

Виконано аналіз енергонавантажності самовентилюваного дисково-колодкового гальма вантажного транспортного засобу, яка суттєво впливає на експлуатаційні параметри його пар тертя. Поверхні температури пар тертя гальма перевищують допустиму для матеріалу полімерної накладки. Внаслідок цього відбувається вигорання його сполучних компонентів, зниження динамічного коефіцієнта тертя і гальмівного моменту. Це негативно впливає на ефективність гальмування транспортних засобів і безпеку руху.

Розглянуто інтенсивність відведення теплоти від поверхонь гальмівного диска з урахуванням розташування та якості механічної обробки. Встановлено, що інтенсивність перенесення теплоти від зовнішніх та внутрішніх матових і полірованих поверхонь самовентилюваного диска гальма конвекцією у багато разів є нижчою за інтенсивність процесів дисипації теплоти в тілі правого та лівого напівдисків при теплопровідності. Пояснюється це тим, що темп нагрівання в десятки разів перевищує темп вимушеного повітряного охолодження.

Встановлено закономірності зміни теплопровідності стінки напівдиска, оберненою величиною до якої є термічний опір товщини напівдиска. Розглянуто конструкцію та процеси теплообміну в самовентилюваному дисково-колодковому гальмі транспортного засобу. Досліджено інтенсивність теплопередачі самовентилюваного диска гальма транспортного засобу в залежності від швидкості його руху з урахуванням коефіцієнтів тепловіддачі від зовнішніх та внутрішніх поверхонь диска. Встановлено вплив площ поверхонь диска на інтенсифікацію процесу теплопередачі по товщині диска.

Отримані результати досліджень дозволяють конструктору ще на стадії проектування самовентилюваних гальмівних дисків цілеспрямовано варіювати їхніми конструктивними параметрами (максимальним радіусом диска, радіусами його пояса тертя, зведеною товщиною напівдисків і диска в цілому). За величиною енергонавантажності стає можливим спрогнозувати напружено-деформований стан його пояса тертя і знос фрикційних накладок

**Ключові слова:** дисково-колодкове гальмо, самовентилюваний гальмовий диск, пояс тертя, пара тертя, поверхні теплообміну

# FEATURES OF THE ESTIMATION OF THE INTENSITY OF HEAT EXCHANGE IN SELF-VENTILATED DISK-SHOE BRAKES OF VEHICLES

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## 1. Introduction

According to experts' forecasts in the near future, disk-shoe brakes will remain the most common type of braking devices for subcategories of vehicles. At the same time, their design should be improved constantly based on computational methods of self-ventilated disks functional parameters. Brakes must meet the ever-growing requirements for increasing friction power caused by increasing driving speeds and axle loads due to an increased carrying capacity of vehicles. The priority objective for most of foreign companies (Mercedes, Volkswagen, Citroen, etc.) is the creation of reliable and efficient friction brake pairs with high energy

intensity. The solution, according to their view, is to examine new structural materials for the manufacture of friction pairs of brakes and the optimization of their design parameters. In the present time, the forced cooling of self-ventilated brake disks, which are used in the vehicle's categories, is not effective. This is due to the irrational ratio of matte and polished disk's surface areas. Intensity of heat transfer in disk-shoe brakes with different disk designs considering their thermal resistance must be studied. Evaluation of forced air cooling efficiency of self-ventilated brake disks during the movement is relevant. Solution to this problem will allow defining the rational ways of brake disk design development to increase their energy capacity.

## 2. Literature review and problem statement

It is known that the law of friction requires that the total area ( $\Sigma A$ ) of a contact increases in proportion to the specific nominal contact load. The results of experimental studies, which define the thermal resistances of friction pairs' microprotrusions' contact spots prove that  $\Sigma A$  must increase in proportion to the specific load. Hence, the average contact size  $a$  should remain constant, and the number of spots  $n$  of the actual contact should increase in proportion to the load.

