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THE INFLUENCE OF IMPORTANT FACTORS ON THE DISTRIBUTION OF HEAT FLOWS IN ELEMENTS OF DRUM BRAKES OF VEHICLES

Summary. *The movement of motor vehicles at high speeds is impossible without a braking system capable of ensuring high braking efficiency. It has been established that the most unstable link of the braking system is the brake mechanism, since from the energy point of view, braking with friction brakes is the process of converting part of the mechanical energy of the motor vehicle into heat.*

Braking is a long process during which many counterbody parameters change, in particular, thermophysical parameters due to temperature changes, friction coefficient, etc.

If, under these circumstances, the surface and volume temperatures exceed the permissible values, then the frictional properties of the friction pairs and the conditions of the interaction of the parts change, which leads to a change in the characteristics of the brake mechanisms and the brake system as a whole. The standards of most countries and international prescriptions regulate braking performance meters not only for one-time emergency braking with cold brakes but also for emergency braking performed after the conversion of a given amount of energy into heat during a given time. It was found that the preservation of the necessary braking efficiency after the conversion of a given amount of energy into heat will be ensured only if the braking system has sufficient energy capacity.

The object of the research is the distribution of heat flows in the elements of the brake mechanism, which determine the critical temperature of the friction surfaces. It was established that F. Charron's formula cannot correctly estimate such a distribution due to taking into account only the thermophysical properties of materials of friction pairs. It is shown that the influence of the design parameters of the brake and its modes of operation on the distribution of heat flows in the drum brake of a motor vehicle can also be estimated on grid thermal models with the involvement of the "Fourier-2xyz" software complex.

Keywords: *motor vehicle, drum brake mechanism, test modes I, heat flow distribution coefficient, thermal model.*

1. INTRODUCTION

In conditions of intensive vehicular traffic in various road conditions (plains and mountain roads), the energy load of brake systems and transmission elements differs sharply [1–3]. Therefore, such a condition requires a comprehensive study to improve the design and increase operation efficiency [4–6].

The experience of operating motor vehicles shows that their reliability and driving safety depends on the thermal stress of braking mechanisms, which in turn is determined by the energy load of the braking system. One of the most loaded modes of the brakes is the cyclic mode of their operation, which is typical for the operation of automatic transmission in urban conditions [7, 8]. Therefore, it is no coincidence that the international method of checking the effectiveness of vehicles` brakes also includes repeated short-term braking (test I) [9].

When determining the thermal state of friction pairs of brake mechanisms, it is assumed that for a set combination of counterbodies, the coefficient of distribution of heat flows in the brake is constant, since only their thermophysical parameters are taken into account. But recent research on disc brake mechanisms has shown [10] that this coefficient is variable, if we also consider the influence of the design and operational parameters of the brakes. Therefore, the question of a similar study regarding drum brakes of vehicles has become relevant.

2. RESEARCH STATEMENT

The study aims to identify the regularity of the change in the heat flow distribution coefficient in the car's drum brake elements.

The following tasks should be solved to achieve this goal, depending on its structural and operating parameters:

– creation of a thermal grid model of the brake mechanism of the front wheel of the vehicles;
– simulation of test mode I of brake mechanisms of vehicles in the software environment "Fourier-2xyz";
– determination of patterns of influence of design and mode parameters of drum brakes on the flow distribution coefficient in its elements;
– creation of practical recommendations for taking into account research results.

3. THERMAL MODEL OF A DRUM BRAKE

The distribution of temperature in space and time in brake mechanisms in the presence of internal heat sources and the independence of thermophysical coefficients from temperature is described by the nonlinear equation of thermal conductivity [11]:

$$\frac{\partial}{\partial x}\left[\lambda(x,y,z,T)\frac{\partial T}{\partial x}\right]+\frac{\partial}{\partial y}\left[\lambda(x,y,z,T)\frac{\partial T}{\partial y}\right]+\frac{\partial}{\partial z}\left[\lambda(x,y,z,T)\frac{\partial T}{\partial z}\right]+Q=c\rho(x,y,z,T)\frac{\partial T}{\partial \tau}, \quad (1)$$

where x, y, z – current coordinates; $\lambda(x, y, z)$ – thermal conductivity coefficient; T – temperature; $Q(x, y, z)$ – heat flow density; with$\cdot\rho(x, y, z)$ – volumetric heat capacity; τ – time.

