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INFLUENCE OF MODE AND DESIGN FACTORS ON THE TEMPERATURE CONDITION OF AUTOMOBILE DRUM BRAKES

Summary. *The current state of the highway pavement makes it possible to implement prompt transportation operations. These circumstances lead to the movement of vehicles at high speeds, which is impossible without a braking system capable of ensuring high braking efficiency and optimal flow of the process from the standpoint of stability and controllability.*

One of the main requirements for a modern automatic transmission braking system is the stability of the initial parameters, that is, parametric reliability. Therefore, it is important to have data on the brakes' operating modes and energy consumption. Only with such data is it possible to create a braking system whose output characteristic will be sufficiently stable under conditions of high energy load.

Therefore, it is no coincidence that the international methodology for testing the effectiveness of vehicle brakes (UNECE Rule 13) provides for Test I, which is characterized by cyclic braking (urban conditions), and Test II, which is characterized by prolonged braking (mountain conditions).

The brake mechanism is the most unstable link of the brake system; one way to increase its efficiency is to ensure sufficient energy capacity, which is limited by the temperature of the friction surface.

The object of the study is the question of the equivalence of the change in the drum radius and the width of the friction belt of the brake, taking the invariance of the temperature of the friction surface under the selected test mode as the criterion of equivalence. It is shown that the role of the drum's side wall on the brake's temperature mode under different test modes can also be evaluated on grid thermal models with the involvement of the "Fourier–2 x,y,z" software complex.

The effect of the heat transfer coefficient on the temperature mode of the brake due to the consideration of the gap between the drum wall and the wheel rim is shown. Derived formulas for determining the friction belt's equivalent width under the equality of heat flows, masses, and cooling surfaces.

Keywords: *motor vehicle, drum brake mechanism, structural parameters of the drum, test modes I and II, thermal model.*

1. INTRODUCTION

Improvements in the quality of materials and improvement of tire designs provide the possibility of reducing their sizes with the same loads on the wheel. As a result, the volume of space in which the brake mechanism must be inserted decreases [1, 2]. At the same time, the energy load of braking mechanisms is continuously increasing [3, 4], and the task of choosing their design parameters, considering the need to ensure sufficient energy capacity [5, 6], is becoming increasingly important.

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One of the most heavily loaded modes of the brakes is the long one, typical for the operation of vehicles in mountain conditions [7]. Still, it should be noted that the cyclic mode of their operation (urban conditions) is also characterized by significant temperature gradients that affect the friction coefficient of the brake counterbodies [8]. Therefore, the modes mentioned above are provided by the international method of testing the effectiveness of automotive vehicle brakes [9] – test I (cyclic) and II – long. Test I of the automotive vehicle for the M3 category involves 20 "acceleration-braking" cycles up to a speed of 60 km/h with a deceleration to 30 km/h with a subsequent check of the braking efficiency, which should not be lower than 20 % of the initial (main stage). Test II involves the movement of the automotive vehicle on a road 6 km long with a gradient of 6 % at a speed of 30 km/h with a subsequent check of the braking efficiency, which should not be lower than 25 % of the initial (main stage).

The test modes are taken as the basis of their influence on the design parameters of the brakes from the point of view of the temperature state of the friction pairs [10], which is of actual importance.

2. RESEARCH STATEMENT

The purpose of the study is to identify the influence of the design parameters of the drum brake on its temperature mode during tests I and II.

The following tasks should be solved to achieve this goal:

– to create a thermal model of a drum brake mechanism;
– to simulate the test I and II of automotive vehicle brake mechanisms in the "Fourier-2 x,y,z" software environment;
– to determine the patterns of influence of structural and mode parameters of drum brakes on their temperature state;
– to create practical recommendations regarding research results.

3. THERMAL MODEL OF A DRUM BRAKE

The nonlinear thermal conductivity equation describes the temperature distribution in space and time in braking mechanisms in the presence of internal heat sources and the independence of thermophysical coefficients from temperature [11].

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

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

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

Regarding the generated energy on the friction surfaces of the brakes, the dependencies that consider the differences in conducting tests I and II can be used [12].

