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ON CONTACT THERMAL RESISTANCE IN THE SYSTEM OF ACTIVE EFFECTIVE COOLING OF THE LOCOMOTIVE DISK BRAKE

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Summary. This paper considers the methods of theoretical estimation of the contact thermal resistance of the elements of the active liquid-air cooling system of the locomotive disk brake. It gives the possibility to carry out estimation of heat exchange in the contact of heterogeneous materials taking into account their microscopic and macroscopic parameters. Its application enables to carry out the necessary thermal contact correction of the integration: brake disk – heat removing element aimed at the provision of optimal functioning of the active cooling system of the locomotive disk brake.

Key words: heat resistance, active cooling, locomotive disk brake.

INTRODUCTION

The growth of brake horse-power of disk brakes is restrained with intensive heat flows generated while braking. It is very likely that one of the perspective ways of this problem solution consists in the use of liquid-air active cooling system of the disk brake [3, 4]. The design of such a system requires the solution of the contact heat-exchange problem between its elements. The most important is integration of the brake disk and the element sending heat to the heat transfer agent which is circulating in the cooling system.

The main purpose of the contact heat-exchange is the determination of interconnection between heat flows and the difference of temperatures on the interacting surfaces. In addition to that the thermal resistance of the contact explained by imperfection of the mechanical joint of integrated surfaces has influence on the heat flows. There fulfilled theoretical estimation of contact thermal resistance indicated over the integration of the elements of the cooling system of the locomotive disk brake.

The value of the contact thermal resistance is

$$R_c = \frac{1}{a_c} = \frac{\Delta T}{q};$$

where: a_c - thermal conductivity of the contact, $Wt/(m^2 \cdot K)$;

ΔT - difference of average temperatures of the contacting surfaces, K;

q – density of heat flow, transferred under the contact heat-exchange, Wt/m^2.

Due to discrete nature of the contact the heat transfer process through the zone of contact surfaces represents rather complicated picture and at present the theoretical solution of the problem remains open. There are some empirical and semi-empirical calculation models, generalizing extensive experimental material and enabling to determine an integral value of thermal resistance of the contact of two bodies. There used the methods of the calculation of heat exchange in the contact of heterogeneous materials taking into consideration their microscopical and macroscopical parameters [1, 8, 10]. These methods are supplemented and corrected for the conditions of functioning the locomotive disk brake system.

CALCULATION MODEL

The equivalent diagram of the thermal contact of the brake disk surfaces and the element for heat abstraction (heat removing element) is shown in fig. 1(a). According to it general thermal resistance of the contact R_c can be represented in such a way:

$$\frac{1}{R_c} = \frac{1}{R_{c1} + R_{c2} + R_{f1} + R_{f2}} + \frac{1}{R_{mc}} + \frac{1}{R_r}; \qquad (1)$$

where: R_{c1}, R_{c2} - thermal resistances, conditioned by contraction of heat flow lines to the spots of actual contact of the surfaces of the brake disk and heat removing element;

R_{f1}, R_{f2} - thermal resistances of oxide films of the surfaces of the brake disk and heat removing element;

R_{mc} - thermal resistance of the medium filling intercontact clearances;

Rr - thermal resistance to the heat flow, transmitted by means of radiation through intercontact clearances.

Fig. 1(a) shows thermal resistances of the materials of integrated surfaces R_{b1} and R_{b2}.

The presence of the components R_{c1}, R_{c2} in (1) is explained by the following. In the homogeneous medium the heat flow line represents a curve (lines 1 in the fig. 1(b)), formed in such a way, that the temperature gradient vector directed from one isothermal surface to the other one (lines 2 in fig. 1(b)), is oriented along the tangent to this line. Under the heat contact of solids the bending of isothermal surfaces takes place. As heat conductivity of contact spots is considerably higher than heat conductivity of the medium which fills the clearances therefore the lines of heat current are contracted to these spots. The isothermal surfaces parallel to each other a long way from the contact zone acquire complicated nature in the area of contact. Density of heat flow near the spots of contact increases greatly and it brings to the increase of temperature gradient in the zone of contact.

