Heat abstraction from contact zone of working elements of disc brake

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Summary. The work presents the results of theoretical and experimental research of influence of abstraction of part of heat power which is generated in friction contact zone on the process of mechanical braking due to the use of the system of forced cooling which is built on the base of recuperative oleopneumatic heat exchange apparatus. The mathematical description of the system of cooling of disk brake is given, checking of its adequacy is performed as well as the efficiency of the use of this cooling system on the basis of bench experimental testing. Recommendations concerning constructive parameters of cooling system necessary for its use at the rolling stock of railways are presented.
Key words: disk brake, friction coefficient, braking effectiveness, cooling of friction contact zone.

INTRODUCTION

Braking tools of modern high speed trains reach the specific power consuming up to 40 MJ [2, 18, 19]. While absorbing such amount of energy during the braking the heating of friction elements is 800-900°C that causes instability of parameters of disk brake and as a result decreases the operating parameters of rolling stock which are connected with the necessity of compliance of prescribed braking distance and train schedules [11, 14, 16, 20].

Natural factor of the impact on the friction coefficient is cooling of operating friction elements of disc brake [1, 12, 15]. Existing constructions implement the principle of cooling of working elements due to the directing air flows to more thermally strained surfaces during the rotation of disk brakes but the effectiveness of such measures is not satisfactory because using them it is possible to outlet a small proportion of heat from the friction zone [9, 10, 17].

More effective is forced cooling of operating elements of friction brake due to the liquid which is a heat carrier and it is able to outlet enough amount of heat (up to 50%) [3] from the friction zone to the environment. This significantly stabilizes the friction coefficient.

DESCRIPTION AND JUSTIFICATION OF USING THE PROPOSED COOLING SYSTEM

The basic scheme [4] of such system is presented at the Fig. 1. Heat that is generated in the contact of brake disk 1 with linings 2 is taken away from the outward surface of brake disk due to the elements for heat removal 3 which have the system of inner canals, the cooling liquid circulates through them. Their contact with the surface of brake disk is provided (and regulated) by the elastic elements 4. The transportation and cooling of liquid is carried out due to heat exchange apparatus 7.

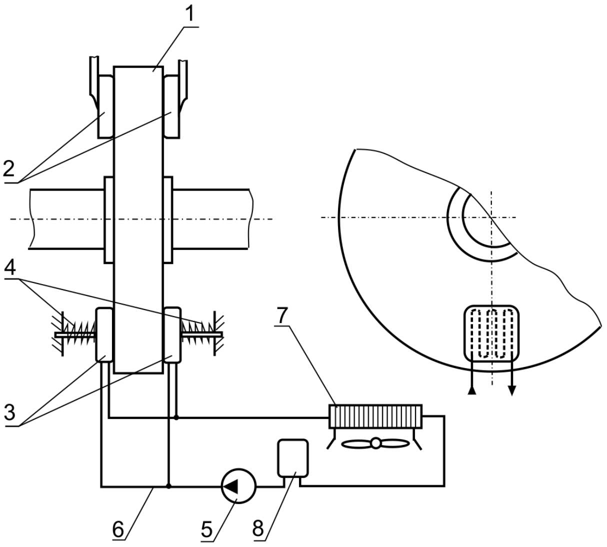


Fig. 1. The scheme of the system of cooling of locomotive disc brake

The mathematical modeling of considered above cooling system is based on the joint mathematical description of following its elements and processes [5, 6]: contact of brace disk with linings (determination of the size of air flow which is generated during the braking); heating and cooling of brake disk material due to the heat conduction as well as convective and radiant heat exchange with the environment; contact heat exchange between brake disk and elements; functioning of heat exchange apparatus which cools the liquid that circulates through the elements for heat removal. The mentioned mathematical models are considered below.

The basis of the model is the equation of Fourier-Kirchhoff (1) in three-dimensional orthogonal coordinate system without internal heat sources considering convectional temperature change (Fig. 2).