Disk-shoe brakes of vehicle subcategories under UN-ECE Rule 13 (United Nations Economic Commission for Europe) [1] are affected by cyclic and long braking modes. At the same time, on the surfaces of brakes' friction pairs, developing temperatures exceed the permissible values for polymer friction element materials, i.e. 300...350 °C. This leads to a negative influence on the performance parameters of brakes' friction pairs, significantly reducing effectiveness, since the braking disk is a battery of thermal energy and affected by a stress-strain state [2]. This indicates the need to reduce energy load. The heat load of buses disk brakes is studied in [3]. However, the ways of heat load reduction are not mentioned. Furthermore, when calculating the thermal balance of friction pairs, it is not considered that the thermal emission coefficient is a variable, which depends on the vehicle's speed. There are more accurate thermal calculations made by the authors [4, 5]. But the advised analytical dependencies are extremely bulky. Reasonable mathematical expressions in engineering calculations were acquired by the authors [6, 7]. However, the heat emission intensity is not considered, which significantly reduces the accuracy, particularly at high vehicle speeds. Temperature distribution is shown and train disk brakes heat streams are analyzed [8]. But the reasons of the origins and development of thermal cracks on the disk friction surface are not reviewed. Dynamic deformation modes of metal friction elements are studied by the authors [9, 10], as well as environmental influence on the distribution intensity of cracks on their working surfaces. Yet the thermal stresses are not considered along with temperature influence on the friction pair durability. Moreover, the ratios of the design parameters of solid and self-ventilated brake disks, which are aimed at the intensification of forced air cooling of the brake friction pairs, were not determined. As a result, the energy load is reduced, which, in turn, depends on the brake disk diameter and thickness, as well as metal capacity.

Microcracks occur due to high temperature stresses on the disks' friction belt, accelerating its failure. Temperature stresses can occur due to the anisotropy of the material properties, as well as differences in thermophysical and mechanical characteristics of the structural components of the part (coefficient of linear expansion, thermal conductivity, elastic modulus).

The frictional interaction of microprotrusions of the "metal – polymer" friction pair has a pulsed nature. Wherein an ohmic, blocking and neutral contacts are formed on their spots [11, 12]. The contacts are characterized by different energy levels. The main physical parameters of the metal friction element are the electrothermal resistance and the thermal conductivity of its material, which are included in the analytical relation to determine the heat transfer coefficient. It is the opposite of thermal resistance. The magnitude of the thermal conductivity coefficient of a metal friction

element material affected by thermal stress influences its stress-strain state and the probability of microcracks.

## 3. The aim and objectives of the study

The aim of the study is to estimate the intensity of heat transfer in the self-ventilated disk-shoe brake of the vehicle. This will allow reducing the energy load of brake friction pairs and, as a result, improving the wear and friction properties.

To achieve the proposed aim, the following objectives were stated:

- determination of thermal resistance and energy level of friction pairs' microprotrusions contact spots;
- examination of the design features and heat transfer processes in the self-ventilated disk-shoe brake of the vehicle;
- determination of the heat transfer efficiency of a vehicle self-ventilated brake disk.

## 4. Thermal resistance of contact spots in microprotrusions of friction pairs and their energy level

In friction pairs of brake devices, the heat flow is transmitted through adjacent surfaces during friction interaction of microprotrusions of rubbing elements.

After the next braking at a certain temperature  $T_2$ , the metal friction element will come to a contact with the surface of the polymer patch with the temperature  $T_1$  (Fig. 1). The diagram is illustrated in the coordinate axis  $\Delta T_k - \rho_k$  with  $\rho_k = 0$ , the temperature difference  $\Delta T_k$  is required to pass through the contact zone of a given heat flux  $q$ . With a further decrease in temperature  $T_2$  and, consequently, an increase in  $\Delta T_k$ , the contact pressure  $\rho_k$  will increase with respect to the line  $A$ , and the heat flux will increase between the microprotrusions of friction pairs. The thermal resistance of the contact  $R_k$  considering an increase in  $\rho_k$  and, consequently, the temperature difference  $\Delta T_k$  required to pass through the contact spot of the given heat flux  $q$  with respect to the curve  $B$  will decrease. The preset heat flux passes through the contact spot (point 3 of the intersection lines  $A$  and  $B$ ) with a certain  $\Delta T_k$ . A further increase in the specific load  $\rho_k$  contributes to the heat flux stabilization with a minimum thermal resistance  $R_k$  of the contact.

