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Theoretical and Experimental Study of Cooling of Brake Friction Pairs

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Abstract

Theoretical and experimental studies of the dynamics of convective heat transfer in the friction pairs of moving vehicle brakes washed by air flows have made it possible to determine: the choice of the method of thermal similarity theory with its theorems, boundary conditions, dimensionless criteria and, in this case, the obtained criteria equation for assessing convective heat transfer in friction pairs of a vehicle drum-shoe brake; the heat load of cyclic and long-term braking modes with their features of vehicles according to UNECE Regulation No.13 is affected by the formation of dynamic and thermal layers of varying thickness on the matte and polished surfaces of the drum rim, which affects the intensity of their heat exchange; with increasing time of action of pulsed electric and thermal currents, the thickness of the electrothermal layers also increases; due to the lower thermal conductivity coefficient of the lining materials than the drum rim material, the thickness of the thermal layer in the near-surface layer in the lining is on average 26% less; when comparing electrical and thermal layers, it is clear that the thickness of the thermal layer is an order of magnitude higher than the thickness of the electrical layer, which has been confirmed by numerous studies.

Keywords: disc-shoe brake, friction pair, thermal model, surface and subsurface layers.

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Əyləclərin sürtünmə cütünün soyudulmasının nəzəri və eksperimental tədqiqi

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Xülasə. Məqalədə istilik xüsusiyyətlərini daha dəqiq qiymətləndirməyə və hava axınları ilə yuyulan əyləc sistemlərinin soyudulmasını yaxşılaşdırmağa imkan verən nəzəri və eksperimental tədqiqatlar, habelə sərhəd şərtləri və ölçüsüz kriteriyalar təqdim olunub. Nəqliyyat vasitəsinin barabanlı-kündəli əyləcinin sürtünmə cütlərində konvektiv istilik ötürülməsini qiymətləndirmək üçün kriteriyalar tənliyi əldə edilib. Nəqliyyat vasitələrinin xüsusiyyətləri ilə tsiklik (dövri) və uzunmüddətli əyləc rejimlərinin istilik yükü Birləşmiş Millətlər Təşkilatının Avropa üzrə İqtisadi Komissiyasının (UNECE) 13 saylı qaydasına uyğun olaraq nəzərə alınıb. Göstərilmişdir ki, istilik yükünə baraban halqasının tutqun və cilalanmış səthlərində müxtəlif qalınlıqda dinamik və istilik təbəqələrinin əmələ gəlməsi təsir edir və bu da öz növbəsində, onların istilik mübadiləsinin intensivliyinə təsir göstərir. Məlum olmuşdur ki, baraban halqasının materialı ilə müqayisədə kündənin materiallarının istilik keçiricilik əmsalı aşağı olduğundan çənbərdə səthə yaxın təbəqədə istilik təbəqəsinin qalınlığı orta hesabla 26% azdır ki, bu da eksperimental tədqiqatlarla təsdiqlənib.

Açar sözlər: diskli-kündəli əyləc, sürtünmə cütü, istilik modeli, səthaltı və səth təbəqəsi, istilik təbəqəsinin qalınlığı.

Теоретическое и экспериментальное исследование охлаждения пар трения тормозов

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Аннотация. В работе приводятся теоретические и экспериментальные исследования, которые позволяют более точно оценить тепловые характеристики и улучшить охлаждение тормозных систем, омываемых потоками воздуха, а также граничные условия и безразмерные критерии. Получено критериальное уравнение для оценки конвективного теплообмена в парах трения барабанно-колодочного тормоза транспортного средства. Рассмотрена теплонагруженность циклического и длительного режимов торможения с их особенностями транспортных средств согласно правилу №13 ЕЭК ООН. Показано, что на теплонагруженность влияет образование динамических и тепловых слоев различной толщины на матовой и полированной поверхностей обода барабана, что, в свою очередь, влияет на интенсивность их теплообмена. Выявлено, что из-за меньшого коэффициента теплопроводности материалов накладки в сравнении с материалом обода барабана, толщина теплового слоя в приповерхностном слое в накладке в среднем на 26% меньше, что подтверждено экспериментальными исследованиями.