Values α for brakes given in the literature and obtained analytically are characterized by significant discrepancies. The most reliable way to obtain values α is by solving the inverse problem of thermal conductivity [13] by mathematical modeling based on the results of full-scale test vehicle brakes, where the average values α= 34.8 W/m^2 degrees [10].

Since this goal concerns non-stationary contact thermal problems in regions of non-classical shape under heterogeneous limit and complex initial conditions, it does not have an exact analytical solution and requires mathematical modeling.

Therefore, the calculation module [10] created based on the software complex "Fourier–2 x,y,z" was used to solve equation (1). It allows for solving two-dimensional and three-dimensional heat transfer problems in dialog mode and obtaining results conveniently and visually.

For this purpose, the sector of the drum brake is modeled on the finite-difference grid (Fig. 1) along the Z coordinate, the materials for the counterbodies of which are cast iron (drum) and asbopolymer overlays [10]. It makes it possible to solve the volumetric problem using the abovementioned method [14].

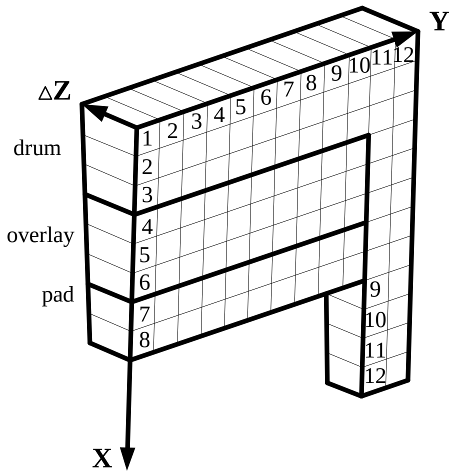

Fig. 1. Sector of the grid model of the drum brake mechanism

4. SIMULATION OF THE INFLUENCE OF MODE AND DESIGN FACTORS ON THE TEMPERATURE STATE OF DRUM BRAKES

It is known that the brake mechanism designer can vary within appropriate limits by the radius of the drum r_b, the width of the friction belt b, the thickness of the drum wall h, and the size of the cooling surface F_{cool}. In addition, the heat transfer coefficient α can be changed by the appropriate organization of air flows. Therefore, when designing a brake mechanism, it is essential to compare the temperatures of the friction surfaces at different values of the heat fluxes Q, b, r_b, h, α, and F_{cool}. It allows the application of the method mentioned above of mathematical modeling, which allows the simulation of long-term braking and braking, the duration of which is measured in seconds. In addition, the method allows the simulation of repeated short-term modes observed during the preliminary stage of test I by the method of cyclic braking. It should be noted that this mode is quite stressful due to the occurrence of significant temperature gradients that affect the friction coefficient of the brake counterbodies [10]. It made it possible to evaluate the influence of the design parameters of the braking mechanisms on the temperatures of the friction surfaces.

On cars of subcategories N_2 and N_3, M_2 and M_3 are widely used pad-drum brake mechanisms, the drum radius of which is within 180–220 mm. Therefore, the range specified above was chosen to evaluate the influence of the drum radius on the temperature mode. According to the simulation results, graphs of the dependence of the average temperature of the friction surface at the end of the preliminary stage of test I in a function from the radius of the drum were built. If we assume that the change in the radius of the drum does not significantly affect the heat transfer coefficient, then with an increase of r_b, the temperature of the friction surface drops noticeably (Fig. 2).