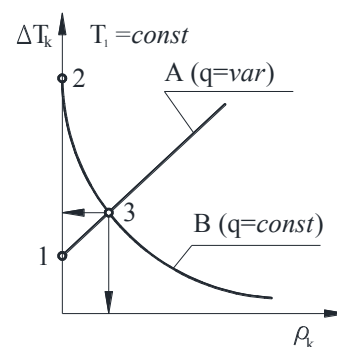
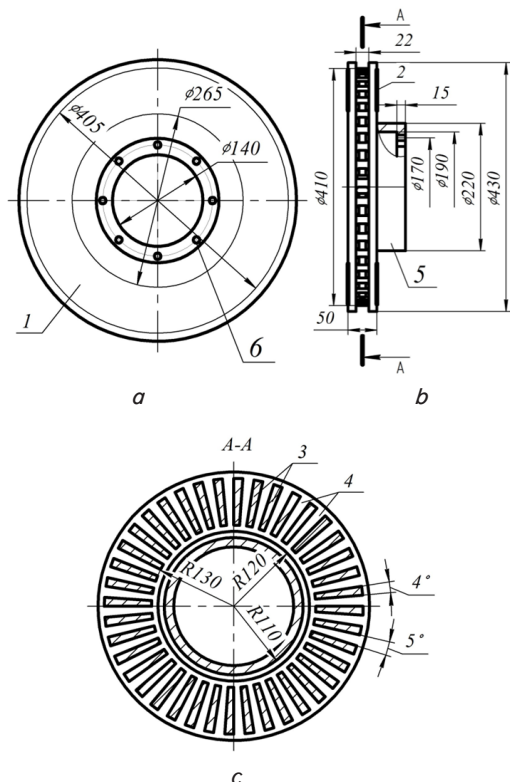


Fig. 1. Heat contact parameter of friction interaction of microprotrusions in friction pairs

The combination of graphical dependences  $A$  and  $B$  is called the heat contact characteristic of microprotrusions in friction pairs.

**5. Design and heat transfer processes of self-ventilated disk-shoe brake in the vehicle**

The disk-shoe brake of a MAN cargo vehicle was chosen as the object of research. A self-ventilated brake disk is used in the design of these brakes (Fig. 2, *a, b*). Air is gathered through the holes in the hub or the base of the central part of the brake disk. The air is absorbed inside the ventilation ducts, then it passes through them and is extruded, cooling the internal cavities of the disk. The speed of the circumambient air flow successfully increased from 5.0 to 10.0 % at different points of the disk, by selecting a rational section of the channels and the curvature radii of the inlet walls.



**Fig. 2** Main (*a*) and side (*b*) views of self-ventilated braking disk with flange, longitudinal section (*c*): 1 – side surfaces of the disk; 2 – friction belts; 3 – radial ribs; 4 – ventilated channels; 5 – disk flange; 6 – fixing holes

The convection heat emission coefficient of a self-ventilating disk is approximately twice as high as of the solid one. However, the cooling capacity of a self-ventilated disk decreases at high vehicle speeds due to an increase in the static pressure of the washing air. The pumping action of the rotor (ribs forming diffusers or confusers between side surfaces) is weakened due to the circumambient air urge to leave the rotor in the front part of the disk's semicircle. In this case, there are positive and negative pressure gradients of the circumambient air along the length of the confusers.

One of the methods for intensifying heat transfer through self-ventilating braking disk sides is to increase the area of one of the surface side by installing ribs [13]. Disk sides are separated by air environments with different temperatures.  $t_1$  and  $t_2$  are considered as the temperatures of the air, which surrounds the outer side walls of the half-disks,  $t_3$  is the temperature of the air, which surrounds the side surfaces of the ventilation channels and inner side walls of the half-disks.

The left half-disk is made with a flange, and the right one is made with ribbing, forming the ventilation ducts in the form of trapezoidal confusers. Metal intensity and areas of mat heat transfer surfaces are different in the left and right half-disks. The end surface of the left half-disk flange conductively (due to thermal conductivity) interacts with the hub of one of the vehicle drivetrains. The efficiency of the above method is high, if the intensity of the forced heat emission of the ribbed inner surface of the right half-disk is significantly higher than that of the outside one. Moreover, the thermal resistance of the right half-disk with ribs is much larger than that of the left half-disk with a flange. Due to the significant difference in thermal resistances of the left and right half-disks of a self-ventilated disk, it is necessary to consider the left and right half-disks separately when evaluating its stress-strain state.

The heat flow, which permeates the disk body, is:

$$q_1 = k_1(t_1 - t_2) \frac{W}{m^2}, \tag{1}$$

where  $k_1$  – heat transfer coefficient;  $t_1$  and  $t_2$  – surface temperatures of disk friction belts.

The main design and weight disk parameters of vehicle categories are defined (Table 1), since their body is a heat energy battery. Analysis of acquired data (Table 1) showed that thermal resistance of disk thickness is a value opposite to the heat transfer coefficient.

