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REVIEW PAPER

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COMPLEX HEAT EXCHANGE IN BRAKE FRICTION PAIRS

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ABSTRACT

In the article, the following issues are considered: general principles of complex heat transfer in friction pairs of brake devices; radiant heat exchange from the rough surfaces of the metal friction elements of the brakes; the combined action of the fields of thermal conductivity and radiation in the metal friction elements of the brakes. The role of the thermal and dynamic boundary layers resulting from air-washing of metal brakes with open friction pairs and their effect on cooling is discussed. A new relationship between the Biot and Nusselt criteria is shown, which is presented in the form of thermal resistances of the surface and near-surface layers of the brake pulley rim. A comparison of the combined action of the fields of thermal conductivity and radiation in the metal friction elements of the brakes showed that the radiation efficiency is 1.5–1.7 $W \cdot m^{-2}$ times higher than the thermal conductivity. Radiation efficiency is the ratio of the power of the heat flux to the unit area through which the heat flux passes.

Keywords: braking device, friction pair, metal friction element, convective, radiation and thermal conductivity

INTRODUCTION

It would not be an exaggeration to say that heat transfer processes are essential and often decisive in almost all areas of thermodynamics and heat transfer; they increase the importance of knowing the laws governing heat transfer processes. At present, the theory of heat transfer represents an extensively well-developed area of knowledge, without which the design and operation of friction pairs of braking devices is impossible.

In all of these cases, when a moving working body interacts with the surface of structural elements of friction pairs of brake devices, there is an intensive heat exchange process called heat transfer (Goodz, Globchak, Koliasa & Yavorskiy, 2007; Sakhin, 2013; Dzhanakhmedov et al., 2016a, 2016b; Kindrachuk, Volchenko, Volchenko, Stebeletskaya & Voznyi, 2017). The main components of heat transfer, and the most complex and laborious in calculations, are radiation, convection and thermal conductivity fields of metal friction elements washed by transverse and longitudinal air flows.

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The operating temperature of brake friction pairs and the ratio of the actual surface temperature of their metal friction elements depends on the degree of equal load on the friction pairs, the friction heat resistance of the linings, and a number of other factors. Therefore, none of the above parameters allows one to objectively judge the energy intensity of the brake friction units. With the same values of operating parameters, one brake mechanism may have sufficient energy consumption, and the other may be insufficient. In view of this, in the study of the processes, phenomena and effects of heating and free and forced cooling of friction units, the experiment, brought in accordance with the requirements of the similarity theory, is of great importance (Balakin & Sergienko, 1999; Nosko, 2010). The results of such an experiment can be used to determine homogeneous quantities in systems in which similar phenomena occur.

According to this theory, research results must be processed using similarity criteria and simplifications, and the dependencies between them should be presented only in the form of criterion equations. Since such an equation is valid only for a group of similar phenomena, it is necessary to know the similarity condition.

In accordance with the third similarity theorem (Dzhanakhmedov et al., 2021), similar phenomena are those in which similar conditions of uniqueness occur. In our case, the conditions of uniqueness are geometric, physical, initial, and boundary conditions. The exact mathematical formulation of the unambiguity conditions is very complex and is fulfilled only for some of the simplest systems of differential equations.

For the brakes of hoisting-and-transport machines, these conditions were formulated by Aleksandrov. He also obtained a criterion equation that connects the temperature complex with similarity criteria, simplexes, which follow from the uniqueness condition. A similar work on the thermal calculation of the brakes of lifting and transport equipment based on the planning of the experiment was carried out by Volchenko (Dzhanakhmedov et al., 2016a, 2016b).

A criterion equation for determining the temperatures of friction surfaces and heat transfer coefficients from metal friction elements of brakes has been proposed by a number of authors (Nosko, 2010; Sakhin, 2013; Dzhanakhmedov et al., 2016a, 2016b).

Considering these equations, it is clear that, along with the positives, they are not devoid of significant drawbacks, the main of which are the following:

- limited scope due to the fact that there is no complete geometric and hydromechanical similarity between the brake mechanisms;
- the difference in the specific power loads of the braking mechanisms is not taken into account;
- some average temperature of the friction surface is determined;
- the setting of boundary conditions of the second kind is carried out on the basis of data on a certain long-term or intermittently operating mode;
- it is impossible to accurately predict the temperature mode of the brakes while the car is moving along the specified real route.

