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DETERMINATION OF PRESSURE AND TEMPERATURE ON THE RUBBING SURFACES OF THE DISK BRAKE

Анотація. *Мета статті – для вибраних шляхом математичного моделювання раціональних параметрів основних елементів дискового гальма шахтного локомотива з багатосекторним гальмовим диском обчислити максимальну температуру та визначити найбільший тиск на робочій поверхні. На основі математичного моделювання знайдені максимальна температура і найбільший тиск на робочій поверхні дискового гальма з багатосекторним гальмовим диском. Показано, що максимальна температура на поверхні тертя основних елементів дискового гальма з вибраними параметрами в специфічних шахтних умовах при найбільш несприятливих умовах роботи не перевищить допустиме значення.*

Ключові слова: *фрикційна пара, коефіцієнт зчеплення, дискове гальмо, гальмовий момент, колесо локомотива, рейкова колія.*

Аннотация. *Цель статьи – для выбранных путем математического моделирования рациональных параметров основных элементов дискового тормоза шахтного локомотива с многосекторным тормозным диском вычислить максимальную температуру и определить наибольшее давление на рабочей поверхности. На основе математического моделирования найдены максимальная температура и наибольшее давление на рабочей поверхности дискового тормоза с многосекторным тормозным диском. Показано, что максимальная температура на поверхности трения основных элементов дискового тормоза с выбранными параметрами в специфических шахтных условиях при наиболее неблагоприятных условиях работы не превысит допустимое значение.*

Ключевые слова: *фрикционная пара, коэффициент сцепления, дисковый тормоз, тормозной момент, колесо локомотива, рельсовый путь.*

Abstract. *Article purpose – for the rational parameters of basic elements of a disk brake of the mine locomotive chosen by mathematical modeling with a multisector brake disk to calculate the maximum temperature and to determine the largest pressure on a working surface. On the basis of mathematical modeling the maximum temperature and the largest pressure on a working surface of a disk brake with a multisector brake disk are found. It is shown that the maximum temperature on a surface of friction of basic elements of a disk brake with the chosen parameters in specific mine conditions at the most adverse conditions of work will not exceed admissible value.*

Keywords: *frictional couple, coupling coefficient, disk brake, brake moment, locomotive wheel, railway line.*

Force of adhesion of wheels of the locomotive with rails depends both on a condition of a railway line, and on conditions of interaction of frictional couple a wheel rail [1]. Much attention is paid to research of implementation process of the greatest possible force of adhesion. The key parameter characterizing the force of adhesion of wheels with rails is the coupling coefficient. The brake moment created on a wheel by a wheel and block brake depends on the speed of the movement of the mine locomotive, a condition of a railway line and heating of a brake shoe that does not allow to implement possible coefficient of coupling fully. The disk brakes applied in transport systems have no this shortcoming [2].

The technique of the choice of the constant brake moment enclosed to an axis of wheel couple is given in work [3]. For the purpose of prevention of failure of coupling and the movement of wheels the skid (at the same time force of adhesion sharply falls and flats on wheels are formed) for miner electric locomotives recommends to implement 80% of the greatest possible brake moment.

The strongest impact on reliability of work of a brake mechanism is exerted by temperature condition. Underestimation of the thermal phenomena in brakes of modern machines can lead to a deviation of their performance data from rated and even to accident [2]. In relation to brake mechanisms of mine locomotives safety issues are on the first place. The brake overheat above maximum permissible temperature can become a cause of explosion of methane-air mixture and death of people. Thus, thermal calculation of elements of a brake mechanism of any machine working in mine - one of the most important tasks at its designing.

In the monograph [4] the problem of heating and cooling of a disk brake of mine hoist engines at the coefficient of mutual overlapping of a disk and frictional pads of blocks equal to unit is considered.

Article purpose – for the rational parameters of basic elements of a disk brake of the mine locomotive chosen by mathematical modeling with a multisector brake disk to calculate the maximum temperature and to determine the largest pressure on a working surface.

We will consider the choice of rational parameters of a disk brake with a multisector brake disk on the example of the mine E10 locomotive. Considering design features of an eight-wheel mine electric locomotive of E10, it is reasonable to place a disk brake on a shaft of the engine of each drive cart. It will allow to create two disk brakes the brake moment on all four axes. At placement of disk brakes on axes of four wheel couples their quantity would double. Besides, the necessary brake moment on an axis of wheel couple M_t is significantly more than necessary brake moment on an engine shaft M'_t ($M_t = uM'_t/2$ where u – a gear ratio of a reducer). Therefore, it would lead to increase in the geometrical sizes and a moment of inertia of brake disks, or to increase in their quantity, i.e. would complicate a design of brake system and would increase its cost.

When calculating frictional devices, the friction coefficient is usually considered as a constant, disregarding its dependence from changing in the course of work of temperature, speed and pressure. Take its smallest possible value for the considered frictional couple for a calculated value of coefficient of friction under existing conditions of work [2].

At determination of the geometrical sizes of a brake disk the internal radius of a working zone is chosen minimum admissible for constructive reasons, and external radius as it that during creation of the maximum brake moment pressure in a working zone did not exceed admissible value for considered frictional couple [2].