**Table 1**

**Defining of design and weight parameters of brake disks of vehicle's categories**

Parameter	Calculated relations	$R^2$	$\Delta$ , %
Ratio of the solid flange disk radii	$R_m/R_t = R_l/R_m$ ; (2)	–	–
Ratio of cooling and heating disk areas:			
Solid	$A_c/A_h = C_m/C_p$ ; (3)	–	–
Self-ventilated [4]	$\sqrt[3]{A_c/A_h} = C_m/C_p$ ; (4)	–	–
Diameters and weights: solid and self-ventilated disks of passenger and cargo vehicles	$D = 0.032\delta^2 + 1.8714\delta + 233.05$ ; (5)	0.82	5.0
	$m = 0.0174\delta^2 - 0.3309\delta + 5.8772$ ; (6)	0.96	13.0
Solid disks for vehicle categories	$D = -0.264\delta^4 + 11.602\delta^3 - 189.23\delta^2 + 1362.5\delta - 3400.3$ ; (7)	0.91	1.8
	$m = -0.0067\delta^5 + 0.3621\delta^4 - 7.6649\delta^3 + 79.872\delta^2 - 408.81\delta + 823.82$ ; (8)	0.93	14.1
self-ventilated disks for vehicle categories	$D = 4.3741\delta - 193.51$ ; (9)	0.91	3.0
	$m = 0.0181\delta^2 - 0.3624\delta + 6.0937$ ; (10)	0.97	9.6

In Table 1 dependencies, the following designations are used: radius: from the edge of the mounting hole to the end  $R_t = R_l + R_m$ ; top and bottom point of the pad ( $R_p$ );  $A_c$  and  $A_h$  – disk areas: cooling and heating,  $C_m$  and  $C_p$  – emissivity coefficients of matt and polished disk surfaces;  $D$ ,  $\delta$  – diameter and thickness of the disk and half-disk;  $m$  – disk mass;  $R^2$  and  $\Delta$  – proximity value and average deviation for the disk diameter and mass.

Graphical dependencies were created (Fig. 3, 4) for the manufactured disks of different vehicle's categories based on system synthesis and analysis considering their weight.

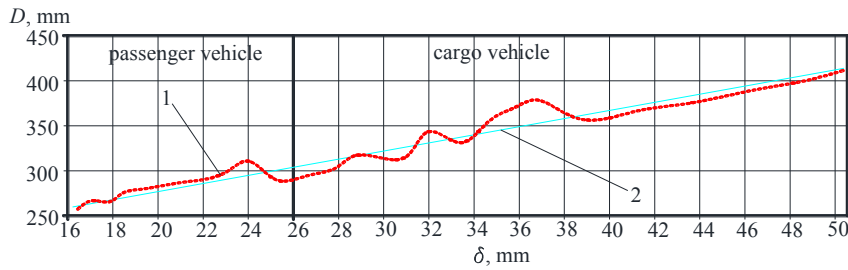


Fig. 3. Diameter variation patterns of self-ventilated (b) brake disks depending on their thickness for vehicle's categories with a mass: passenger's - 1.485-2.850 t; cargo - 2.9-26.0 t: 1 and 2 - curves: unsystematic and polynomial

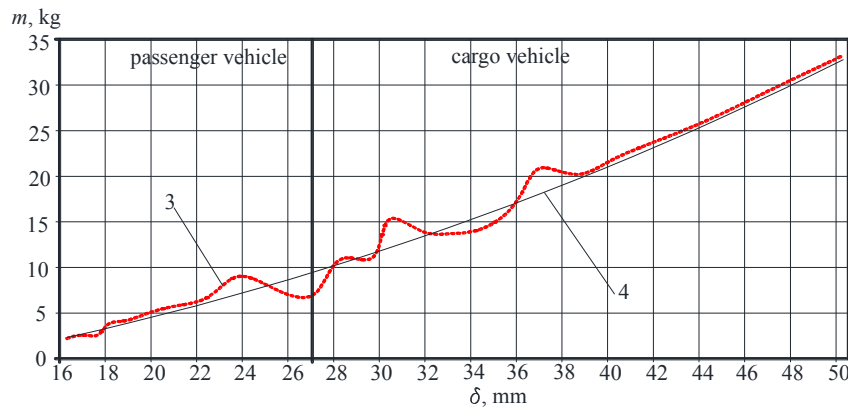


Fig. 4. Mass variation patterns of self-ventilated brake disks depending on their thickness for vehicle's categories: 3 and 4 - curves: unsystematic and polynomial

Analysis of acquired data from the dependences (2)–(10) allow stating the following:

- dependency ratios (2) give an ability to move away the middle friction belt line to its end, and the friction belt bottom radius - to the vertical component of the flange; the first solution will increase the breaking momentum of the brake friction pairs, and the second one - will decrease the surface temperature gradients of the disk, which is confirmed by the disk friction belt lower circle offset to the horizontal component of the flange;

- percent deviation between acquired values in dependences (3) and (4) reached, respectively, 5-8 % and 10-13 % for the solid and self-ventilated brake disks.