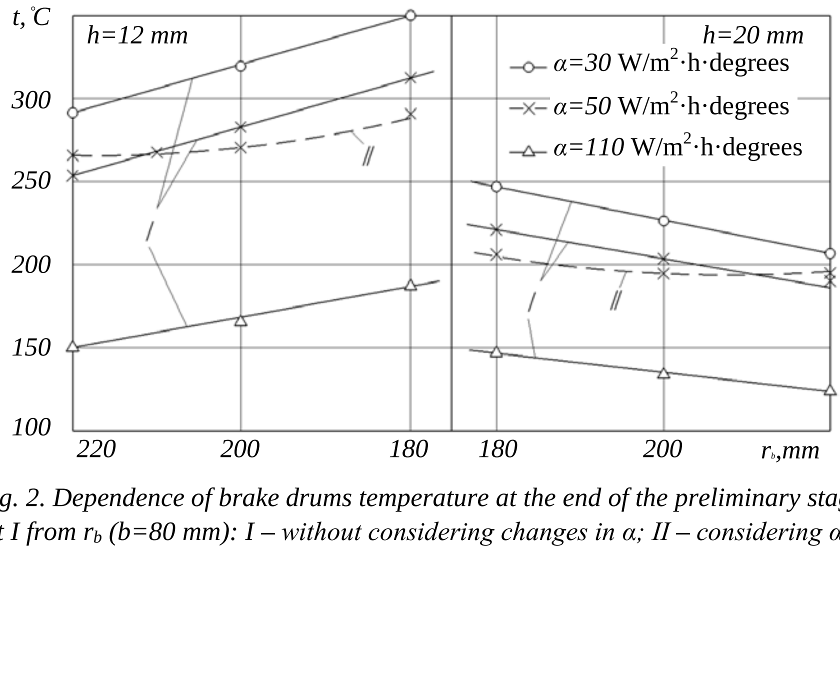

Fig. 2. Dependence of brake drums temperature at the end of the preliminary stage of test I from r_b (b=80 mm): I – without considering changes in α; II – considering α=f(S)

In particular, the increase of r_b from 180 to 220 mm will decrease the temperature at the end of the preliminary stage of test I from 352 to 290 ºC, that is, by 17 %. However, this conclusion cannot be considered as strict. Under other different conditions, a change in the drum radius leads to a change in the gap *S* between the drum and the wheel rim, resulting in a change in the heat transfer coefficient. The preliminary stage of test I, carried out by the method of cyclic braking, is long enough (15–20 min). As a result, the heat transfer conditions significantly affect the surface and volume temperatures. The simulation results are shown in Fig. 3. For the brake mechanism with $r_b = 180$ *mm* ($b = 80$ *mm*, $h = 12$ *mm*), as α increases, the temperature of the friction surface decreases quickly enough according to a law close to linear.

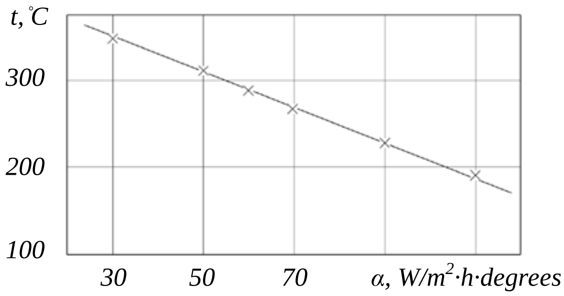

Fig. 3. The influence of α on the temperature of the brake drum at the end of the preliminary stage of test I

In particular, an increase of α from 30 to 70 W/(m²·h·degree) reduces the temperature at the end of the preliminary stage of test I from 352 to 268 ºC, that is, by 24 %. Therefore, the influence of the drum radius on the size of the cross-sectional area of the gap *S* and, consequently, on the value *α* cannot be neglected.

Studies show that as the gap increases, its influence on α decreases. Using these data, it is possible to obtain curves of the dependence of the temperature at the end of the preliminary stage of test I from the radius of the drum, taking into account the change in α due to the change in the gap between the drum and the rim. Analysis of the graphs shown in Fig. 2 showed that reducing the radius of the drum from 210 to 180 mm does not significantly increase the temperature at the end of the preliminary stage of the test; that is, the effect of reducing r_b is compensated to some extent by an increase in *α*.

The question of the impact of the radius of the drum on the change in temperature during the main stage of the test becomes essential. Based on the results, it is possible to conclude about the sufficiency of the energy capacity. Research conducted on the model of the rear brake mechanism of bus A-091 showed (Fig. 4) that the improvement of heat transfer due to the reduction of the radius of the drum reduces the temperature of the friction surface at the end of the preliminary stage and, as a result, at the end of the main stage of tests.