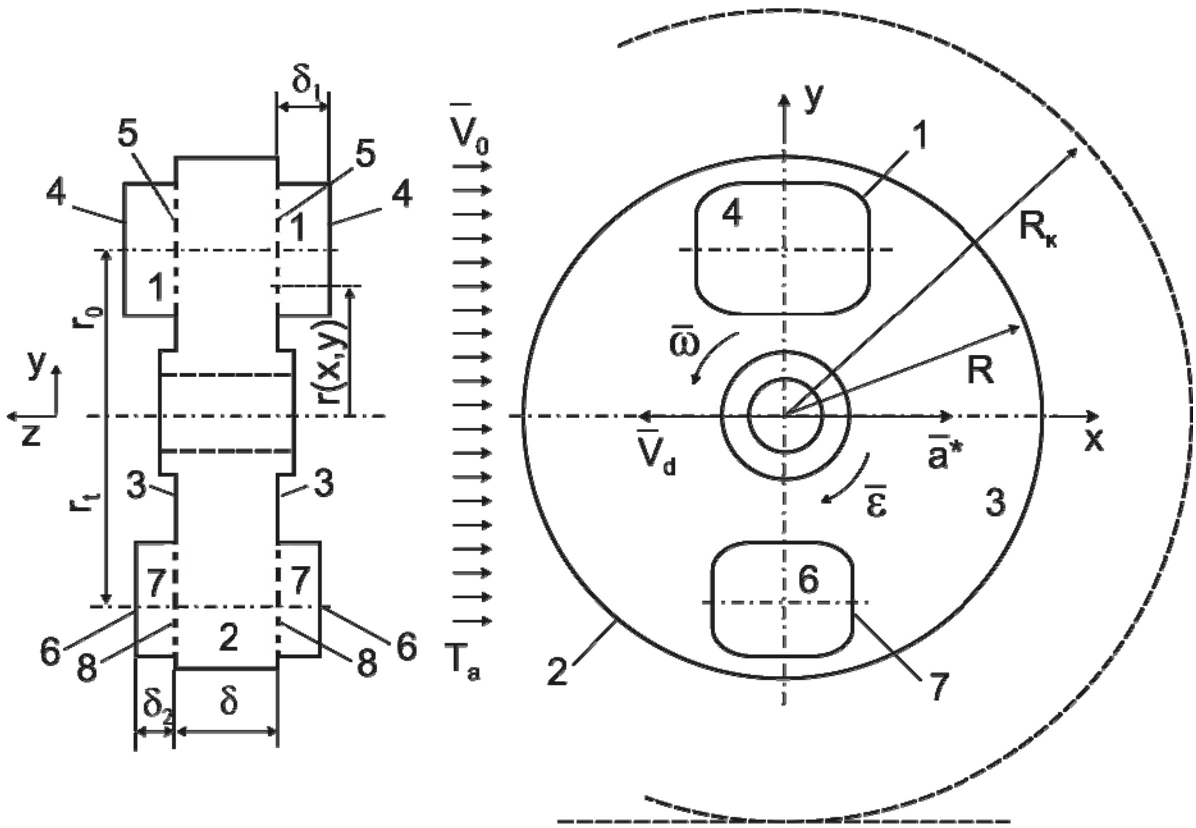


Fig. 2. The estimated scheme of locomotive disc brake with elements of cooling system:
1, 4 – outward surfaces of brake linings,
2, 3 – outward surfaces of brake disk,
6, 7 – outward surfaces of element for heat removal,
5, 8 – surfaces of contact of brake disk with linings and elements for heat removal severally

The model allows determining the temperatures of the considered surfaces of friction brake (both average integral and local) and also heat flows that come through the mentioned surfaces during the braking process:

$$\frac{\partial T}{\partial t} = \frac{1}{\rho \cdot c_{pm}} \cdot \left[\frac{\partial}{\partial x}\left(\lambda \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(\lambda \frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(\lambda \frac{\partial T}{\partial z}\right)\right] - \left(v_x \frac{\partial T}{\partial x} + v_y \frac{\partial T}{\partial y} + v_z \frac{\partial T}{\partial z}\right), \quad (1)$$

where: ρ_m – specific mass,

t – current value of time during braking process,

c_{pm} – specific isobar heat capacity,

λ_m – coefficient of thermal conductivity,

T – absolute temperature (T = f(x, y, z, t),

x, y, z – orthogonal coordinates),

v_x, v_y, v_z – projections of vector v of the linear speed of movement to the point of the outer surface with the coordinates x, y, z on the relevant coordinate axes.

There are following notations on the scheme:

R, $R_к$ – radius of brake disc and of locomotive wheel correspondingly,

r_0, r_t – average radius of friction of lining and of element for heat removal correspondingly,

V_0 – speed of oncoming air,

T_a – average air temperature far from the brake disc,

V_d – current locomotive speed,

$\overline{\varpi}$ – angular speed of rotation of brake disc,

$\bar{a}^*$ – deceleration during braking,

$\bar{\varepsilon}$ – angular acceleration of the disc brake during braking.