Ключевые слова: дисково-колодочный тормоз, пара трения, тепловая модель, поверхностный и приповерхностный слои, толщина теплового слоя.

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Introduction

The dynamics of the braking process is inextricably linked with the friction properties of the friction pair materials, which in turn depend on the speed, load and temperature conditions on the friction contact, as well as the influence of the environment. The impact of these factors on the dynamic coefficient of friction for different materials is different. This is due to their physical and mechanical properties, the nature and intensity of electrical, thermal and chemical processes on the friction contacts of the microprotrusions of the surface layers of the friction pair materials.

In this connection, during a non-stationary process of electrothermomechanical friction, which is usually the frictional interaction of conjugate local surfaces, the change of all process parameters over time is interconnected and interdependent. In other words, changes in speed, load, braking torque, temperature during braking are interconnected and depend on the frictional, mechanical, thermal-physical, electrical properties of the materials of the friction pair, the design of the friction elements and the operating mode of the brake.

Heat exchange by convection due to air flows washing friction pairs of drum-shoe brakes of vehicles and reduces their energy load. Based on this, it is necessary to consider the dynamics of convective heat exchange.

Analysis of literary data and problem statement. The traditional source of information on temperature, convective, radiation and heat-conducting fields during frictional interaction of brake friction pairs are theoretical and experimental studies, which are carried out on full-scale samples and models.

It should be said that due to the presence of accompanying transfer processes, complex heat exchange is much more complex than purely conductive, convective and radiation heat exchange, which significantly complicates its analytical and experimental studies. In this regard, complex heat exchange processes are currently relatively poorly understood.

Let us dwell on the convective heat exchange that takes part in the complex forced cooling of friction pairs of braking devices used in mechanical engineering.

Dependences of the heat transfer coefficient (fictitious value) on the speed of the vehicle according to E.A. Chudakov, V.M.Kazarinov, B.P. Karvatsky, A.L. Kryukov, D.M. Kaminsky and other researchers [1] showed the following. At V = 20 m/s, the values of α obtained by some researchers fall within the range of 40.5-52.2 W/(m^2·°C), while others fall within the range of 104.4-116.0 W/(m^2·°C). The latter data are clearly overestimated.

The diagram of the thermal model in the coordinate system "brake drum rim thickness - washing air temperature" is presented in [2]. The latter involves 23 parameters, 8 of which are temperature. The air temperature varies from 1.0 °C to 3.5 °C, which is impossible to record under operating conditions.

Convective heat exchange in a homogeneous medium (heat transfer) [3, 4] was illustrated for the first time using dynamic and thermal air layers on the surface of a metal element. However, nothing was said about the behavior of air layers during forced cooling of friction pairs of braking devices.

The aim of the work is to substantiate the occurrence of heat exchange layers during frictional interaction of brake friction pairs and to develop thermal models with their participation for assessing convective heat exchange.

Research methods

The method is the theory of thermal similarity with its theorems, boundary conditions, dimensionless criteria and, in this case, the obtained criteria equation for assessing convective heat exchange in friction pairs of a drum-shoe brake of a vehicle.

Thermal models of brake friction pairs. It is known that the mathematical description of the process of forced convective heat transfer is made up of the following equations: heat conductivity, movement of the washing medium and its continuity; heat exchange together with the conditions of uniqueness. In most cases, it is not possible to solve this system of equations analytically, therefore, it is necessary to rely on an experiment to estimate the energy load in time of the air-washed brake drum.

The results of theoretical research are generalized in relation to the laws of similarity of processes, effects and phenomena.

Similarity laws have great potential for studying/defining/forecasting the power-to-weight ratio of various friction pairs of brake devices. The latter are considered in pairs as those that are operated under conditions of geometric similarity, as well as washing media of the brake friction pair. In case of compliance with the specified similarity conditions, different types of friction pairs of brake devices are connected by some regularities capable of improving some elements of heat exchange theory. In this case, it is envisaged to create such a design of the brake friction pair in which the surface temperature would not exceed the permissible one for the materials of the friction lining.