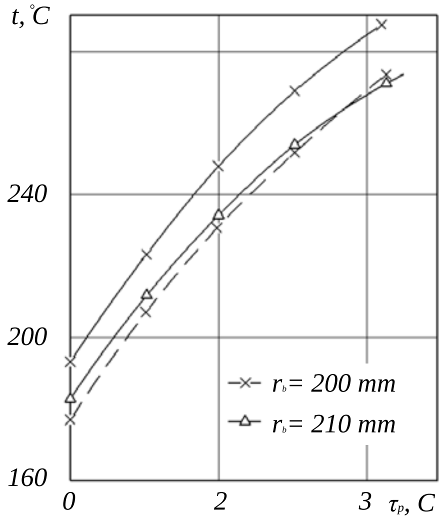

Fig. 4. Temperatures of the friction surfaces of the rear brake mechanism of bus A-091 during the main stage of test I: I – without considering the change in α; II – considering α = f (S).

The difference between the temperatures of the friction surface at the end of the main stage of the test with drum radii of 200 and 210 mm is insignificant (270 and 274 ºC). Thus, the reduction in the mass and cooling surface because of reducing the radius of the drum to 200 mm is primarily compensated by the increase in the heat transfer coefficient.

Recently, there has been a tendency in the automotive industry to reduce the drum's radius and increase the friction belt's width. Therefore, it is advisable to consider the equivalence of the change in the drum's radius and the friction belt's width, choosing the invariance of the temperature of the friction surface under the selected test mode as the criterion of equivalence.

It is known that when changing the radius of the drum $\Delta r_{b.n}$ or the width of the friction belt, the mass of the drum, the cooling surface, and the relative heat flow are changed. Elementary considerations allow us to obtain the following formulas for determining the equivalent width of the friction belt $b_e = b_n + \Delta b$ in the condition that the heat flows $(\Delta b)_q$, mass $(\Delta b)_m$ and cooling surfaces $(\Delta b)_F$ are equal.

$$(\Delta b)_q = \frac{b_n}{r_{b.n} - \Delta r_b}\Delta r_b, \tag{2}$$

$$(\Delta b)_F = \frac{b_n}{r_{b.n} - \Delta r_b + h}\Delta r_b, \tag{3}$$

$$(\Delta b)_m = \frac{b_n}{2r_{b.n} - 2\Delta r_b + h}\Delta r_b, \tag{4}$$

where $r_{b.n.}$, b_n – nominal (initial) values of the radius of the drum and the width of the friction belt; $(\Delta b)_q$, $(\Delta b)_F$,$(\Delta b)_m$ – equivalent changes in the width of the friction belt, provided that the heat flow, cooling surface, and mass respectively are constant; h – drum wall thickness.

It follows from formulas (2)–(4) that with an equivalent change in the drum's radius and the friction belt's width, there is no complete correspondence between the equality of heat flows and the equality of the cooling surface and mass. This can be judged by the coefficients:

$$K_1 = \frac{(\Delta b)_F}{(\Delta b)_q}, \tag{5}$$

$$K_2 = \frac{(\Delta b)_m}{(\Delta b)_q}. \tag{6}$$

Taking into account the above dependencies, received:

$$K_1 = \frac{1}{1 + \frac{h}{r_{b.n} - \Delta r_b}}, \tag{7}$$

$$K_2 = \frac{1}{K + \frac{h}{2(r_{b.n} - \Delta r_b)}}. \tag{8}$$

The graphs shown in Fig. 5 show that the values of these coefficients depend on the values of $r_{b.n.}$ and Δr_b. Coefficient K_1 differs relatively little from 0.95, and the coefficient K_2 from 0.92. Since the coefficients K_1 and K_2 are relatively close to one, the following variation in the values of r_b and b can be accepted as an initial premise for modeling, at which heat flows will remain unchanged. The radius of the drum can be varied within 180–220 mm, and the width of the friction belt within 80–160 mm (h=12 mm = const).