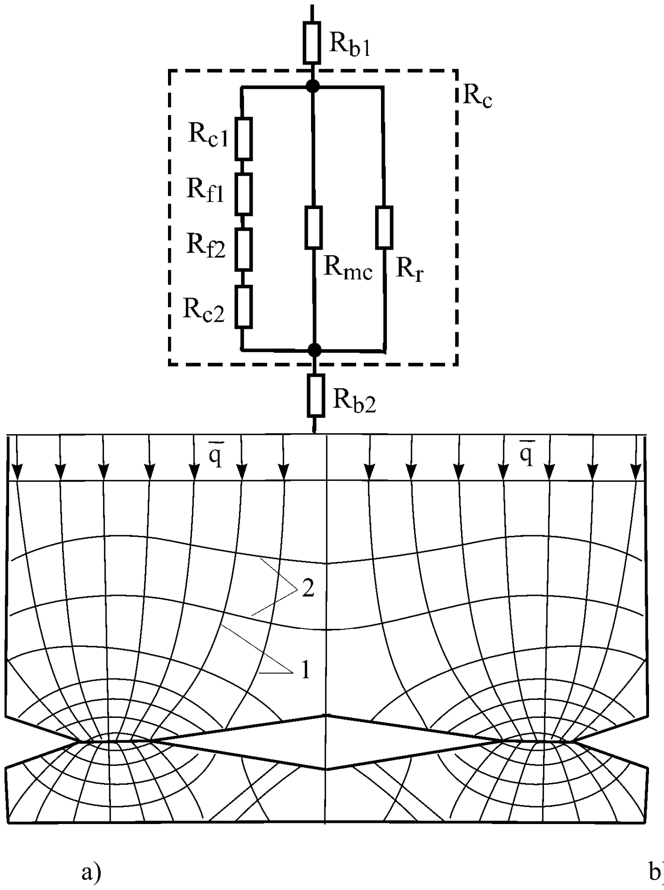

Fig. 1. Equivalent diagram of thermal resistance of the contact of brake disk surfaces and heat removing element (a); diagram of contraction of heat flow lines to the spots of actual contact (b)

Registration in (1) of heat exchange by means of heatconductivity and radiation through gas space in the intercontact clearance (components R_{mc} and R_r) is stipulated by high temperatures in the zone of contact. At that time, limited size of intercontact clearances prevents from the origin of convection flows. This factor enables to neglect convection heat exchange while forming the diagrams of thermal resistance of the contact of the surfaces of a brake disk and a heat removing element.

Use of the given below calculation dependences is defined by the following assumptions:

- Real contact area constitutes insignificant part of nominal area and is formed by the contact spots of average identical size, evenly distributed along the surface of the contact.

- Heat models of rough surfaces, used for obtaining integral dependencies, are represented in the form of the cylinders with spherical contiguous surfaces (each surface has a solitary spot of the contact).
- Thermal resistance of real contact connected in series with each boundary of two mediums, close to ideal, therefore it is excluded from the equivalent diagram (fig. 1(a)) as a sequent component which has a small value (that is, thermal resistance of real contact is defined by thermal resistance of contraction of the heat flow lines to the spots of real contact of the surfaces of a brake disk and a heat removing element).
- Heat resistance of oxide films while obtaining calculation dependences is recognized as a small one, because due to the absence of calculation dependences on the process of film growth, their resistance to compression force and so on, the preference is given to experimental ways of their definition [10] and further correction of calculation values of summary contact resistance.