Analysis of the reviewed works revealed the following shortcomings (Kernytskyy et al., 2022):

- the heat exchange processes, despite their different nature, must be represented in pairs “convection – heat conduction”, “heat conduction – radiation”, and “convection – radiation” in the fields of their joint interaction.
- to establish the influence of thermal and dynamic boundary layers, which are washed by air flows, on the cooling efficiency of brake friction pairs when they are open.

The research mainly refers to:

- general principles of complex heat transfer in friction pairs of braking devices;
- radiant heat exchange from the rough surfaces of the metal friction elements of the brakes;

– the combined action of the fields of thermal conductivity and radiation in the metal friction elements of the brakes.

The purpose of the work is to substantiate the features of complex heat transfer in friction pairs of brakes and to establish the reasons for their low efficiency when they are open.

GENERAL PRINCIPLES OF COMPLEX HEAT TRANSFER IN FRICTION PAIRS OF BRAKE DEVICES

In friction pairs of brakes, the main system of heat storage and heat dissipation is their metal friction elements (brake pulleys and drums: solid and self-ventilated discs).

Figures 1a and b show the friction pairs of a disc-pad brake. They consist of friction linings (2), which are located on fixed brake pads. During the frictional interaction of the working surfaces of the linings (2) with the rotating brake disc (1) under the influence of normal pressing force (N), a friction treadmill (3) is formed.

Figure 1c illustrates the friction unit of the band-shoe brake of a drilling drawworks. When the brake band (4) is tightened under the action of normal pressing force (N), the working surface of the friction lining (2) interacts with the friction track of the rim of the pulley (5). The latter is connected to the drum flange using the mounting lug (8).

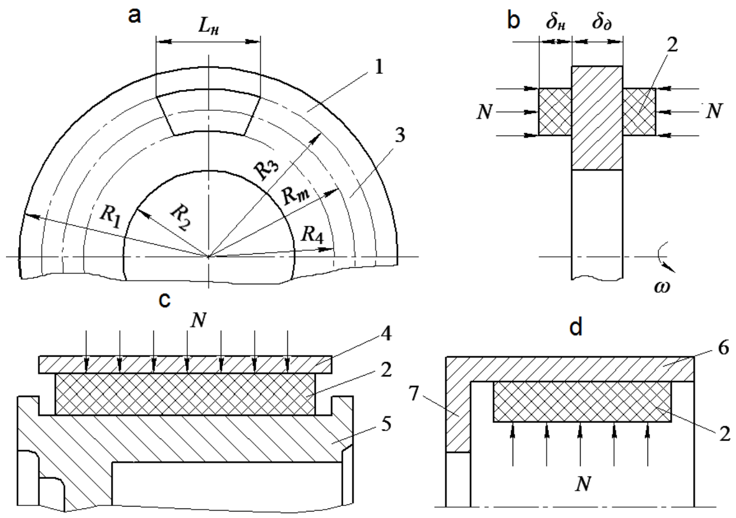

Fig. 1. Diagrams of various types of friction units of brake devices: a – disk-block (longitudinal section); b – disk-block (transverse); c – band-block (cross-section); d – drum-block (cross-section)

Source: own work.

The friction unit of a car's drum-shoe brake is shown in Figure 1d. The unit contains a brake drum rim (6) with a flange (7), as well as friction linings (2) located on the brake pads. When the latter are decompressed in their toe and heel parts, the working surfaces of the linings (2) frictionally interact with the inner (working) surface of the rim of the brake drum (6).

The heat storage capacity of metal friction elements is estimated by the specific heat flux (q) [$W^{-1} \cdot m^{-2}$], which penetrates their body and contributes to the creation of a heat conduction field in their cross-section. The polished and matt surfaces of the metal friction elements create radiation and convection fields around them. However, on their edge, there are thermal and dynamic boundary layers of the washing air with open friction pairs of the brake.

The thermal boundary layer is the layer on the working surfaces of the pulley or drum rim, in which the thermal disturbance of the air washing them is significantly manifested. Thermal boundary layer thickness (δ_T):

$$\frac{\delta}{\delta_T} = \mathrm{Pr}^{0.5}, \tag{1}$$

where:
Pr – Prandtl criterion [-],
δ – dynamic boundary layer thickness [mm].