The temperature of the friction surface is significantly affected by the side wall. Therefore, it is advisable to consider the issue of the stability of the temperature mode of the drum separately when reducing r_b and the equivalent increasing b without taking into account the effect of the side wall and then taking it into account. The heat generated during each acceleration-braking cycle in the preliminary stage of test I and during emergency braking (the main stage) depends on the car's mass and the speed at the beginning and end of braking. If we assume that G_a = 8000 kH, β_d= 0.46, V_p= 60 km/h, V_{to}= 30 km/h, then the amount of generated heat for one cycle of the preliminary stage will be equal to 180 kJ, and during the main stage – to 240 kJ.

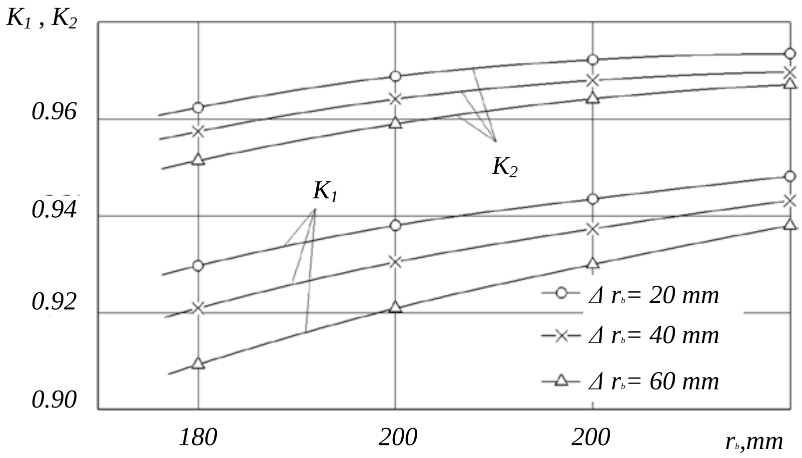

Fig. 5. Dependencies of coefficients K_1 and K_2 from r_b

An increase or decrease in G_a leads to a corresponding change in Q_c. Therefore, three heat values per cycle were chosen for simulation: 100, 200 and 400 kJ. The design and mode factors taken for modeling are given in Table. 1.

Table 1

Design and mode factors of brakes for modeling

Drum radius r_b, mm	Nominal width of friction belt b_n, mm	The equivalent width of the friction belt b_e, mm	Relative heat flow q, W/(m²·h)	Equivalent relative heat flow q_e, W/(m²·h)
220	80 160	– –	43.74 21.74	43.74 21.87
200	80 160	88 176	48,20 24.10	43.74 21.77
180	80 100	97.8 195.6	53.47 25.70	43.74 21.87

Fig. 6 shows the curves of the temperature dependence of the friction surface of the drum from Δr_b without a side wall. Analysis of these curves allows us to state that an increase in Δr_b leads to an increase in the temperature of the friction surface of the brake drum (curves I). If at the same time as the radius of the drum decreases, the width of the friction belt increases by the value calculated according to formulas (2)–(4), then the invariance of the temperature at the end of the preliminary stage of test I (curves II of quadrants a and d) is ensured. Under the same conditions, the temperatures at the end of the main stage of the test increase slightly. In particular, at Δr_b= 40 mm and b_n=160 mm, the temperature rises from 254 to 266 ºC, i.e. by about 5 %. It gives us the right to assume that if we do not take into account the role of the side wall as a heat-accumulating and heat-dissipating element of the brake drum, the fulfillment of dependence (2) will ensure an almost constant temperature mode of the brake mechanisms. The simulation results of test II also confirm it. When reducing r_b and equivalently increasing b, the temperature at the end of the preliminary stage remains unchanged. At the end of the main stage, it is almost unchanged (a slight increase in temperature with the increase in $\Delta r_b$is within the measurement accuracy).

Fig. 7 shows the research results conducted on models of real brake drums [10]. Reduction of r_b and an equivalent increase in b leads to some increase in temperature at the end of the preliminary step. This seemingly paradoxical phenomenon is explained by the fact that as the width of the friction belt increases, the amount of heat transferred to the side wall by heat conduction decreases.