Physical and thermophysical characteristics included in the equation (1) belong to the materials of brake disc, linings and elements for heat removal. For surfaces 1 – 4, 6, 7 (Fig. 2) boundary conditions of the 3rd kind (combination of convective and radiant heat exchange without internal heat sources) are used:

$$\lambda_m \left(\frac{\partial T}{\partial n}\right)_s = \alpha \cdot (T_s - T_a) + \varepsilon \cdot \sigma \cdot (T_s^4 - T_a^4), \quad (2)$$

where: λ_m – coefficient of heat conductivity,

α – heat transfer coefficient between the corresponding surface and air,

ε – degree of surface blackness (coefficient of emission),

σ – Stefan-Boltzmann constant,

n – unit vector (normal to the boundary of the study area).

Index «s» indicates that this value is assigned to the outer surface. For the surfaces 5 and 8 (sliding contact zone of brake disc with lining and element for heat removal correspondingly) boundary conditions of the 4th kind with surface heat source are used:

$$\lambda_{m1}\left(\frac{\partial T_1}{\partial n}\right)_s \pm q = \lambda_{m2}\left(\frac{\partial T_2}{\partial n}\right)_s$$

where: q – specific heat flow which is generated at sliding contact of brake disc with lining (sign “+”) or which is removed by the element for heat removal (sign “-”). Here and further index “1” belongs to brake disc, index “2” – to lining or element for heat removal.

$$q(r,t) = \frac{1}{\iint rdS} \cdot \frac{m \cdot R_\varepsilon^2 \cdot \varepsilon}{n_n} \cdot r(x,y) \cdot (\omega^* - \varepsilon \cdot t),$$

where: m – braking mass,

n_n – number of braking linings that are involved in braking,

r – radius vector, that describes the contact zone of disc with lining (element for heat removal),

where x, y – current coordinates of the point on the surface S,

ϖ^* – angular speed of disc rotation that precedes braking ($\varepsilon > 0$),

$t \in [0; t_к]$, $t_к = (V^* - V_к)/a^*$ – time of braking (time interval from the moment of the beginning of braking until reaching the final speed of movement by locomotive).

Here $V_к$ – final speed of the locomotive ($a^* > 0$), V* – the speed of the locomotive that precedes braking. Other surfaces (that do not shown at the Fig. 2) are considered to be thermally insulated.

The most important parameter of convective heart exchange between outward surfaces of elements of friction brake with environment is the coefficient of heat emission that is included to the equation 2. The peculiarity of this case is that forced convection is caused not only by air flow that blows longitudinally at the disc but also by disc rotation [13]. Moreover as the analysis shows these factors make approximately the same impact on the process of forced convection. Therefore the usage of speed of airflow which incidents on brake disc (considering the speed of oncoming wind and the rotation of the disk) as the heat transfer characteristics during the braking process allows considering heat transfer as stages of convective heat exchange which consistently change each other: forced convection, joint action of natural and forced convection, natural convection [7].

In order to determine the coefficient of heat emission α, depending on the speed of the locomotive, the following criteria equations are proposed:

$$\left.\begin{aligned}
&V_d = 0\ldots15\ \text{km/hour}:\\
&Nu = 0{,}4\cdot(Re_\omega^2 + Gr)^{0,25},\\
&V_d = 10\ldots35\ \text{km/hour}:\\
&Nu = 0{,}18\cdot[(0{,}5\cdot Re_\omega^2 + Gr)\cdot Pr]^{0,315},\\
&V_d = 30\ldots57\ \text{km/hour}:\\
&Nu = 0{,}135\cdot[(0{,}5\cdot Re_\omega^2 + Re_a^2 + Gr)\cdot Pr]^{0,33},\\
&V_d = 42\ldots125\ \text{km/hour}:\\
&Nu = 0{,}037\cdot(Re_a^{0,8} + Re_\omega^{0,4})\cdot Pr^{0,33}.
\end{aligned}\right\} \quad (3)$$

Reynolds number which are caused by the rotation of the brake disc (Re_ω) and by its blowing of oncoming air flow (Re_a):

$$Re_\omega = \frac{4\cdot\omega\cdot R^2}{\nu} = 4\cdot\left(\frac{V_d}{R_к}\right)\cdot\frac{R^2}{\nu}, \qquad Re_a = \frac{2\cdot V\cdot R}{\nu}, \quad (4)$$

where: ν – kinematic viscosity of air,

$V = V_d + V_0 = (V^* - a^*\cdot t) + V_0$ – current speed of the air flow which incidents on the brake disc,

Grashof, Prandtl and Nuselta numbers for air:

$$Gr = 8\cdot\beta\cdot g\cdot R^3\cdot(T_s - T_a)/\nu^2;\ Pr = c_p\cdot\mu/\lambda, \qquad Nu = 2\cdot\alpha\cdot R/\lambda, \quad (5)$$

where: β – coefficient of volumetric expansion of air,

λ – coefficient of thermal conductivity of air,

c_p – specific isobar heat capacity of air,

μ – dynamic viscosity of air,

α – surface heat transfer coefficient of brake disk.