In addition to the basic equation, the mathematical model of the phenomenon of thermal conductivity should also descript the initial distribution of temperatures and ratios indicating the nature, magnitude and place of application of extreme thermal influences [11]. The latter takes place on the surfaces of the brake friction pairs, where the generated energy is absorbed at the test modes (limit conditions of the 2nd kind) and their outer surfaces, which are evaluated by heat transfer coefficients (limit conditions of the 3rd kind).

In particular, for the vehicle category M_3, the preliminary test stage I is characterized by 20 braking cycles from a speed of V_1 = 60 km/h to V_2 = 30 km/h. At the same time, the energy absorbed by the brakes is [12]:

$$E_1 = 20 \cdot G_a \cdot (V_1^2 - V_2^2)/(2 \cdot 3{,}6^2) = 2083 \cdot G_a, \quad (2)$$

where G_a is the mass of vehicle, kg; V_1 and V_2 are regulated by speed regulations, km/h.

In general, the heat transfer coefficient α depends on the shape and dimensions of the cooled surface, speed, temperature and thermophysical properties of the cooling medium, body temperature, and other factors [10], which is difficult to determine. Therefore, the values of α for brakes given in the literature and obtained analytically are characterized by significant discrepancies. The most reliable is to obtain values of α by solving the inverse problem of thermal conductivity [13] by mathematical modeling based on the results of full-scale tests of vehicle brakes, where the average value of α = 34.8 W/m^2 degr [10].

Since this goal concerns non-stationary contact thermal problems in regions of non-classical shape under heterogeneous boundaries and complex initial conditions, it does not have an exact analytical solution. Therefore, in engineering practice, along with the experiment, approximate ones are used analytical and, especially, numerical methods, which have turned into a powerful mathematical apparatus for solving field theory problems.

Given this circumstance, the development of three-dimensional models of brake mechanisms is a significant step forward [10], the advantage of which was the possibility of simultaneous investigation of temperature fields in the drum or disk, lining or pad of the mechanism.

The calculation module [10], created based on the software package, was used to solve Equation (1). "Fourier-2xyz", which allows solving two-dimensional and three-dimensional problems of heat transfer in dialog mode and obtaining results in a convenient and visual form for use.

For this, the sector of the drum brake is modeled on the finite-difference grid (Fig. 1) according to the Z coordinate, which makes it possible to solve the volumetric problem according to the mentioned method [14].

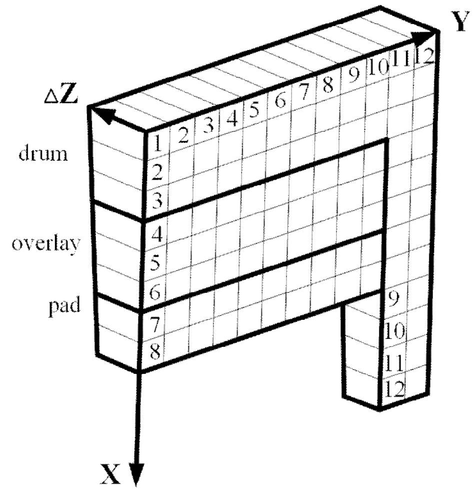

Fig. 1. Sector of the mesh model of the drum brake mechanism of the BOGDAN A091 bus

4. SIMULATION OF THE EFFECT OF IMPORTANT FACTORS ON THE COEFFICIENT OF DISTRIBUTION OF HEAT FLOWS IN A DRUM BRAKE

As mentioned above, an important characteristic that determines the temperature regime of the brake mechanism is the coefficient of heat flow distribution which is the ratio of the heat flow entering the studied element Q_{el} to the total amount of heat generated in the friction pair $\sum Q$, i.e.

$$m = \frac{Q_{el}}{\sum Q}. \tag{3}$$

Parts of friction pairs are made of materials that differ significantly by thermophysical properties (Table 1).