The basic provisions of the theory of similarity belong to three main theorems. The first of them states: similar processes, phenomena and effects have the same similarity criteria - dimensionless complexes composed of quantities that describe them. The second asserts that the dependencies between variables that determine processes, phenomena and effects can be represented as a dependency between similarity criteria $K_1, K_2, \dots K_n$:

$$f\,(K_1, K_2, \dots K_n) = 0 \qquad (1)$$

Ratio (1) is called the similarity equation or the criterial equation. The third theorem answers the question of what conditions are necessary and sufficient for processes, phenomena and effects to be similar. The latter will be similar when the unambiguity conditions are similar, and the criteria include quantities that are in the unambiguity conditions and are the same. The latter determines the invariance of the defining criteria, which confirms the similarity. The sameness of the criteria in magnitude, which also includes quantities that do not appear in the unambiguity conditions, emerges by itself - as a consequence of the established defining criteria of similarity.

Dimensionless characteristics complexes - criteria retain their specific values in any measurement system. In addition, similarity criteria determine, as they say, the "correct" relationship for the flow of processes, effects and phenomena of variable dimensional values.

Let us dwell on the physical meaning of the similarity criteria that will appear in the criteria equations for determining heat transfer coefficients

$$Bi = CNu^m Re^n Pr^l\,, \qquad (2)$$

where $Bi = \frac{\alpha_2 \delta}{\lambda_1} = \alpha_2 R = \frac{\alpha_2}{K_1}$; $Nu = \frac{\alpha_1 d}{\lambda_2}$;

$$Re = \frac{wd}{\nu}; \quad Pr = \frac{\nu}{a} = \frac{\mu c_1}{\lambda_2} = \frac{\nu \rho c_1}{\lambda_2};$$

– dimensionless complexes, numbers, terms, invariants, similarity criteria: *Bio*, *Nusseli*, *Reynolds*, *Prandtl*, respectively: C, m, n, l - constants; α_1, α_2 - heat transfer coefficients from different sides of the metal friction element; δ - thickness of the metal friction element; R - thermal resistance of thermal conductivity; K - heat transfer coefficient through the metal friction element; λ_1, λ_2 - thermal conductivity coefficients of the metal and the washing medium; d - characteristic size; w - velocity; $\nu = \mu / \rho$ - coefficient of kinematic viscosity; ρ - density; μ - coefficient of dynamic viscosity; a - thermal diffusivity coefficient ; Cp - isobaric heat capacity.

Criterion *Bi* can be considered as the ratio of the parameters of the forced and natural cooling processes when washing a mixture of components of the washing medium with high-speed currents to corresponding to the parameters of the processes of conductive heating of the layers of the metal friction element when the thermal currents reach a given value its depths.

Reynolds criterion (*Re*) determines the hydrodynamic similarity of the washing coolant, and the Prandtl criterion (*Pr*) is a thermophysical characteristic of the coolant (it contains only physical constants). In the case of constancy of the criterion *Re* condition of constancy of criterion *Pr* provides thermal similarity (similarity of heat flows and temperature pressure fields). Together, the condition *Re* = idem and *Pr* = idem are the conditions of invariance of the criteria. In this case $Re\,Pr = \frac{\nu d}{a} = Pe$ - the *Peclet* similarity criterion. In convective heat exchange processes, the Nusselt criterion characterizes the idem ratio and is a consequence of the established similarity.

On the other hand, the Nusselt criterion describes the heated boundary layer of air, the *Re* criterion - boundary layer of the washing air flow, and the criterion *Pr* - the relationship between the above-mentioned boundary layers and at the same time $\delta_{н}/\delta_w = \left(\sqrt{Pe}\right)^{-1}$; (when *Pr* = 1, the layers have the same thickness).

The analysis of the values of similarity criteria in a conceptual approach from the position of synergetics and fractals in tribology made it possible to use the Janahmadov-Volchenko criterion to assess the state of the cooled friction belt of the brake disc and the energy load of its surfaces and near-surface layers.