For the first equation of the system 3 numbers of similarity Re_ω and Gr are calculated in the following way:

$$Re_\omega = \frac{V_d\cdot R^2}{R_к\cdot\nu_a};\quad Gr = 8\cdot\beta\cdot g\cdot R^3\cdot\pi^{1,5}\cdot(T_s - T_a)/\nu^2.$$

For the other equations of the system 3 specified numbers of similarity are defined by the expressions 4-5.

For modelling the work of recuperative heat exchanger apparatus in the system of active cooling of locomotive disc brake there was used a mathematical model based on the heat balance equation [8]:

$$\left.\begin{aligned}
&Q = \int_0^F k_i\cdot\Delta t_i\cdot dF_i = k\cdot\Delta t\cdot F,\\
&Q = Q_1 = Q_2 + \Delta Q,\\
&Q_1 = W_1\cdot(t_1' - t_1'');\ Q_2 = W_2\cdot(t_2'' - t_2'),\\
&\Delta t = \frac{(t_1' - t_2'') - (t_1'' - t_2')}{\ln\dfrac{t_1' - t_2''}{t_1'' - t_2'}},\\
&\delta t_1 = t_1' - t_1'' = (t_1' - t_2')\cdot Z,\\
&\delta t_2 = t_2'' - t_2' = (t_1' - t_2')\cdot\frac{W_1}{W_2}\cdot Z,\\
&Z = \frac{1 - e^{-(1 - W_1/W_2)\cdot(k\cdot F/W_1)}}{1 - (w_1/W_2)\cdot e^{-(1 - W_1/W_2)\cdot(k\cdot F/W_1)}},
\end{aligned}\right\} \quad (6)$$

where: $Q_{1,2}$ – amount of heat given by hot (cooling liquid) and received by cold heat carrier (ambient air) correspondingly,

ΔQ – heat loss to the environment,

Δt – medium integral temperature difference of heat carriers along the length of the heat exchanger,

t_1', t_1'' – temperatures of hot heat carrier at the inlet and the outlet of the heat exchanger correspondingly,

t_2', t_2'' – temperatures of cold heat carrier at the inlet and the outlet of the heat exchanger correspondingly,

k – total (complete) heat transfer coefficient of the heat exchanger,

F – effective area of heat exchange surface,

$W_{1,2}$ – water equivalent for hot and cold heat carriers correspondingly.

The system of equations 6 is constructed for the case of cross-scheme of heat carriers movement.

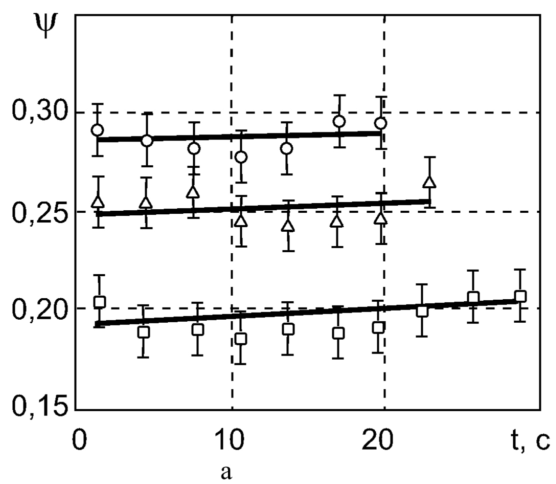
Effect of cooling on braking efficiency was evaluated by experimental determination of friction coefficient and temperature in the friction interaction zone under different operating conditions of the last one.

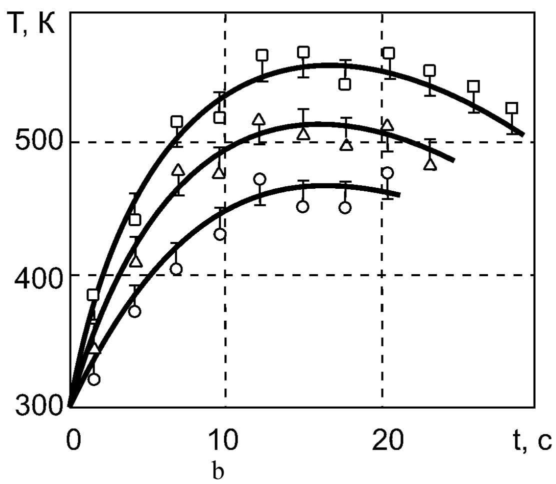
EXPERIMENTAL RESEARCH AND VERIFICATION OF MATHEMATICAL MODEL ADEQUACY

Experimental research of braking process was performed using a laboratory full-scale brake bench. The bench allows to accumulate kinetic energy, to record the frequency of rotation and the duration of the drive work and to register the following input parameters of brake and drive as: braking moment, efforts of braking traction, braking time, temperature of friction surfaces.