Table 1

Thermophysical characteristics of drum brake elements [15]

Parameters	Units of measurement	Value	
		asbestos polymer overlays	cast iron drum
1. Density	g/cm^3	2–2.5	7.3
2. Specific heat capacity	kJ/kg degrees	0.88–1.17	0.5
3. Thermal conductivity	W/m degree	0.4–0.52	29

Therefore, as a rule, the coefficient of distribution of heat flows m is determined considering the thermophysical characteristics of friction pairs according to the formula of F. Charron [10]:

$$m = \frac{\sqrt{\lambda_2 c_2 \gamma_2}}{\sqrt{\lambda_1 c_1 \gamma_1} + \sqrt{\lambda_2 c_2 \gamma_2}}, \tag{4}$$

where m is the heat flow distribution coefficient; λ, c, γ – thermophysical properties of materials (1-drum, 2 – lining).

Having analyzed F. Charron's formula, we should assume that for this combination of brake mechanism materials, the coefficient of heat flow distribution is constant, which is highly doubtful. Therefore, to study the influence of various factors on the coefficient of distribution of heat flows during tests of type I by the method of sequential braking, studies on the thermal models, which are described below, were carried out. The analysis of the curves (Fig. 2), constructed based on the results of the study of the first, sixth and twentieth braking, shows that the brake drum is mainly a heat accumulator.

a *b*

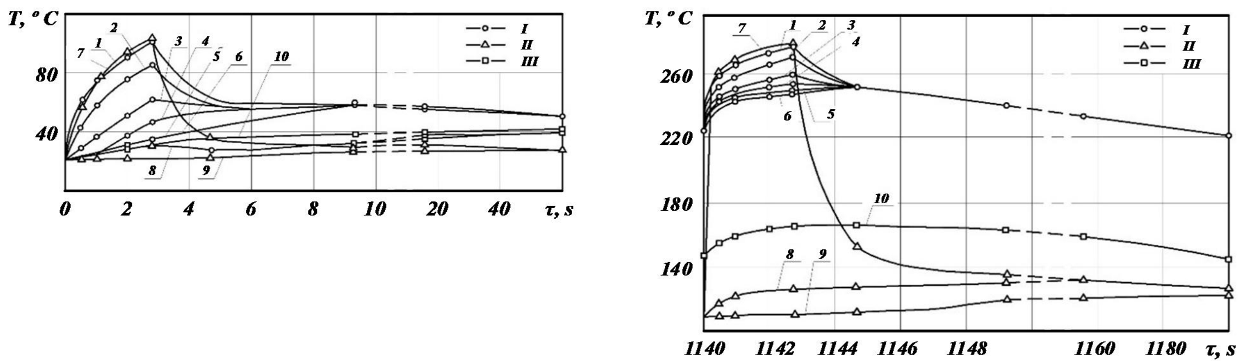

Fig. 2. Temperature change of elements of the front brake mechanism of the BOGDAN A091 bus during tests I: a – 1st cycle; b – 20th cycle; I – brake drum; II – overlay; III – side wall of the drum; 1 and 7 – friction surfaces; 2,3,4,5 and 6 – respectively at a distance of 1.5 mm, 4.5 mm, 10.5 mm, and 12 mm from the friction surface; 8 and 9 – respectively at a distance of 2.1 mm and 6.4 mm from the friction surface; 10 – the middle of the side wall of the drum

