK. It occurred 0.05s earlier, 1.9 s from the beginning of braking. The characteristic collapse on the brake pad temperature charts results from the contact of materials with different properties (heat conduction from one element to the other).

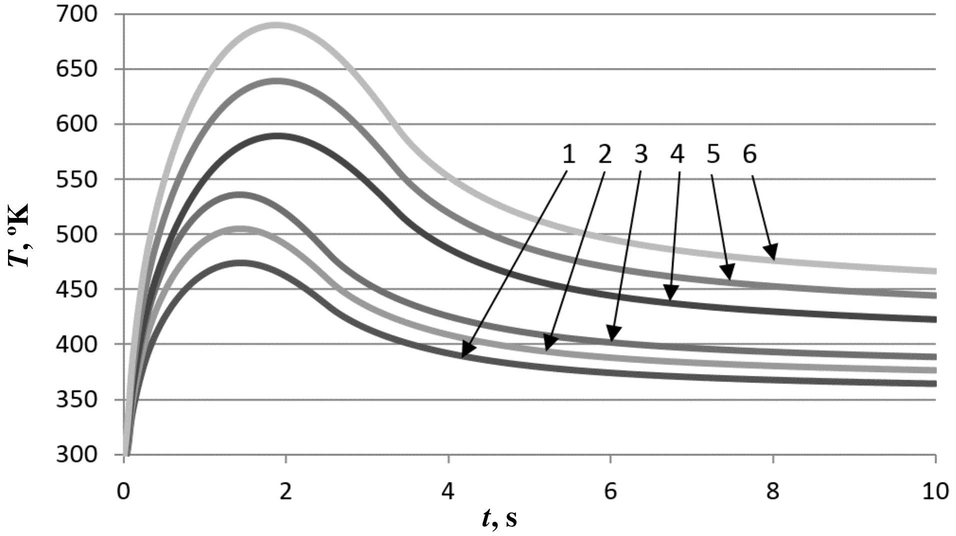

Fig. 2. **Temperature profile of brake disc:** 1 – initial speed 90 km·h^{-1}, total mass 3000 kg; 2 – initial speed 90 km·h^{-1}, total mass 3500 kg; 3 – initial speed 90 km·h^{-1}, total mass 4000 kg; 4 – initial speed 120 km·h^{-1}, total mass 3000 kg; 5 – initial speed 120 km·h^{-1}, total mass 3500 kg; 6 – initial speed 120 km·h^{-1}, total mass 4000 kg

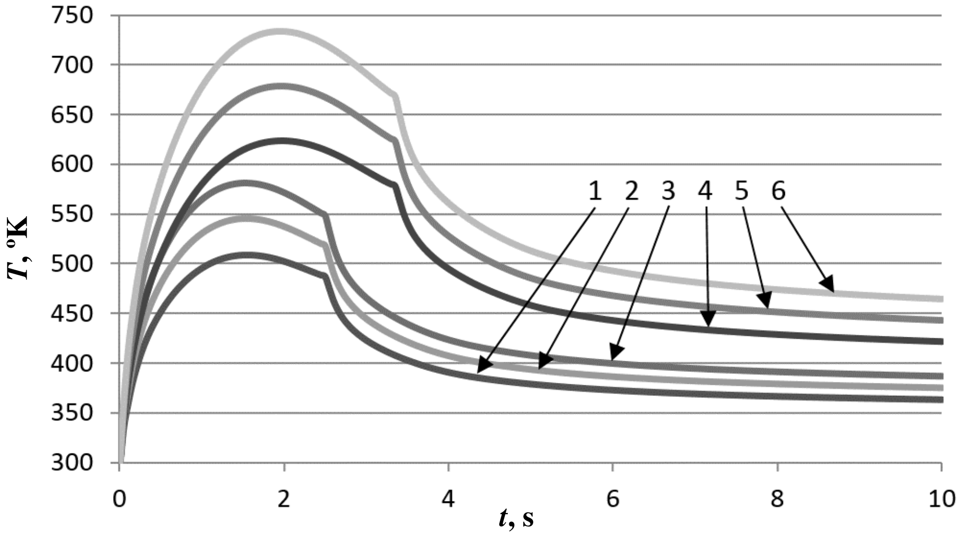

Fig. 3. **Temperature profile of brake pad:** 1 – initial speed 90 km·h^{-1}, total mass 3000 kg; 2 – initial speed 90 km·h^{-1}, total mass 3500 kg; 3 – initial speed 90 km·h^{-1}, total mass 4000 kg; 4 – initial speed 120 km·h^{-1}, total mass 3000 kg; 5 – initial speed 120 km·h^{-1}, total mass 3500 kg; 6 – initial speed 120 km·h^{-1}, total mass 4000 kg

Table 3

Amount of energy generated in the braking process

Initial speed, km·h^{-1}	Total mass, kg	Generated energy, W
90	3000	144155
	3500	168196
	4000	192233
120	3000	256310
	3500	299038
	4000	341766

It should be remembered that the brake pads are made of a composite material, which resin is the matrix. It is a material that has limited resistance to heat load. Depending on the type of resin, damage to its structure may occur when heated to 400-500 ºC [28; 29]. The weakest of them burn already at 260 ºC (533 ºK) [30]. Therefore, attempts are being made to develop some modifications to improve resistance to high temperatures [31]. However, in the case of conventional solutions, intensive heating of rubbing couple causes the occurrence of the fade phenomenon. In the present case, it turns out that 2/3 of the stopping processes under investigation are threatened by the appearance of a gas cushion in the friction couple. Safe driving conditions can be considered traveling with a fully loaded vehicle at a speed of 90 $km \cdot h^{-1}$. Increasing the speed or load carries the risk of overheating the brakes.

Conclusions

1. A mathematical model of the delivery vehicle braking process was developed in the paper.
2. Performed simulation tests based on the developed model allowed to measure the temperature of the braking system components at any time and at any point.
3. The highest temperatures were achieved when braking from 120 $km \cdot h^{-1}$ with a total weight of 4000 kg.
4. Overloading the vehicle or moving at excessive speeds carries the risk of overheating the brakes, which can lead to fade.

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