Summary thermal resistance of contraction can be recognized as [2, 7]:

$$\frac{1}{R_{c1}+R_{c2}} = 1{,}25 \cdot \lambda_c \cdot \left(\frac{P_n}{H}\right)^{0{,}95} \cdot \left(\frac{M}{\sigma}\right); \tag{2}$$

where $\lambda_c = \dfrac{2 \cdot \lambda_1 \cdot \lambda_2}{\lambda_1 + \lambda_2}$ - average geometrical heat conductivity of the materials of interacting surfaces, Wt/(m·K). Here λ_1, λ_2 - coefficients of heat conductivity of the materials of the brake disk and heat removing element correspondingly;

P_n - nominal contact pressure, Pa;

$\sigma = (\sigma_1^2 + \sigma_2^2)^{0{,}5}$ - surface roughness, m. Here σ_1, σ_2 - average height of the protrusions of microunevenness in the area of the contact of the surfaces of a brake disk and a heat removing element correspondingly. Standard parameter of the surface roughness can be used as these values: average arithmetic deviation of microunevenness protrusions (Ra);

$M = (M_1^2 + M_2^2)^{0{,}5}$ - average angle of the microunevenness slope, rad. Here M_1, M_2 - average angle of the slope of microunevenness for integrated surfaces of a brake disk and heat removing element (slope of the generating line of the microunevenness cone);

H - average surface microhardness, Pa. The given value can be found by the ratio:

$$H = \frac{P_n}{\left(\dfrac{P_n \cdot M}{1{,}62 \cdot 10^6 \cdot \sigma \cdot c_1}\right)^{1/(1+0{,}071 \cdot c_2)}};$$

where: c_1, c_2 - Vikkers microhardness coefficients of interacting surfaces of the brake disk and heat removing element correspondingly:

$$c_1 = 3{,}178 \cdot 10^9 \cdot \left(4 - 5{,}77 \cdot k + 4 \cdot k^2 - 0{,}61 \cdot k^3\right);$$

$$c_2 = -0{,}57 + k / 1{,}22 - k^2 / 2{,}42 + k^3 16{,}58;$$

where: $k = 0{,}315 \cdot 10^{-9} \cdot Hb$, Pa. Here Hb – Brinell microhardness computed for each from the surfaces separately.

Thermal resistance of the medium, filling intercontact clearances, is defined in the following way [2]:

$$\frac{1}{R_{mc}} = \frac{\lambda_m}{\Delta + G}; \tag{3}$$

where λ_m - medium heat conductivity in the intercontact clearances, Wt/(m·K);

Δ - distance between average planes of the contacting surfaces, m.

G - gas parameter, m.

The given values can be found in the following way:

$$\Delta = 1{,}53 \cdot \sigma \cdot \left(\frac{P_n}{H}\right)^{-0{,}097};$$

$$G = \frac{2 \cdot \gamma}{Pr \cdot (\gamma + 1)} \cdot \left(\frac{2 - \alpha_1}{\alpha_1} + \frac{2 - \alpha_2}{\alpha_2}\right) \cdot L;$$

where α_1, α_2 - accommodation coefficients on the surface boundaries of a brake disk and a heat removing element with intercontact medium. They represent a value of energy exchange effectiveness on the section surface between gas and a solid, having different temperature.

γ - heat rate (statistics coefficient);

Pr - Prandtl number, determined for the medium in the clearances by average temperature of the integrated surfaces;

L - average molecular run in the clearance at the current temperature and pressure, m.

$$L = L_0 \cdot \frac{P_0}{P} \cdot \frac{T}{T_0};$$

where: L_0 - average molecular run at pressure P_0 and temperature T_0. Under the normal conditions: $P_0 = 1 \cdot 10^5$ Pà and $Ò_0 = 293$ Ê, value $L_0 \approx 5 \cdot 10^{-8}$ m.

Thermal resistance to the heat flow, which is transferred by means of radiation can be found with the help of the ratio [8].

$$\frac{1}{R_r} = \frac{C_0 \cdot \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]}{(1/\varepsilon_1 + 1/\varepsilon_2 - 1) \cdot (T_1 - T_2)}; \tag{4}$$

where: T_1, T_2 - average temperatures of the contacting surfaces of the brake disk and heat removing element correspondingly, K;

$C_0 = 5{,}67$ - radiation coefficient of the black body, $Wt/(m^2 \cdot K^4)$;

ε_1, ε_2 - blackness rate of the surfaces of a brake disk and a heat removing element correspondingly.