Criterion ratios:

$$Bi/Nu = b\lambda_c/(\lambda h_c) = R_1/R_2 = JV,$$

where R_1, R_2 are the thermal resistances of the layers of the metal friction element and the circulating medium, which is in a vapor or liquid state between the microprotrusions of the friction surfaces; JV is the Janahmadov-Volchenko criterion.

The obtained ratio can be considered as the product of the thermal resistances of the surface and near-surface layers of the friction belt of the disk, to which the layers of the washing medium "stick" to the surface temperature of the polymer lining below the permissible temperature for its material. At high surface temperatures (above the permissible temperature), the binders of the polymer material burn out [5, 6]. The resulting products mix with the circulating air layers. Then, islands of liquid appear on the surface of the lining, which stick to the working surface of the drum rim. The controlling effect on the value of thermal resistance of the surface layer of the polymer lining is its phase state (solid, liquid, gaseous). For the surface and subsurface layers of the brake drum rim, the controlling effect is their energy load. According to the value of the Janahmadov-Volchenko criterion, it is possible to predict the energy load of the surface and subsurface layers of metal-polymer friction pairs [7].

Therefore, the similarity equation - the criterial equation (3) for forced convective heat transfer processes has the form

$$f(Bi, Nu, Re, Pr) = 0,$$

or $$Bi = (Nu, Re, Pr) = 0.$$

and in its final form the criterial equation looks like

$$\frac{\alpha_2}{K_1} = C\left(\frac{\alpha_1 d}{\lambda_2}\right)^m \left(\frac{wd}{\nu}\right)^n \left(\frac{\nu\rho c_p}{\lambda_2}\right)^l. \quad (3)$$

Thus, α_2 and K_1 were subsequently determined with known coefficient C and exponent m, n and l.

Calculations performed using the criteria equation (3) allowed us to obtain the values of the heat transfer coefficients from the outer and inner (with the brake open) surfaces of the drum rim. Thus, for the outer surface it varies from 20.0 to 50.0 W/(m^2·°C), and for the inner surface – from 5.0 to 15.0 W/(m^2·°C) [8-10].

Let us dwell on the braking modes of the KrAZ-250B vehicle, which is subject to cyclic (type I) and long-term (type II) braking according to regulation No. 13 of the UNECE [10]. The cyclic mode is carried out on horizontal road sections and includes 20 brakings: acceleration of the vehicle to a speed of 80 km/h and braking to 40 km/h and then after 15 seconds a new cycle (duration is one minute). The long-term braking mode is carried out in mountainous terrain for 12 minutes on a slope of 6°, a length of 6 km with a constant intensity of vehicle braking - 30±5 km/h.

Fig. *a*, *b* shows a schematic representation of the thermal model of friction pairs in the working (a) and open (b) state.

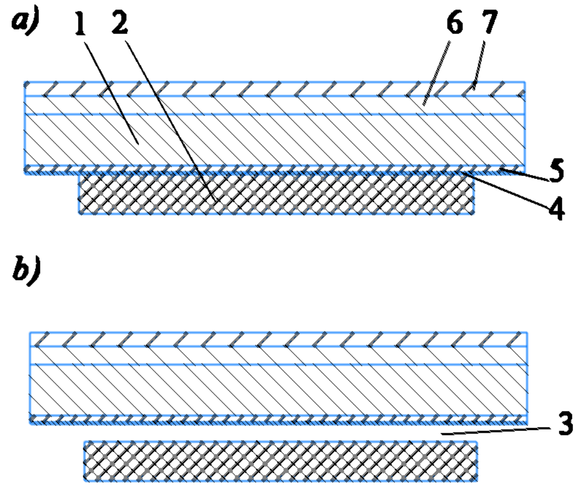

Figure a, b - Schematic representation of the thermal model of friction pairs: a, b – in the working and open state: 1 – brake drum; 2 – friction lining; 3 – gap between the friction pair; 4,5 – surface and subsurface layer; 6, 7 – dynamic and thermal layer

According to Fig. *a*, *b*, which illustrates the dynamic 6 and thermal 7 layers on the outer surface of the drum rim, it is the dynamic layer 6 that is the substrate for the thermal layer 7 and is located between the microprotrusions of the outer surface of the drum rim. In this case, the dynamic layer 7 "sticks" to the said surface and rotates together with the drum rim, removing heat from it into its layer. The latter is torn off when the density gradient of the dynamic layer 7 becomes less than the density gradient of the washing air flows moving with a longitudinal velocity. A small part of the washing air flow "settles" between the microprotrusions of the outer surface of the drum rim, forcibly cooling it. This is how convective heat exchange of vehicle brake drums occurs.