The thickness of the dynamic boundary layer is determined based on the assessment of the inertia force of the external air flow and the friction force in the boundary thermal layer, from the opposition of which this layer arises (Sakhin, 2013). Dynamic boundary layer thickness (δ):

$$\delta = B \cdot \mathrm{Re}^{-0.5}, \tag{2}$$

where:
B – pulley or drum rim width [mm],
Re – Reynolds criterion [-].

After expression substitution (1) in (2), we get that

$$\delta_T = \frac{B \cdot \mathrm{Re}^{-0.5}}{\mathrm{Pr}^{0.5}}. \tag{3}$$

Let us consider several different “standard” criteria, on the basis of which we obtain one of the main “non-standard” criteria that establish a connection between the thermophysical parameters of friction pairs and high-speed air currents washing them, as well as mixtures in a gaseous and liquid state. Four criteria will be presented below: the Reynolds number, the Prandtl number, the Biot number and the Nusselt number.

The effect of viscosity on the movement of the coolant is characterised by the Reynolds number (Re), which is defined as the ratio of the mass inertia force to the frictional force in the flow.

The inertia force of one-dimensional motion, referred to as the unit volume of the coolant, is equal to $\rho \frac{\mathrm{d}v_x}{\mathrm{d}t}$, and after formal transformations, we can write that $\rho \frac{\partial v_x}{\partial x} \frac{\partial x}{\partial t} = \rho v_x \frac{\partial v_x}{\partial x}$, where $\frac{\partial v_x}{\partial x}$ is a change in speed when moving from one point in the flow to another. The friction force per unit volume is $\frac{\partial \tau}{\partial y} = \mu \frac{\partial^2 v_x}{y^2}$.

Let us introduce constant characteristic flow parameters, such as density (ρ), viscosity (μ), a certain characteristic velocity (v_c), and a certain characteristic size (B). Then, the order of the sought forces will be $\rho v_x \frac{\partial v_x}{\partial x} \sim \rho \frac{v_c^2}{B}$; $\mu \frac{\partial^2 v_x}{\partial y^2} \sim \mu \frac{v_c^2}{B}$.

Let us define Re as the ratio of inertial forces to friction forces:

$$\rho \frac{v_c^2}{B} \frac{B^2}{\mu v_c} = \frac{\rho v_c B}{\mu} = \frac{v_c\, B}{\nu} = \mathrm{Re}. \tag{4}$$

The steady air movement, depending on the ratio of inertial forces and forces (4), can be laminar and turbulent.

The Prandtl number, defined as the ratio of kinematic viscosity to thermal diffusivity, is of great importance in describing the relationship between the velocity and temperature fields (similar to kinematic thermal conductivity) of the coolant, i.e. the ratio of the intensity of the exchange of momentum at the molecular level of a substance to the exchange of heat between the layers of the flow:

$$\mathrm{Pr} = \frac{\nu}{a} = \frac{c_p \mu}{\lambda}. \tag{5}$$

The second important number is the Biot number:

$$\mathrm{Bi} = \alpha_c \frac{b}{\lambda}, \tag{6}$$

where:

α_c – coefficient of heat transfer from the matt and polished surface of the pulley to the high-speed currents of the environment washing them [$\mathrm{W \cdot m^{-2} \cdot {}^\circ C^{-1}}$],

b – effective depth of penetration of heat into the body of the pulley rim [m],

λ – coefficient of thermal conductivity of the pulley material [$\mathrm{W \cdot m^{-1} \cdot {}^\circ C^{-1}}$],

α – coefficient of thermal diffusivity of the coolant [$\mathrm{m^2 \cdot s^{-1}}$],

c_p – isobaric specific heat capacity [$\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$].

The Bi can be considered as the ratio of the parameters of the cooling processes when the mixture of environmental components is washed with high-speed currents to the corresponding parameters of the conductive heating processes of the layers of the brake pulley when the thermal currents reach a given depth of its rim.

The next criterion is the Nusselt number:

$$\mathrm{Nu} = \alpha_c \frac{h_c}{\lambda_c}, \tag{7}$$

where:

α_c – coefficient of heat transfer from the matt and polished surface of the pulley to the high-speed currents of the environment washing them [$\mathrm{W \cdot m^{-2} \cdot {}^\circ C^{-1}}$],

λ_c – coefficient of thermal conductivity of layers of high-speed currents of the washing medium [$\mathrm{W \cdot m^{-1} \cdot {}^\circ C^{-1}}$],

h_c – thickness of layers of high-speed currents of the washing medium [m].