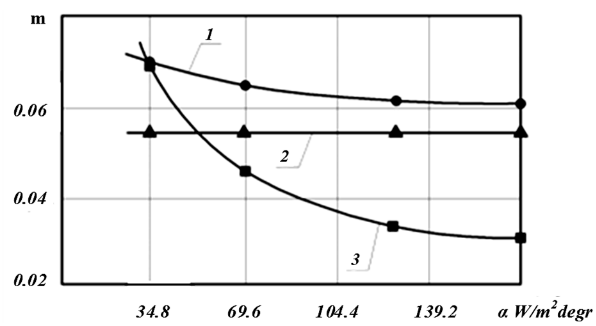

Fig. 3. Dependence of the coefficient of heat flow distribution on the heat transfer coefficient (test I, Q=251.2 kJ):
1 – α/α' = 1.43 = const; 2 – according to F. Charron's formula; 3 – α' = 24.4 W/m^2 degr = const; α = var

At the moment of braking, the friction surface of the drum and the pad have almost the same temperature (see Fig. 2, curves 1 and 7), but during cooling after 3–5 seconds, they significantly differ. Such a difference in the flow of temperature change curves during cooling down is explained by a significant difference in the amount of heat accumulated. The simulation results (Fig. 3) show that with the same relative increase in the coefficient of heat transfer from the drum and lining, the coefficient of heat flow distribution changes insignificantly. If you intensify the heat removal only from 34.8 W/m^{2}degr to 174 W/m^{2}degr, then the amount of heat perceived by the overlay decreases by approximately 45 %.

A comparison of the results of studies (Fig. 3, curve 1) carried out under nominal conditions with the results of calculations carried out according to formula (4) (Fig. 3, curve 2) shows that the coefficient of heat flow distribution for α = 34.8 W/m^{2}degr exceeds the value by 27 %, and for α = 174 W/m^{2}degr – by 10–12 %.

It is also interesting to investigate the change in the coefficient of distribution of heat flows depending on the amount of heat released and the thickness of the drum wall. According to the simulation results presented in Fig. 4, with an increase in the thickness of the drum wall, the amount of heat perceived by the lining, and the coefficient of distribution of heat flows decrease. In particular, an increase in wall thickness from 6 mm to 13 mm causes, other things being equal, a decrease in the amount of heat accumulated by the overlay by 60–65 %.

A decrease in the coefficient of distribution of heat flows is also observed with an increase in the amount of generated heat (see Fig. 4). It can be explained by the fact that as Q increases, the role of the side wall of the drum increases and it begins to affect the distribution of heat flows in the same way as an increase in the thickness of the drum wall. Simultaneously with the increase in the mass in which the generated heat is distributed, the heat transfer surface of the drum increases due to the heat dissipating surface of the side wall. At the same time, the product $\alpha \cdot F_{ox}$ increases with a constant value $\alpha' \cdot F'_{ox}$.

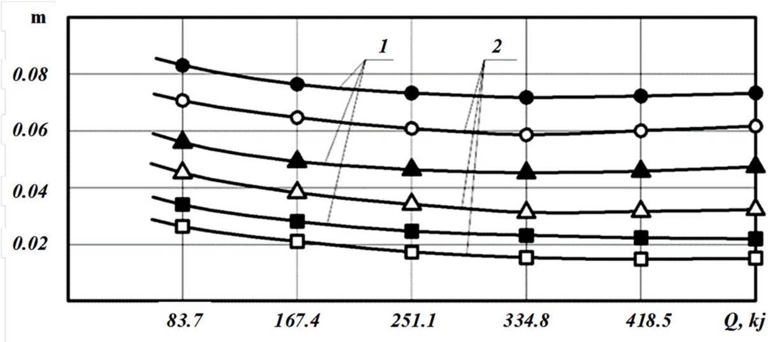

Fig. 4. Dependence of the heat flow distribution coefficient on h and Q during tests I: 1. α = 34.8 W/m²degr; 2. α = 174 W/m²degrees;
–■– h=0.013m; –▲– h=0.01m; –●– h=0.006m;

A distinctive feature of the repeated short-term operation mode of the brake mechanism, which affects the distribution of heat flows between the parts of the friction pairs, is that in this case, the heating and cooling processes alternate. During cooling down, the generated heat is distributed in the masses of the parts and is partially discharged into the environment. The distribution of the heat generated during each subsequent braking will depend on the average capacitive temperatures of elements at the end of the previous "braking-acceleration" cycle. Therefore, it can be assumed that the heat flow distribution coefficient will change from cycle to another.