Let us move on to examining the electrothermal layer that arises in the mating friction pairs of the brake friction unit.

Determination of the thickness of the electrothermal layer on the rim of a brake drum. Electrothermomechanical friction is characterized by electrical and thermal currents, which are on microprotrusions form electrothermal contact spots. The thickness of the latter in the friction pairs of the brake plays a significant role in the formation of their specific electrothermal resistance, and as a consequence, the energy load of the surface and near-surface layers of the brake friction pairs (Table 1) [6].

Table 1 - Calculation dependencies for determining the ratio $\frac{t_{1,2}(\delta_{1,2},\tau)}{t_{1,2}(0,\tau)-t_c}$

Name of dependency	General appearance
One-dimensional control thermal conductivity during pulsed heat supply to friction pairs under the following conditions:	$\frac{\partial t_1}{\partial \tau} = \alpha_1 \frac{\partial^2 t_1}{\partial z_1}$; (7)
initial;	$t_1(\infty,\tau) = t_c$; (8)
boundary.	$q_1(0,\tau) = -\lambda_1 \frac{\partial t_1}{\partial z_1} = q_1 = const$, (9) $\frac{\partial t_1(\infty,\tau)}{\partial z_1} = 0$; (10)
The solution of the linear equation under given conditions allows us to obtain the temperature distribution with an increase in the bulk temperature in the layers $Z_{1,2} = \delta_{1,2}(\tau)$	$t_{1,2}(\delta_{1,2},\tau) - \frac{2q_{1,2}\sqrt{\alpha_{1,2}\tau}}{\lambda_{1,2}} ierfc \frac{\delta_{1,2}}{2\sqrt{\alpha_{1,2}\tau}}$; (11)
On the surface of the friction track of the drum rim at $z_{1,2} = 0$ we obtain	$t_{1,2}(0,\tau) - t_c \frac{2q_{1,2}\sqrt{\alpha_{1,2}\tau}}{\lambda_{1,2}\sqrt{\pi}}$; (12)
Taking for equation (5), we $\delta_{1,2} = k_1\sqrt{\alpha_{1,2}\tau}$ obtain (Table 2)	$t_{1,2}(\delta_{1,2},\tau) - \frac{2q_{1,2}\sqrt{\alpha_{1,2}\tau}}{\lambda_{1,2}} ierfc \frac{k_1}{2}$; (13)
Dividing the left and right parts of dependencies (6) and (7) we obtain (Table 3)	$\frac{t_{1,2}(\delta_{1,2},\tau)}{t_{1,2}(0,\tau)-t_c} = \sqrt{\pi} ierfc \frac{k_1}{2}$. (14)

At the initial moments of time, electrothermomechanical friction is of a pulsed nature and therefore the brake disc does not have time to warm up to its entire thickness. In friction pairs with materials with different thermal conductivity coefficients $\lambda_1 >> \lambda_2$ at the end of braking $t_1(\delta_1, \tau_T) = t_2$ (where t_1, t_2 are the temperatures of the surface layer with thickness δ_1), of the friction belt at the beginning and end of braking over time τ_T). In this case, the increments of bulk temperatures in the layers $z_{1,2} = \delta_{1,2}(\tau)$ are negligibly small compared to the increment of temperatures on the surface of the belt friction of the drum rim.

$$\alpha_{1,2} = \lambda_{1,2} / (c_{1,2}\rho_{1,2}), \quad (4)$$

where $\alpha_{1,2}$ is the coefficient of thermal diffusivity of the materials of the friction pair; $c_{1,2}$ - specific heat capacity and density of the friction pair materials, J/K; $\rho_{1.2}$ – density of friction pair materials, kg/m^3.