After the disk diameter and mass calculation, evaluation of the main operational parameters (unit loads  $p$  or dynamic friction coefficient  $f$ ) at the sliding velocity  $V_{sl} = \text{const}$  depending on the heat flow, acting on the disk friction belts during the braking process of a vehicle, is performed.

According to the Fourier law, the heat flow is expressed by the dependence:

$$q = \frac{\lambda}{\delta} \nabla T, \tag{11}$$

where  $\lambda$  - coefficient of the thermal conductivity;  $\delta$  - disk thickness;  $\nabla T$  - temperature gradient operator.

The heat flow increment after the next cyclic braking of the vehicle can be written as:

$$\delta W = -\frac{\lambda}{\delta} \delta'(\nabla T) - (\delta' \frac{\lambda}{\delta}) \nabla T, \tag{12}$$

where  $\delta'$  - variation symbol; in a specific event, when  $\lambda = \text{const}$ , the relation (1) is acquired.

Thermal resistance over the thickness of a solid disk and self-ventilating half-disks (one half-disk has a flange, and the second one has ventilation channels formed by ribs) plays an essential role in the processes of conductive heat transfer [14]. It is expressed by the ratio  $\delta/\lambda$ , and has a unit of measurement  $(\text{m}^2 \cdot \text{C})/\text{W}$ , i. e. is the inverse of the heat transfer coefficient through the thickness of the studied disk (Fig. 5, a, b).

According to the presented graphical dependences, as the disk thickness rises, it is necessary to vary the thermal conductivity of the materials used to manufacture brake disks. Solving the equation  $q = \frac{\lambda}{\delta} (t - t_0) \cdot 10^{-2}$  (where  $t, t_0$ , temperatures: volume disk and environmental), the graphical dependency is built  $q = f(\frac{\lambda}{\delta}, t)$  (Fig. 4). In the last formula, the value  $\lambda/\delta$  appears, which is the opposite of the brake disk thickness thermal resistance.

During the design process of different disk types of vehicle categories, calculations must be done not only to define the design and weight parameters, but also the energy capacity of the friction belts.

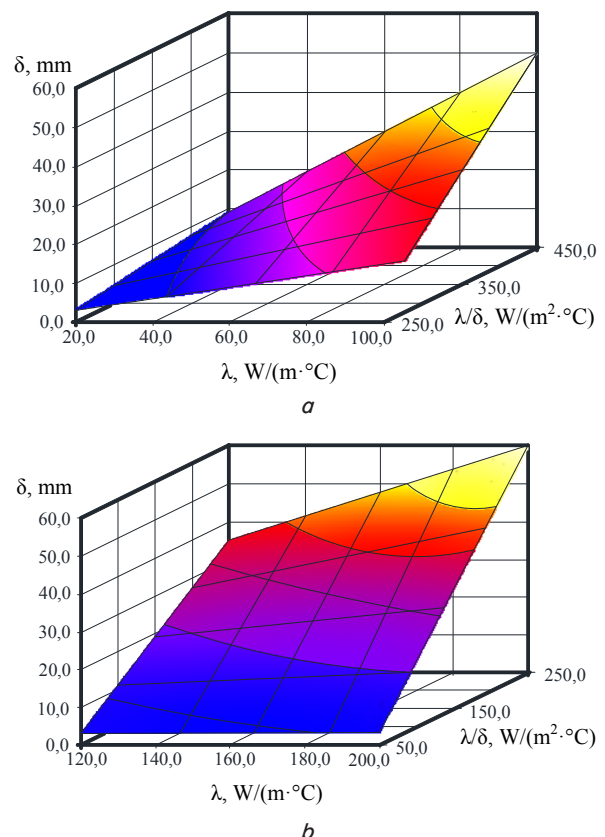


Fig. 5. Patterns of change in thermal resistance ( $\delta/\lambda$ ) of brake disks depending on their thickness ( $\delta$ ) and materials' heat transfer coefficient ( $\lambda$ ), changing within: a - (20...100) W/(m·C), b - (120...200) W/(m·C)