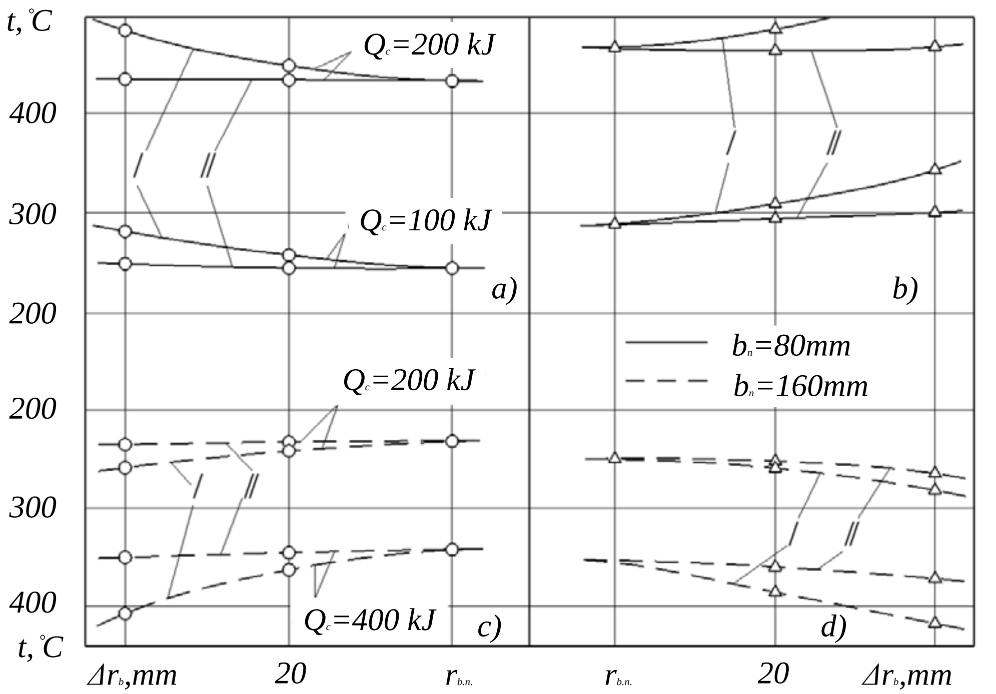

Fig. 6. Dependences of temperatures of the friction surfaces of brake drums without a side wall at the end of the preliminary (a and d) and main (b and c) stages of test I from Δr_b and b (α = 30W/(m²·h·degree, h=12mm): I – b = const; II – b =b_e=var.