If we accept

$$\frac{t_{1,2}(\delta_{1,2},\tau)-t_c}{t_{1,2}(0,\tau)-t_c} = 0{,}01$$

i.e. the increment in volume temperature in the layer $z_{1,2} = \delta_{1,2}$ is 1.0% of the increment in temperature on the friction surface at $k_1 = 3.2$ [11-13].

Thus, the thickness of the thermal layer on the surface of the friction track is determined by the dependence of the type

$$\delta_{1,2}(\tau) = 3{,}2\sqrt{\alpha_{1,2}\tau}. \quad (5)$$

As for the thickness of the electric layer that appears on the friction belt of the drum rim, it is determined by the dependence of the type

$$\delta'_2(\tau) = 0{,}05\sqrt{\rho_e/(\nu\mu)}, \quad (6)$$

where ρ_e is the specific electrical resistance, (Ohm·mm^2)/m; ν - the oscillation frequency of the microprotrusions, s^{-1}; μ - the relative magnetic permeability of the materials of the microprotrusions [14].

Table 4 presents the results of calculations of the electrothermal layers of the drum rim of the friction pair “SCh15 – FK – 24A” with its maximum energy load.

Thus, the relationships have been established: the thickness of the dynamic thermal layer on the matte and polished surface of the drum rim, as well as the thickness of the electrothermal layers on the working surface of the brake friction pairs.

Table 2 - Meaning of the *ierfc function* $k_1/2$ depending on $k_1/2$

$k_1/2$	0	0.2	0.5	0.8	1.0	1,2	1.4	1.6
ierfc $k_1/2$	0.5642	0.3866	0,1996	0.0912	0,0503	0,0260	0,0127	0.0058

Table 3 - The value of k_1 for the relationship $\frac{t_{1,2}(\delta_{1,2},\tau)}{t_{1,2}(0,\tau)-t_c}$, varying from 0.01 to 0.25

$\frac{t_{1,2}(\delta_{1,2},\tau)}{t_{1,2}(0,\tau)-t_c}$	0.01	0.05	0.10	0.15	0.20	0.25
k_1	3.20	2.40	1.94	1.69	1.44	1.28

Table 4 - Results of calculations of the electrothermal layers of the friction belt of the friction pair "SCh15 - FK-24A" of the drum-shoe brake

Thickness of electric heat - Number of layers, mm:		Time of pulsed electric and thermal currents, $\tau \cdot 10^{-4}$, s							
		1.0	3.0	5.0	7.0	9.0	11.0	13.0	15.0
overlays	δ_1	0.025	0.043	0.055	0.066	0.074	0.082	0.089	0.096
disk	δ_2	0.094	0.163	0.211	0.249	0.283	0.313	0.340	0.365
	δ'_2	0,013	0.023	0.026	0.031	0.033	0.035	0.034	0.037

Discussion of the results

Theoretical and experimental studies of the dynamics of convective heat exchange of the friction pairs of the brakes of moving vehicles washed by air flows have made it possible to determine:

- selection of the method of thermal similarity theory with its theorems, boundary conditions, dimensionless criteria and, in this case, the obtained criteria equation for assessing convective heat transfer in friction pairs of a drum-shoe brake of a vehicle;

- the thermal load of cyclic and long-term braking modes with their characteristics of vehicles according to UNECE Regulation No. 13 affects the formation of dynamic and thermal layers of different thicknesses on the matte and polished surfaces of the drum rim, which affects the intensity of their heat exchange;

- with an increase in the duration of action of pulsed electric and thermal currents, the thickness of the electrothermal layers also increases;

- due to the lower thermal conductivity coefficient of the lining materials than the drum rim material, the thickness of the thermal layer in the surface layer in the lining is on average 26% smaller;

- when comparing electrical and thermal layers, it is clear that the thickness of the thermal layer is an order of magnitude higher than the thickness of the electrical layer, which is confirmed by numerous studies.

Conclusion

Thus, the convective heat exchange of friction pairs of the drum-shoe brake of the vehicle was assessed using the dynamics of the flows of washing air of the environment.

Conflict of Interests

The authors declare there is no conflict of interests related to the publication of this article.

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