As a result, using the ratios (1)-(4) it is possible to estimate the value of contact thermal resistance of friction pair: brake disk – heat removing element. The given estimation is carried out below.

INITIAL DATA AND CALCULATION RESULTS

Contact surfaces are made of alloyed steel (brake disk) and copper (surface layer of the heat-removing element). Nominal area of the contact makes up 0,126 m^2 (a circle with the radius 0,2 m). Interacting surfaces are rough but not wavy. Roughness parameters for brake disk material: $Ra = 2 \cdot 10^{-6}$ m, heat removable element: $Ra = 5 \cdot 10^{-6}$ m. Average angle of the slope of microunevenness constitutes $M \approx tg(M) \sim 0{,}05$. Heat-physical and mechanical characteristics of the materials used in calculations are considered as the temperature function (according to the reference book [5, 9]). The following values (corresponding to the normal conditions) are used as initial values: $\lambda_1 = 35$ Wt/(m·K), $\lambda_2 = 398$ Wt/(m·K), $\lambda_m = 0{,}04$ Wt/(m·K); Pr = 0,7; $Hb_1 = 4{,}0 \cdot 10^9$ Pa, $Hb_2 = 7{,}0 \cdot 10^9$ Pa; $\alpha_1 = 0{,}3$, $\alpha_2 = 0{,}7$; $\gamma = 1$; $\varepsilon_1 = 0{,}5$, $\varepsilon_2 = 0{,}6$.

The results of computations obtained with the help of above mentioned methods are given below in fig. 2.

The dependence of heat contact resistance of the integration: brake disk – heat removable element as the function of nominal contact pressure and the difference of average temperatures of the contacting surfaces is shown in fig. 2(a). The dependence of the heat flow value passing through the zone of contact as the function of heat contact resistance and the difference of average temperatures of contacting surfaces is shown in fig. 2(b). The value 363 K (90 °C) is used as an average temperature of the surface of the heat removing element. This value is conditioned by the fact that water or other liquid having the boiling point about 373 K (100 °C) under normal pressure is circulating in the cooling system. Maximum value of contact difference of temperatures makes up 400 K, minimum – 25 K. Nominal contact pressure fluctuates within 10 – 1000 H.