The design of the bench provided the item purpose of which is to remove heat from the contact zone of disc and pad. It is a container dimensions of which are caused by the estimated amount of heat that is required to take out from the work area. The container is connected with the heat exchanger. The cooling section of the locomotive 2TE116 (the effective area of heat exchange ≈ 52 m^2) was used as the heat exchanger. It was cooled by means of an axial fan ВОК–4,0 (with a maximum productivity of 4500 m^3/hour). The maximum discharge of heat carrier (water) was 0.9 m^3/hour. The effort of pressing of element for heat removal to friction disc (in this case to the brake pulley with diameter of 200 mm) was controlled with model dynamometer and was in the range 0.05…0.1 kN.

The results of this series of experiments are presented at Fig. 3. Thus Figure 3 (a) shows the nature of the change of the friction-slip coefficient during the braking (t – time during braking), and Fig. 3 (b) shows the average temperatures of contacting friction surfaces.





○ – removal of ~50% of generated heat energy;
△ – removal of ~25% of generated heat energy;
□ – cooling system does not function;
— – theory

Fig. 3. Friction coefficient (a) and average temperature (b) of frictional interacting surfaces during braking

The initial rotational speed of the brake pulley was equivalent of the linear speed of movement of 60 km/hour. The value of the linear deceleration during braking was 1 m/s^2. The experimental values shown on the graph were obtained by averaging a series of parallel measurements. The boundaries of confidence intervals correspond to confidence probability of 0.95. On the graphs shown in Figure 3 (b) only halves of the symmetric confidence intervals are built in order to improve the perception of information. The values of heat energy which is given out by the cooling system (shown on the graphs) were evaluated theoretically using mathematical models considered above. Finite element method using the software package Comsol Multiphysics® was used for solving the differential equations in partial derivatives of the form 1-2. The estimated three-dimensional grid consists of 583247 tetrahedral. The outward surfaces 3 (Fig. 2) contains 48424 triangles, surfaces of contact 5 and 6 contain 2924 and 2108 triangles correspondingly (the maximum size of the element for surfaces 3, 4 and 6 does not exceed 0.005 m).

CONCLUSIONS

1. There was proposed the system of forced liquid-air cooling of working elements of disc brake, the heat carrier is process water. There is the element for heat removal in the construction of the brake. It provides dose removal of heat that generates in the contact zone.
2. In the case of using as a heat carrier technical water or close to it on thermophysical properties liquid with a maximum operating temperature of 363 K which is cooled by the atmosphere air with its initial temperature of 298 K for abstraction of 100 kW of heat energy it is necessary the following surface area of heat exchange:

   direct flow scheme of heat carriers movement: 190...960 m2,

   cross scheme of heat carrier movement: 100…420 m2.
3. It is shown theoretically that the offset from 50 to 100 kW may be provided by liquid-air lamellar heat exchanger with dimensions: from 340×875×925 mm to 950×1325×1145 (effective surface area from 82 to 157 m2, the average coefficient of heat transfer is from 38 to 45 W/(m2·K),) when using a fan with productivity from 4750 to 14500 m3/hour and discharge of hot heat carrier (water) from 1.02 to 1.34 m3/hour.

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ОТВОД ТЕПЛОТЫ С ЗОНЫ КОНТАКТА РАБОЧИХ ЭЛЕМЕНТОВ ДИСКОВОГО ТОРМОЗА

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Аннотация. В работе представлены результаты теоретико-экспериментального исследования влияния на процесс механического торможения отвода части тепловой энергии, генерируемой в зоне фрикционного контакта, за счет использования системы принудительного охлаждения, построенной на базе рекуперативного жидкостно-воздушного теплообменного аппарата. Приведено математическое описание системы охлаждения дискового тормоза, выполнена проверка его адекватности, а также эффективности использования данной системы охлаждения на основе стендовых экспериментальных испытаний. Представлены рекомендации относительно конструктивных параметров системы охлаждения, необходимые для ее использования на подвижном составе железных дорог.

Ключевые слова. дисковый тормоз, коэффициент трения, эффективность торможения, охлаждение зоны фрикционного контакта.