In the written expression, the parameters can be considered as relative values of the heat flux density transferred in the heat transfer process from the polished and matt surfaces of the brake pulley when they are washed by high-speed currents of the medium, and the heat flux density through its layers of thickness (h) due to thermal conductivity.

The analysis of the values of the similarity criteria in the conceptual approach from the standpoint of synergy and fractals in tribology made it possible to propose the Volchenko criterion for assessing the state of the cooled working surface of the brake pulley rim and the energy load of its surface and near-surface layers which is determined by the following dependence (Dzhanakhmedov et al. 2021):

$$\frac{\mathrm{Bi}}{\mathrm{Nu}} = \frac{b \lambda_c}{\lambda h_c} = \frac{R_1}{R_2} = V_0. \tag{8}$$

The resulting ratio can be considered as the product of the thermal resistances of the surface and near-surface layers of the brake pulley rim, to which thermal boundary layers, caused by passing dynamic boundary layers of the washing medium, seem to "stick" at the surface temperature of the polymer lining below the permissible level for its materials. At high surface temperatures (higher than permissible for friction lining materials), the binder components of the polymer material burn out. The resulting products are mixed with the circulating layers of air. Then, islands of liquid appear on the surface of the lining, which seem to stick to the working surface of the pulley rim. The control effect on the value of the thermal resistance of the surface layer of the polymer patch is its phase state (solid, liquid, gaseous). For the surface and subsurface layers of the brake pulley rim, the control effect is their energy load. By the value of the Volchenko criterion, it is possible to predict the energy loading of the surface and subsurface layers of metal-polymer friction pairs.

RADIANT HEAT TRANSFER FROM THE ROUGH SURFACES OF THE METAL BRAKE FRICTION ELEMENTS

A number of researchers have obtained solutions for the case when the surface is represented by a model with the following properties:

i) The root mean square roughness (h) is less than the electromagnetic wavelength (λ) more than 100 µm.
ii) The surface is perfectly conductive.
iii) The distribution of roughness heights obeys the Gaussian distribution.

According to Berkhovskykh's method, we consider a plane monochromatic wave with amplitude A = 1. At an arbitrary point P(x, y, z) of space, the field of the wave diffracted from the surface is sought (Fig. 2a).

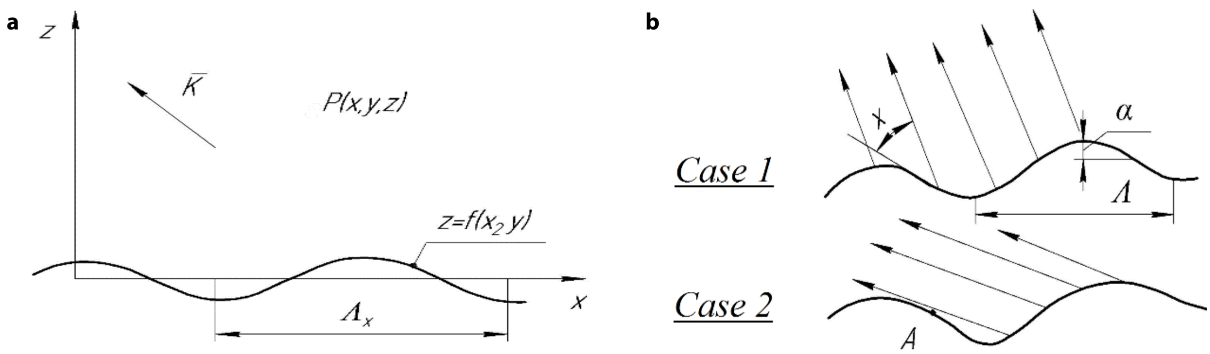

Fig. 2. Considered model of the brake: a – reflection of radiation from the rough surface of a metal friction element; b – Cases 1 and 2

Source: own work.