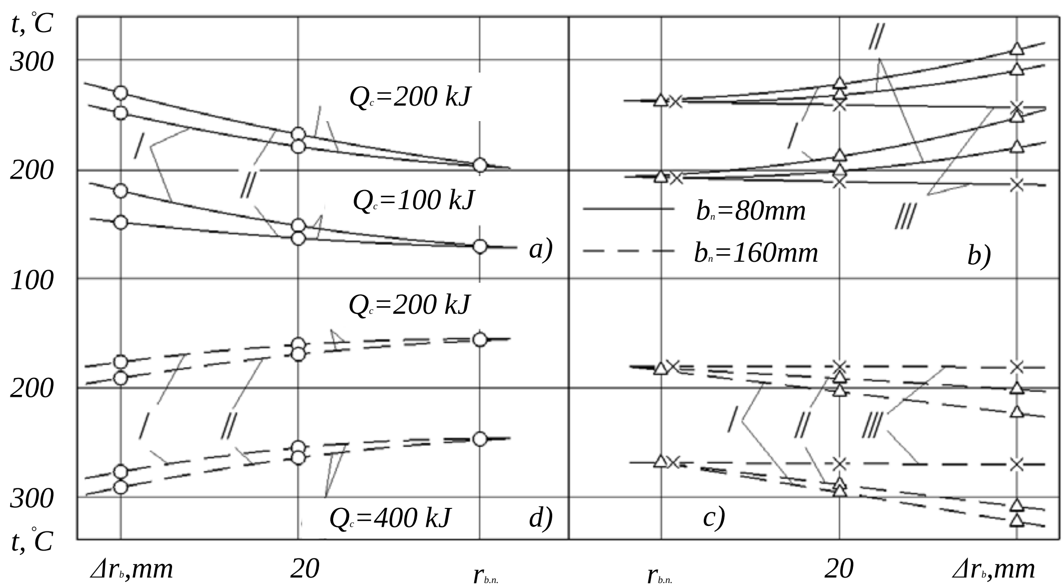

Fig. 7. Dependences of the temperatures of the friction surfaces of real brake drums at the end of the preliminary (a and d) and main (b and c) test stages from Δr_b and b (α = 30W/(m²·h·degree, h=12mm):1 – b_n=80 mm; 2 – b_n=160 mm; I – b = const; II – b =b_e=var; III – t_{start} = const

Since the primary condition for the main test stage is the temperature characteristic of the brake drum at the end of the preliminary stage, these conditions also affect the temperature at the end of the main stage (Fig. 7). In particular, at Q_c= 200 kJ reducing the drum radius by 40 mm and increasing the width of the friction belt by an equivalent value of 17.8 mm causes an increase in the temperature at the end of the main stage by 46ºC compared to the temperature at $r_{b.n}$ and b_n.

Varying the radius of the drum and the width of the friction belt following dependencies (2) – (3), as research has shown, ensures the practical invariance of the temperature of the friction surface at the end of the preliminary and the end of the main stages of test II.

Thus, for a real drum, according to formulas (2)–(3), it is impossible to solve the problem of an equivalent increase in the width of the friction belt with a decrease in the radius of the drum. The equivalence conditions are fulfilled for long-term braking (test II) and are not accurately fulfilled for the re-short-term mode carried out during test I.

The results of the experiments shown in Fig. 8 allow us to determine the necessary increase in b during the reduction of r_b, which ensures that temperatures remain practically unchanged at the end of the preliminary stage of test I. Suppose we imagine that the brake mechanism of a motor vehicle generates 200 kJ of heat in one cycle of the preliminary stage of test I. In that case, the temperature of the friction surface of the brake drum with b=80 mm, r_b=200 mm and h=12 mm will equal 240 ºC. When reducing the radius of the drum to 180 mm, it is necessary to accept the width of the friction belt of 105 mm to maintain the same temperature.

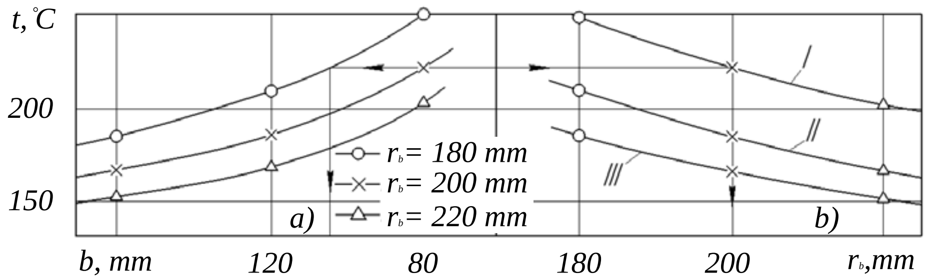

Fig. 8. Dependence of the temperatures of the friction surfaces of the brake drums at the end of the previous stage of tests I from b (a) and from r_b (b) (Q_c=200 kJ, α = 30 W/(m²·h·degree), h=12 mm): I – b = 80 mm; II – b = 120 mm; III – b = 160 mm.

An analysis of drum designs whose radius falls within the above limits shows that the wall thickness is within 12–20 mm. To assess the degree of influence of the drum wall thickness on its thermal state under various design parameters, we introduce an indicator that characterizes the temperature decrease per unit of thickness $\Delta t / \Delta h$. According to the results of research using the modeling method, it can be concluded that the accepted indicator of the change in the friction belt's width and the drum's radii do not significantly affect it. In particular, during tests I and II, the increase of wall thickness by 4 mm for drums with r_b=180, 200 and 220 mm leads to a decrease in the indicator $\Delta t / \Delta h$ at the end of the preliminary stage of the test by 17–22 % both at b = 80 mm and b = 160 mm.