The obtained results allow us to state that when solving specific engineering problems, it is necessary to pre-estimate the coefficient m, which allows us to predict the surface and capacitive temperatures of friction pairs more accurately.

5. CONCLUSIONS

1. We have established the inability of F. Charron's formula to estimate the distribution of heat flows in the elements of the drum brake correctly because of considering only the thermophysical parameters of the materials of the friction pairs.
2. A finite-difference grid model of the drum brake was created in the "Fourier-2xyz" software environment, which allowed obtaining the flow of temperature curves of heating and cooling of its counterbodies during tests I of the bus BOGDAN A091.
3. Modeling of the process made it possible to obtain the nature of the influence of the heat transfer coefficient, the thickness of the walls of the counterbodies and the intensity of heat release in friction pairs on the coefficient of distribution of heat flow in the elements of the drum brake during the vehicle tests.
4. The results obtained should be considered during the design of braking mechanisms and the preliminary assessment of the heat flow distribution coefficient in them, which depends on the surface and capacitive temperatures of friction pairs, which determine the effectiveness of vehicle braking.

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ВПЛИВ ВАГОМИХ ЧИННИКІВ НА РОЗПОДІЛ ТЕПЛОВИХ ПОТОКІВ В ЕЛЕМЕНТАХ БАРАБАННИХ ГАЛЬМ ТРАНСПОРТНИХ ЗАСОБІВ

Анотація. *Зазначено, що рух автотранспортних засобів на значних швидкостях неможливий без гальмівної системи, здатної забезпечити високу ефективність гальмування. Встановлено, що найбільш нестійкою ланкою гальмівної системи є гальмовий механізм, оскільки з енергетичної точки зору гальмування фрикційними гальмами становить процес перетворення у тепло частини механічної енергії автотранспортного засобу.*

Гальмування – власне тривалий процес, упродовж якого змінюються багато параметрів контртіл, зокрема, теплофізичні параметри внаслідок температурних змін, коефіцієнт тертя тощо.

Якщо за цих обставин поверхневі та об'ємні температури перевищують допустимі значення, то змінюються фрикційні властивості пар тертя й умови взаємодії деталей, що обумовлює зміну характеристик гальмових механізмів та гальмівної системи загалом. Стандартами більшості країн та міжнародними приписами регламентовано вимірники гальмівних властивостей не тільки за одноразових екстрених гальмувань холодними гальмами, але й за екстрених гальмувань, що здійснюються після перетворення в тепло заданої кількості енергії впродовж заданого часу. З'ясовано, що збереження необхідної ефективності гальмування після перетворення у тепло заданої кількості енергії буде забезпечено лише у випадку, коли гальмівна система володіє достатньою енергоємністю.

Об'єктом дослідження є розподіл теплових потоків в елементах гальмового механізму, які визначають критичну температуру поверхонь тертя. Встановлено неспроможність формули Ф. Шаррона коректно оцінити такий розподіл через врахування тільки теплофізичних властивостей матеріалів пар тертя. Показано, що на сіткових теплових моделях із залученням програмного комплексу "Фур'є-2xyz" можна також оцінити вплив конструктивних параметрів гальма та режимів його роботи на розподіл теплових потоків у барабанному гальмі автотранспортного засобу.

Ключові слова: *автотранспортний засіб, барабанний гальмовий механізм, режими випробувань I, коефіцієнт розподілу теплових потоків, теплова модель.*