The coordinate system is chosen so that the X_0Z plane lies in the plane of the rise of the wave. Surface coordinates are denoted by X, Y, Z. This surface is covered with periodic protrusion. Reflecting the surface equation $Z = f(X, Y)$, function Z is periodic. Let us denote by $R = \sqrt{(x-X)^2 + (y-Y)^2 - (z-Z)^2}$ the distance between the desired point P and the current coordinates of the rough surface. As a result of the passage of a wave through a rough surface, an electric potential arises between the microprotrusions of the metal element, as well as the contact spots of the microprotrusions and the levels of the liquid phase of the linings. Then, using the Kirchhoff formula (Belyaev, 1989), one can find the electromagnetic potential at the point P:

$$\psi(P) = \frac{1}{4\pi}\int_F \left\{\psi \frac{\partial}{\partial n} x \left(\frac{e^{i\chi r}}{R}\right) - \frac{e^{i\chi r}}{R} \cdot \frac{\partial \psi}{\partial n}\right\} dF. \tag{9}$$

Double electrical layers are formed on the polished surface. Double electrical layers are formed on the polished surface. As a result, the electromagnetic potential varied from 0.5 to 1.0 V.

The difficulty lies in defining the boundary conditions. To determine ψ and $\frac{\partial \psi}{\partial n}$ on the surface, the following technique is applied. The reflected wave is considered to be flat. The connection between the waves is set through the reflection coefficient, taking into account the angle of inclination of the surface element at the place of the rise of the wave. A potential, for example, the electric vector of a wave at an arbitrary point on the surface is written as the sum of the potentials of the reflected wave:

$$\psi(x, y, z) = \exp\{l\chi_{\text{ref}} - nz\}\,[1 + R(\xi)], \tag{10}$$

where:
χ – wave numbers,
l, n – real numbers,
$R(\xi)$ – reflectivity, depending on the angle ξ between the normal n to the surface and the direction of the wave vector (χ_{ref}) of the reflecting wave.

The quantity $\frac{\partial \psi}{\partial n}$ for the reflected wave is determined by the dependence of the form:

$$\left.\frac{\partial \psi}{\partial n}\right|_{\text{ref}} = \chi_{\text{ref}}\psi_{\text{ref}}.$$

The total effect for the derivative (ψ) is determined by the dependence of the form:

$$\frac{\partial \psi}{\partial n} = \left(\frac{\partial \psi}{\partial n}\right)_{\text{ref}} = \chi_o \psi_o [1 - R(\xi)]. \tag{11}$$

In works by Chihos (1982) and Belyaev (1989), examples were considered of using the obtained relations to determine the pattern of radiation reflection from surfaces of various profiles. As a result, relations were obtained to determine the reflection of radiation from surfaces of various metal profiles. As an illustration, we present several diagrams characterising the scattered radiation (Fig. 3).

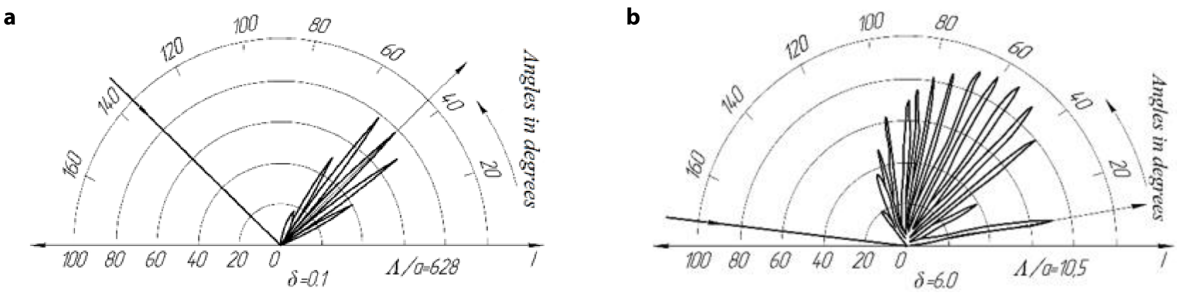

Fig. 3. Scatter diagram of radiation from a rough surface at: a – $\delta = 0.1$, $\frac{\lambda}{\alpha} = 628$; b – $\delta = 6.0$, $\frac{\lambda}{\alpha} = 10.5$

Source: own work.

It can be seen from the diagrams that diffuse scattering $\frac{2\pi}{A}$ = 0.1 takes place even with small surface irregularities. In this case, for the values $\frac{2\pi}{A}$ = 6 and $\frac{2\pi}{A}$ = 3 the maximum of the scattered radiation does not coincide with the direction of the specular reflection, it coincides with the direction of the specular reflection from the inflection points of the sinusoid, where the surface curvature is zero, and the rays experience the smallest discrepancy.

Thus, the Berkhovskikh theory confirms the diffuse nature of scattering even for surfaces that are considered "technically smooth" ($h \approx \lambda$).