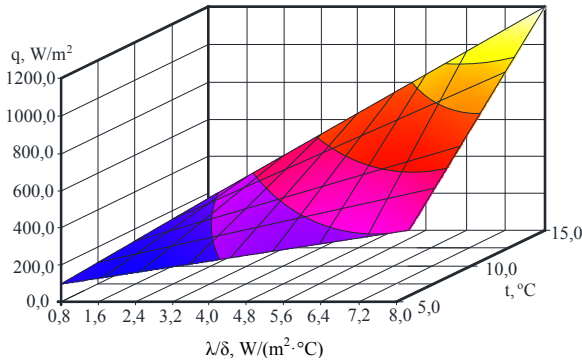


Fig. 6. Regularities of changes in the heat flow  $q$ , penetrating the braking disk friction belt during the frictional interaction of the friction pair “shoe-disk” of the brake, depending on the parameter  $\lambda/\delta$  and the volume temperature in the disk body

**6. Heat transfer of the self-ventilated vehicle brake disk**

The flat wall with a friction belt of a self-ventilated left half-disk is considered as an example. The coefficient is equal to:

$$k = \frac{1}{\alpha_1} + \frac{\lambda}{\delta} + \frac{1}{\alpha_2}, \tag{13}$$

where  $\alpha_1, \alpha_2$  – heat emission coefficients of operating and non-operating half-disk surfaces;  $\delta$  – half-disk thickness;  $\lambda$  – half-disk materials’ heat transfer coefficient.

Formula (13) shows that the increase of the coefficient  $k$  can be reached by rising any of the three thermal resistances  $1/\alpha_1, \delta/\lambda, 1/\alpha_2$  [9].

The internal effect of thermal resistance  $\delta/\lambda$  on the value of the heat transfer coefficient is examined. Since the choice of the material, which is used to make the half-disk, is determined by the technical conditions for this product, hence the thermal conductivity  $\lambda$  is a fixed value. Thus, the time reduction of the heat flow transit (at  $\alpha_1 = \text{const}, \alpha_2 = \text{const}$ ) is possible only by reducing the side thickness  $\delta$ . However, the possibilities of reducing  $\delta$  are extremely limited, since they are related to its strength characteristics. Greater effect can be achieved by changing the heat emission coefficients. Assuming  $\delta/\lambda \rightarrow 0$ , we simplify the formula (13) to the form:

$$k = \frac{1}{\alpha_1} + \frac{1}{\alpha_2}. \tag{14}$$

The last dependency can be rearranged to the next form:

$$k = \frac{\alpha_1}{1 + \frac{\alpha_1}{\alpha_2}} = \frac{\alpha_2}{\frac{\alpha_2}{\alpha_1} + 1}. \tag{15}$$

Equations (14) and (15) imply that with  $\alpha_2 \rightarrow \infty, k \rightarrow \alpha_1$  and vice versa, with  $\alpha_1 \rightarrow \infty, k \rightarrow \alpha_2$ . Thus, a larger critical value of the heat transfer coefficient cannot exceed the value of the smallest heat emission coefficients. Dependency (15) allows drawing another important conclusion. Consider  $\alpha_2/\alpha_1 = k_a \gg 1$ . Then

$$k = \frac{\alpha_1}{1 + \frac{\alpha_1}{\alpha_2}} = \frac{\alpha_1}{1 + \frac{1}{k_a}}$$

$\alpha_2$  is increased  $k_{a2}$  times:  $\alpha'_2 = \alpha_2 k_{a2}, k_{a2} > 1$ .

Then the new value of the heat transfer coefficient equals:

$$k = \frac{\alpha_1}{1 + \frac{\alpha_1}{\alpha'_2}} = \frac{\alpha_1}{1 + \frac{1}{k_a k_{a2}}}$$

Consider the correlation of the form:

$$\frac{k'}{k} = \frac{1 + \frac{1}{k_a}}{1 + \frac{1}{k_a k_{a2}}} = 1 + \frac{k_{a2} - 1}{k_a k_{a2} + 1} = 1 + \epsilon.$$

where  $\epsilon = (k_{a2} - 1)/(k_a k_{a2} + 1)$ .

It is obvious that the value of  $\epsilon$  will be rather small. It is shown on the example. Consider  $k_a = 100, \alpha_2$  is doubled, then  $k_{a2} = 2$ , and  $\epsilon \approx 0.5\%$ .