Thus, the method of mathematical modeling allows us to study the thermal state of brake mechanisms, taking into account changes in structural and mode factors and conditions of heat exchange with the environment.

5. CONCLUSIONS

1. A finite-difference grid model of the drum brake was created in the “Fourier–2x,y,z” software environment, which allowed obtaining the flow of temperature curves in friction pairs during typical tests of the bus A-091 Bohdan.
2. An equivalent relationship between the drum's radius and its friction belt's width is obtained, provided their temperature mode does not change during typical vehicle tests. Reducing the radius of the drum by 40 mm and increasing the width of the friction belt by an equivalent

amount of 17.8 mm causes an increase in the temperature at the end of the main stage by 46ºC compared to the temperature at $r_{b.n}$ and b_n. (at Q_c= 200 kJ).

3. The difference between the temperatures of the friction surface at the end of the main stage of test I with drum radii of 200 and 210 mm is insignificant (270 and 274 ºC). Therefore, the decrease in the mass and cooling surface due to the reduction of the drum radius to 200 mm is compensated mainly by the increase in the heat transfer coefficient due to the rise in the gap between the outer surface of the drum and the wheel rim.
4. The influence of the side wall and the drum wall thickness on the temperature mode of the brake during typical vehicle tests was evaluated. An increase in wall thickness by 4 mm for drums with r_b=180, 200 and 220 mm leads to a decrease in the indicator $\Delta t / \Delta h$ at the end of the preliminary testing stage by 17–22 % both at b = 80 mm and at b = 160 mm.
5. The research results should be considered when designing brake mechanisms for the preliminary assessment of the influence of essential factors on the temperature state of drum brakes, which depend on the braking efficiency of vehicles.

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ВПЛИВ РЕЖИМНИХ ТА КОНСТРУКТИВНИХ ЧИННИКІВ НА ТЕМПЕРАТУРНИЙ СТАН АВТОМОБІЛЬНИХ БАРАБАННИХ ГАЛЬМ

Анотація. *Сучасний стан дорожнього покриття автомагістралей забезпечує можливість оперативного здійснення транспортних операцій. Такі обставини спричиняють рух автотранспортних засобів (АТЗ) із великими швидкостями, що стає неможливим без гальмівної системи, здатної забезпечити високу ефективність гальмування та оптимальний перебіг процесу із позицій стійкості та керованості.*

Однією із головних вимог до гальмівної системи сучасного АТЗ є стабільність вихідних параметрів, тобто параметрична надійність. Тому важливо мати дані стосовно режимів роботи та енергонавантаженості гальм. Тільки з урахуванням таких даних можна створити гальмівну систему, вихідна характеристика якої буде достатньо стабільною в умовах високої енергонавантаженості.

Тому не випадково міжнародна методика перевірки ефективності гальм АТЗ (Правило 13 ЄЕК ООН) передбачає випробування I, які характеризуються циклічними гальмуваннями (міські умови) та випробування II – тривалими гальмуваннями (гірські умови).

Найнестабільнішою ланкою гальмівної системи є гальмовий механізм, одним зі способів підвищення ефективності якого є забезпечення достатньої енергоємності, що лімітується температурою поверхні тертя.

Об'єктом дослідження є питання про еквівалентність зміни радіуса барабана й ширини пояса тертя гальма. Як критерій еквівалентності прийнято незмінність температури поверхні тертя за вибраного режиму випробувань. Показано, що на сіткових теплових моделях із залученням програмного комплексу "Фур'є – 2 x,y,z" можна також оцінити роль бічної стінки барабана на температурний режим гальма за різних режимів випробувань.

Показано вплив коефіцієнта тепловіддачі на температурний режим гальма через урахування зазору між стінкою барабана й ободом колеса. Отримані формули для визначення еквівалентної ширини пояса тертя за умови рівності теплових потоків, мас і поверхонь охолодження.

Ключові слова: *автотранспортний засіб, барабанний гальмовий механізм, конструктивні параметри барабана, режими випробувань типу I та II, теплова модель.*