COMBINED ACTION OF HEAT CONDUCTION AND RADIATION IN METAL FRICTION ELEMENTS OF BRAKES

Let us consider the combined action of the fields of thermal conductivity and radiation from the non-working surface of the metal friction elements of the brakes. To do this, we will estimate the heat flow for the case of the combined action of radiation and thermal conductivity in the form of the expression:

$$q = \frac{C_0\left(T_1^4 - T_2^4\right)}{1+\frac{3\tau}{4}} - \lambda\frac{T_1 - T_2}{\delta}, \tag{12}$$

where:
C_0 – emissivity, from a metal surface [W·m^{-2}·K^{-4}],
τ – heat flux [-],
T_1 – temperature on the polished surface of the metal element of the brakes [K],
T_2 – temperature on the matt surface of metal brake elements [K].

For simplicity, consider the case when $T_1^4 >> T_2^4$. Then:

$$-\frac{q}{C_0T_1^4} = \left[1+\frac{3\tau}{4}\right]^{-1} + \frac{4}{3}\frac{N}{\tau}\left(1-\frac{T_2}{T_1}\right). \tag{13}$$

The value of N characterises the ratio between molecular and "photon" thermal conductivity that is the share of thermal energy transfer by true thermal conductivity in comparison with radiation:

$$N = 4\frac{\lambda}{\lambda_{ph}} = \frac{\lambda}{\frac{4}{3}C_0T_1^3 l_{ph}}, \tag{14}$$

where:
λ_{ph} – photonic thermal conductivity coefficient [W·m^{-2}·K^{-1}],
l_{ph} – photon wavelength, i.e., the proportion of thermal energy transfer by true thermal conductivity compared to radiation.

In Figure 4, the comparison of the approximate solution for the combined action, radiation, and thermal conductivity – calculation by Eq. (9) – and the exact one obtained by Viksant and Grosh, as well as Lax by numerical methods from integro-differential transport equations, can be found.

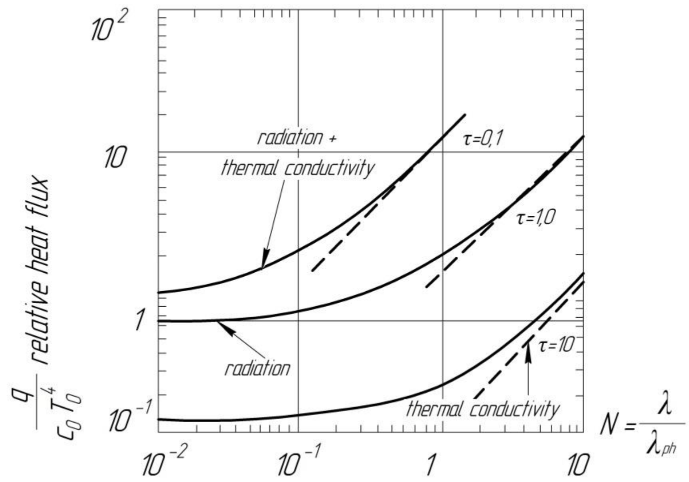

Fig. 4. Determination of the combined effect of thermal conductivity and radiation: the dashes show an approximate solution, and the solid curve shows an exact calculation

Source: own work according to works by Chihos (1982), USSR standard P60-54-97-88 (Gosudarstvennyy Komitet SSSR Po Standartam [Gosstandart SSSR], 1988), Volchenko et al. (2006), Dzhanakhmedov et al. (2016a, 2016b), Volchenko et al. (2019b), Kindrachuk et al. (2019), Volchenko, Volchenko, Polyakov, Fedotov and Evchenko (2019a), Kindrachuk et al. (2021).

Equation (9) was obtained by analytical methods of limiting relations [linearisation of Eq. (9)] near the boundary and taking into account only their thermal radiation far from the working surface of the pulley rim and drum.

Solution for dimensionless heat flux:

$$q_w = \frac{q}{C_0 T_0^4} \tag{15}$$

looks like:

$$q_w = \frac{3(1-\theta)N(T_1)+\left(1-\theta^4\right)}{\frac{3\tau}{4}+\psi(T_1)+\psi(T_2)}, \tag{16}$$

where: $\theta = \frac{T_2}{T_1}$; $\psi = f(N)$.