Hence, an increase of the larger heat transfer coefficient ( $\alpha_2$ ) two times causes the increase in the heat transfer coefficient by only 0.5%. Hereof, having that  $\alpha_2 \gg \alpha_1$ , further increase of  $\alpha_2$ , affected by considerable difficulties (increased purity of the inner surface of the half-disk, intensified heat emission due to changing the mode of circumambient air movement in the disk confusers), is impractical. From a practical point of view, it is important to determine (with a sufficient degree of accuracy) the threshold value of  $\alpha_2$ , which is the starting point of its senseless further increase.

In Fig. 7, a, b, graphical dependences  $k = f(\alpha_1, \alpha_2)$  are shown, described by the formula (15).

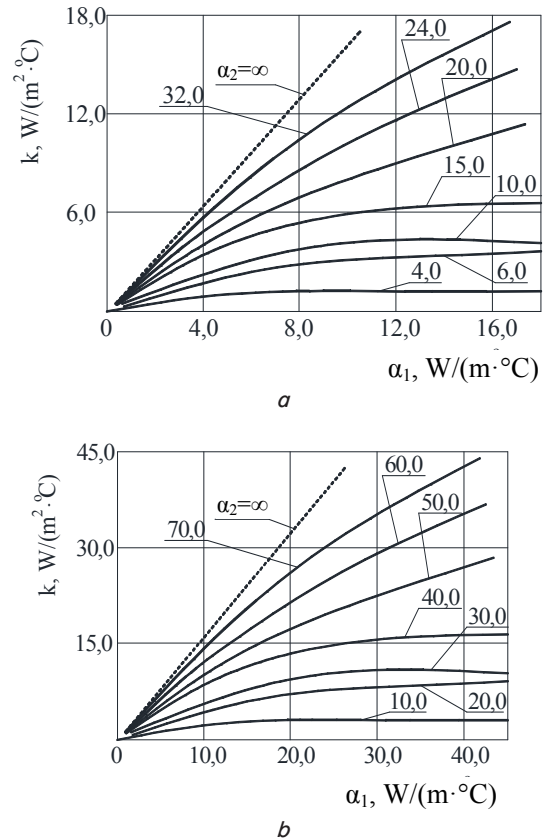


Fig. 7. Patterns of change of heat transfer coefficients  $k$  depending on heat emission coefficients from the outer  $\alpha_1$  and inner  $\alpha_2$  surfaces of the left half-disk of a self-ventilated brake disk of a TGA 26.430 MAN cargo vehicle when driving at speeds: a – 30 km/h; b – 60 km/h

According to the graph,  $k$  increases rapidly with  $\alpha_1$  until  $\alpha_1$  and  $\alpha_2$  become approximately equal. With a further increase in  $\alpha_1$ , the growth of  $k$  slows down and then nearly stops. Thus, with  $\alpha_1 \ll \alpha_2$ , it is necessary to increase  $\alpha_1$  for an increase in  $k$ , which is equivalent to a decrease in thermal resistances  $1/\alpha_1$ . After reaching the equality of  $\alpha_1 \approx \alpha_2$  for heat transfer intensification, you can increase any of the heat emission coefficients.

The second way of heat transfer intensification is to increase the area of internal heat exchange surfaces of self-ventilated brake disks.

If the heat transfer surface area of the disk is increased by ribbing, hence, the heat flow transferred increases. Thermal resistances of the ribbed wall heat emission are proportional to the values  $\frac{1}{\alpha_1 A_1}$  and  $\frac{1}{\alpha_2 A_2}$ . The ribs of the right half-disk inner surface of a self-ventilated brake disk are one of the major ways to intensify heat transfer. It is worth noting that the ribs of different geometry and thermal conductivity work differently even under the same conditions with homogeneous sources and heat sinks. The major simplifying assumptions are formulated in the monograph [6], which allows the use of thermal calculations of the rib edges with different geometry.

The criterion, which is commonly used to accept the ribbing of surfaces, is to achieve approximate equality of both heat emission thermal resistances, whereas  $\alpha_1 A_1 \approx \alpha_2 A_2$ . Thus, if  $\alpha_2 \ll \alpha_1$ , then the inner surface should be ribbed until the equality  $A_1/A_2 = \alpha_2/\alpha_1$  is reached. A further increase in  $A_2$  is ineffective, since with a slight increase in heat flow, the mass of the structure increases significantly, as well as the cost of manufacture, and so on.