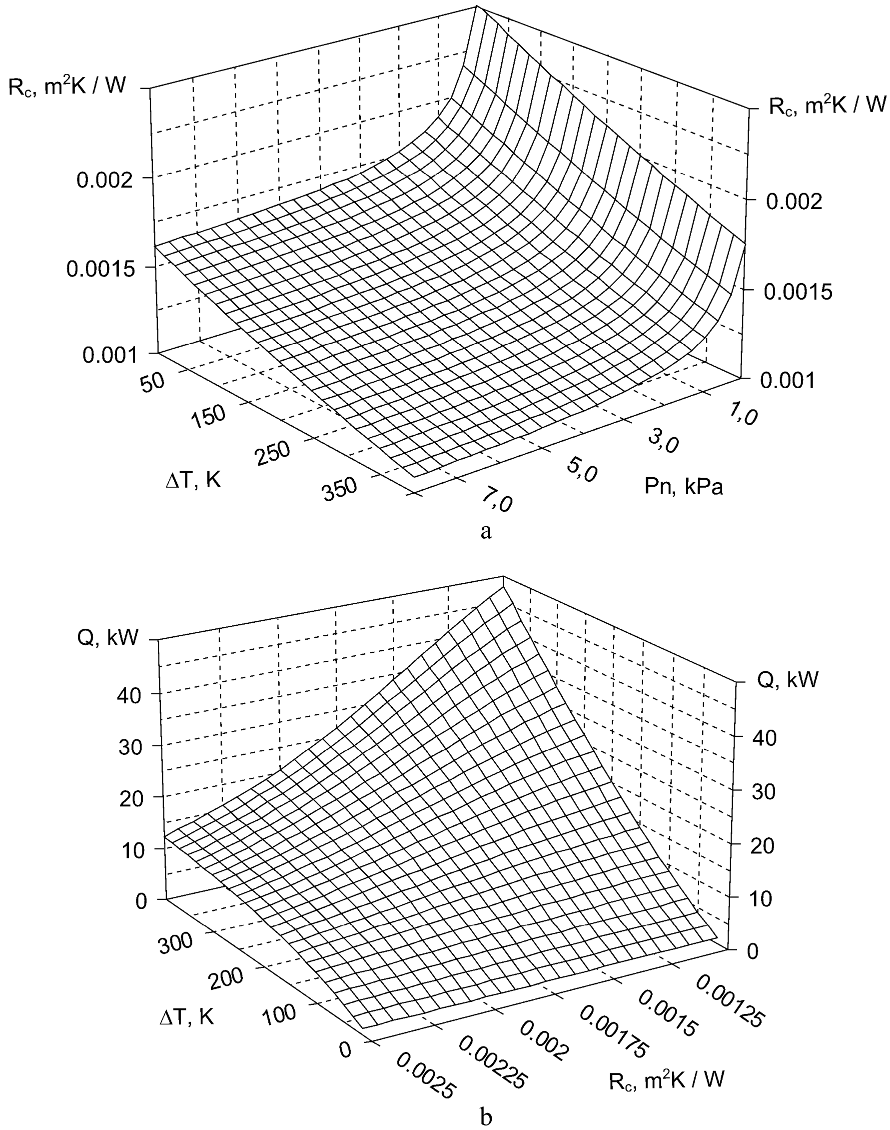

Fig. 2. Dependence of thermal contact resistance on nominal contact pressure and difference of average temperatures of contacting surfaces (a).
Dependence of the heat flow value passing through the zone of contact on thermal contact resistance and difference of average temperatures of contacting surfaces (b).

CONCLUSION

Theoretical estimation gives the possibility to come to the conclusion that for above mentioned conditions contact thermal resistance is found within 0,0011…0,0024 $m^2 \cdot K/Wt$. The heat flow corresponding to these values and passing through integrated surfaces is found within 1,3… 46 kWt.

For functioning the cooling system of the locomotive disk brake it is necessary to maintain heat flow passing through integration: brake disk – heat removing element to provide maximum heat abstraction and not to permit overheat of the heat-transfer agent. It is possible to do it by corresponding regulation of contact thermal resistance by means of clamp force change in the contact or contact temperature difference. So long as the temperature of brake disk is constantly changing, therefore clamp force change of the heat removing element to the brake disk is the simplest way.

According to estimation [3, 6] in the process of the locomotive braking with the axle load of 210 kN as far as full stop at the initial motion speed of 120 km/hour 200 kWt of heat energy is generated by a solitary brake disk. (The time of braking is 33 s during slowing down $1 m/s^2$). To transfer 40 kWt of heat energy through a heat removing element it is necessary to have thermal resistance of the contact about 0,0011…0,0013 $m^2 \cdot K/Wt$. Clamp force within 0,5…1 kN and contact difference of temperatures within 330…350 K correspond to it.

The use of above methods enables to realize necessary correction of thermal contact of integration brake disk – heat removing element aimed at optimal functioning of active cooling system of the locomotive disk brake.

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**О КОНТАКТНОМ ТЕРМИЧЕСКОМ СОПРОТИВЛЕНИИ
В СИСТЕМЕ АКТИВНОГО ОХЛАЖДЕНИЯ ДИСКОВОГО ТОРМОЗА ЛОКОМОТИВА**

Соснов И., Белобородова И., Сергиенко О.

Аннотация. В статье рассмотрена методика теоретической оценки контактного термического сопротивления элементов системы активного жидкостно-воздушного охлаждения дискового тормоза локомотива, позволяющая осуществлять расчет теплообмена в контакте разнородных материалов, с учетом их микроскопических и макроскопических параметров. Ее использование дает возможность проводить необходимую коррекцию термического контакта сопряжения тормозной диск – теплосъемный элемент с целью обеспечения оптимального функционирования системы активного охлаждения дискового тормоза локомотива.

Ключевые слова: термическое сопротивление, активное охлаждение, дисковый тормоз локомотива.