Table 1 shows a comparison of the exact solution for the dimensionless flux (q_w) with the derivation from Eq. (15), as well as a simple summation of the radiation flux, taking into account absorption and the heat conduction flux.

Table 1. Comparison of calculated dimensionless heat fluxes $\tau = 1$; $\theta = 0.5$

N	Exact solution	Acc. the Eq. (15)	Simple summation
0	0.518	0.525	0.536
0.01	0.596	0.573	0.556
0.10	0.798	0.795	0.736
1.00	2.600	2.470	2.536
10.00	20.600	20.300	20.536

From Table 1, for small values of N, the Howell and Goldstein method gives more satisfactory results than the summation of the fluxes. According to Figure 4, the summation of the flows of the combined action of thermal conductivity and radiation of the metal friction element is performed in an approximate and accurate way. Of greatest interest is the fact that using this method, it is possible not only to calculate the flows, but also to construct a temperature profile over the thickness of the pulley rim and drum. In this case, heat transfer, which is "closed" by the radiation field, is considered.

DISCUSSION AND CONCLUSION

Complex heat transfer, in comparison with the combined action of heat conduction, radiation, and free and forced convection due to the presence of concomitant processes, phenomena and transfer effects, is much more complex, which significantly complicates its analytical and experimental studies. Despite these circumstances, it seems possible to assert the following:
– when assessing the efficiency of free and forced air convection of metal friction elements of brakes, it is necessary to take into account the thermal and dynamic air layers. The first is a heat insulator, and the second increases the heat exchange during air exchange. The thermal boundary layer – an area of rapid temperature change – is characterised by a large value of the normal derivative of temperature $\partial t/\partial n$;
– the choice of similarity criteria and their application should occur not only from the modes of movement of the washing of air flows, but also taking into account the thermal resistance to heat transfer;
– when assessing the efficiency of radiant heat transfer from metal friction elements of brakes, it is necessary to take into account their polished and matt surfaces, while the latter are turbolizers of the washed air flows.

Thus, the essence of complex heat transfer in friction pairs of brake devices is disclosed. Thus, based on research:
– the essence of complex heat exchange in friction pairs of brake devices is revealed;
– it has been established that the thickness of the boundary layer of the metal friction element is determined by taking into account the Biot, Nusselt, and Volchenko criteria; using the last criterion, it is possible to predict the energy load of the surface and subsurface layers of friction pairs;
– it has been confirmed that for surfaces considered "technically smooth", there is a diffuse nature of scattering, that is, the root-mean-square roughness (h) of the surface of a metal element is approximately equal to the wavelength (λ) that passes through this surface;
– it has been established that the magnitude of the relative heat flux is significantly influenced by the increasing value of N, which characterises the relationship between molecular and "photon" thermal conductivity; as the ratio increases, the magnitude of the relative heat flux increases.

Authors' contributions

Conceptualisation: I.K. and D.V.; methodology: D.V.; investigation: Y.R., D.Z., B.D., I.B., V.R., R.H., Y.S. and D.B.; writing – original draft preparation: I.K. and D.V.; writing – review and editing: O.S. and O.B.; supervision: I.K. and D.V.

All authors have read and agreed to the published version of the manuscript.

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ZŁOŻONA WYMIANA CIEPŁA W PARACH CIERNYCH HAMULCÓW

STRESZCZENIE

W artykule rozważane są następujące zagadnienia: ogólne zasady złożonej wymiany ciepła w parach ciernych urządzeń hamulcowych; wymiana ciepła przez promieniowanie z chropowatych powierzchni metalowych elementów ciernych hamulców; połączone działanie pól przewodności cieplnej i promieniowania w metalowych elementach ciernych hamulców. Omówiono rolę termicznych i dynamicznych warstw granicznych wynikających z omywania powietrzem par ciernych z otworami i ich wpływ na chłodzenie. Pokazano nową zależność między kryterium Biota i Nusselta, którą przedstawiono w postaci oporów cieplnych warstw powierzchniowej i przypowierzchniowej obręczy koła tarczowego hamulca. Porównanie połączonego działania pól przewodności cieplnej i promieniowania w metalowych elementach ciernych hamulców wykazało, że efektywność promieniowania jest 1,5–1,7 razy większa niż przewodność cieplna.

Słowa kluczowe: urządzenie hamujące, para cierna, metalowy element cierny, konwekcja, promieniowanie i przewodność cieplna