The rib edge geometry (shape and size) can be quite varied. The calculation of temperature fields in the rib edges is rather complicated, despite some simplifying assumptions. These practically important problems of heat transfer are examined in detail in [6].

The mass of self-ventilated brake disks of foreign-made vehicles subcategories ranges from 25.6 to 37.0 kg. The thickness of the half-disks must be calculated based on their strength characteristics. When determining the area of the rib edges of the second half-disk, increase in the mass of the self-ventilating brake disk as a whole should be avoided. According to this fact, inertia momentum of rotational brake disks will increase, which will lead to the braking power increase. Furthermore, this contributes to an increase of vehicles suspended masses, which negatively influences the driving dynamics.

## 7. Discussion of heat exchange intensity research results in self-ventilated brake disks

Research results of the self-ventilated brake disk of a cargo vehicle allow stating the following.

During the design process of self-ventilated brake disks to increase the air washing effectiveness of its inner surfaces, it is important that the total cross-sectional area relation of ventilation ducts to the brought cross-sectional ventilation channel on the level of the disk friction belt average radius is equal to 0.6...0.65.

The inner surface of the left half-disk must be polished to intensify heat exchange. Wherein the relation value  $\sqrt[3]{A_m/A_p}$  is increased. The value should approach  $C_m/C_p$ , which equals 3.748/1.134.

Calculation method analysis of the design and weight parameters of different brake disk types has shown that the prox-

imity value and average deviation for the diameter and mass of the disk vary, respectively, from 0.82 to 0.97 and from 1.8 to 14.1 %. The accuracy is satisfactory for such kind of calculation.

Patterns of change of heat transfer coefficients  $k$  from 6.0 to 45.0 W/(m<sup>2</sup>s) depending on heat emission coefficients from the outer and inner surfaces of the left half-disk (from 4.0 to 40.0 W/(m<sup>2</sup>s)) at average minimum (30 km/h) and maximum (60 km/h) driving speed of the cargo vehicle with a maximum cargo load of 26.0 t are revealed. Applying of acquired data while the strain-stress state of half-disks is assessed will increase the accuracy of thermal stress calculation by 15 %.

Knowledge of the thermal resistance of different brake disk types, which is considered to be the opposite value of the heat transfer coefficient with determined heat flows (on the surface of the friction belt and the one permeating its body), as well as one of the heat emission coefficients from the disk surface, allows defining the second heat emission coefficient.

To increase the effectiveness of disk inner surface air washing, the relation of the total cross-sectional area of ventilated ducts and brought cross-sectional channel area with the level of the disk friction belt average radius is equal to 0.6...0.65.

The inner surface of the left half-disk must be polished to intensify heat exchange. Wherein the value of the relation is increased  $\sqrt[3]{A_m/A_p}$  ( $A_m$ ,  $A_p$  – matt and polished disk surface areas). The value must approach  $C_m/C_p$ , ( $C_m$ ,  $C_p$  – beam emission coefficients of matt and polished surfaces, 3.748/1.134=3.3).

While assessing the strain-stress state of a self-ventilated disk with cylinders, which join the half-disks, they must be considered as a whole with friction surface microprotrusions contact spots.

The drawback of the heat exchange determination method is the fact that it uses the volume temperatures instead of the flash temperatures. This slightly reduces the accuracy.

## 8. Conclusions

1. It is proven that one of the ways to increase forced air cooling intensity of the disk-pad friction pairs during the vehicle movement is to increase the rib area of the right half-disk. Whereas if the rib area is increased by approximately 20 %, the cooling intensity is increased by 7...10 %. Further increase of the rib area is not advisable, since the metal capacity is highly increased.

2. Graphical dependencies are acquired, which allow determining the disk material, according to the set heat flows with known heat transfer and thermal resistance coefficients of the chosen material. The results will allow decreasing the metal capacity of the brake disk to 10 %.

3. Lately, the disks with cylindrical spikes started to outstand the self-ventilated disks with confusers because of the complicated design, metal capacity, high energy intensity and low effectiveness of forced surface air cooling. In the contact problem, while considering the strain-stress state of a self-ventilated disk with cylindrical spikes, they must be represented as a whole with friction surfaces microprotrusions contact spots. Furthermore, disks with cylindrical spikes are a good object for their structural development.

4. Reduction of the metal capacity of self-ventilated brake disks with confusers is achieved by performing the rib system made of plastic, which is insulated from the inner surface of the half-disk.

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