Let's accept quantity of the sectors of a brake disk made in turn of steel 45 HB 415 and the СЧ 15-32 HB 200 gray cast iron, equal to eight, pads of the brake shoes made of frictional material 6KH-1 (press material of cold formation) [5] - in the form of ring sector with the central corner $\alpha = \pi/4$. Friction coefficients for the specified couples of materials of a disk and frictional pads are respectively equal to 0,535 and 0,41 [2].

Let's define the maximum necessary moment of braking on an engine shaft $M'_{t\,max}$ in the assumption that on the locomotive steel wheels are established. Proceeding from quantity of sectors of a brake disk and a form of frictional pads, we come to a conclusion that dependence of the pulsing braking moment on an engine shaft from the angular coordinate of a shaft of the engine φ_1 can with sufficient degree of accuracy be described by expression

$$M'_t = 2\left(M_0 - A\sin\left(n\varphi_2\right)\right)/u = M'_0 - A'\sin\left(n'\varphi_1\right) =$$
$$= M'_0\left(1 - A^*\sin\left(n'\varphi_1\right)\right) = M'_0\left(1 - \frac{\mu_1 - \mu_2}{\mu_1 + \mu_2}\sin\left(n'\varphi_1\right)\right) \quad \left(\mu_1 > \mu_2\right), \tag{1}$$

where M_0, M'_0 – constant components of the moments of braking respectively on an axis of wheel couple and on an engine shaft; n, n' – numbers of the periods of a sinusoid for one turn according to an axis of wheel couple and a shaft of the engine; φ_2 – angular coordinate of an axis of wheel couple; A, A' – amplitudes of fluctuations of variable components of the moments of braking on an axis of wheel couple and on an engine shaft; $A^* = A'/M'_0$; μ_1, μ_2 – friction coefficients for two couples of materials of a disk and frictional pads.

Let's integrate taking into account a formula (1) system of differential equations [5]:

$$\left(\frac{m_c}{4} - m_3 - m_4\right)\ddot{y} = -\left[C_{y3}\left(y - y_3\right) + \beta_{y3}\left(\dot{y} - \dot{y}_3\right) + C_{y4}\left(y - y_4\right) + \beta_{y4}\left(\dot{y} - \dot{y}_4\right)\right];$$
$$m_3\ddot{y}_3 = C_{y3}\left(y - y_3\right) + \beta_{y3}\left(\dot{y} - \dot{y}_3\right) + F_3\left(S_3\right);$$
$$m_4\ddot{y}_4 = C_{y4}\left(y - y_4\right) + \beta_{y4}\left(\dot{y} - \dot{y}_4\right) + F_4\left(S_4\right);$$
$$I_3\ddot{\varphi}_3 = -\left[C_{\varphi3}\left(\varphi_3 - \varphi_2\right) + \beta_{\varphi3}\left(\dot{\varphi}_3 - \dot{\varphi}_2\right) + rF_3\left(S_3\right)\right];$$
$$I_4\ddot{\varphi}_4 = -\left[C_{\varphi4}\left(\varphi_4 - \varphi_2\right) + \beta_{\varphi4}\left(\dot{\varphi}_4 - \dot{\varphi}_2\right) + rF_4\left(S_4\right)\right];$$
$$I_2\ddot{\varphi}_2 = C_{\varphi3}\left(\varphi_3 - \varphi_2\right) + \beta_{\varphi3}\left(\dot{\varphi}_3 - \dot{\varphi}_2\right) + C_{\varphi4}\left(\varphi_4 - \varphi_2\right) + \beta_{\varphi4}\left(\dot{\varphi}_4 - \dot{\varphi}_2\right) - uM'_t/2,$$

where y, y_3, y_4 – linear movements of the locomotive and corresponding wheels; $\dot{y}$, $\dot{y}_3$, $\dot{y}_4$ – linear speeds; $\ddot{y}$, $\ddot{y}_3$, $\ddot{y}_4$ – linear accelerations; $F_3 = \psi_3\left(S_3\right)m_l g/8$, $F_4 = \psi_4\left(S_4\right)m_l g/8$ – forces of adhesion of the corresponding wheels; $\psi_3 = k_1\left[th\left(k_2 S_3\right) - k_3 S_3 + k_4 S_3^3 \right]$, $\psi_4 = k_1\left[th\left(k_2 S_4\right) - k_3 S_4 + k_4 S_4^3 \right]$ – coefficients of coupling of the corresponding wheels (in the mode of braking accept negative values); k_1, k_2, k_3, k_4 – numerical coefficients of the mechanical characteristic of frictional couple; $S_3 = \left(\dot{\varphi}_3 r - \dot{y}_3\right)/\dot{y}_3$, $S_4 = \left(\dot{\varphi}_4 r - \dot{y}_4\right)/\dot{y}_4$ – relative slidings of the corresponding wheels; $\ddot{\varphi}_2$, $\ddot{\varphi}_3$, $\ddot{\varphi}_4$ – angular accelerations of an output shaft of a reducer and the corresponding wheels; r – radius of a circle of swing of wheels; m_l – mass of the locomotive; g – acceleration of gravity; M'_{t} – the braking moment on an engine shaft.

When calculating we will use geometrical, weight, elastic and dissipative and zhestkostny characteristics of elements of a mine electric locomotive of E10. Let's accept the mass of structure to the equal mass of the locomotive, i.e. $m_c = m_l = 10^4$ kg. Let's set the initial speed of the engine $v_0 = 1$ m/s. Numerical coefficients of the mechanical characteristic of frictional couple k_1, k_2, k_3, k_4 we will take for a case when rails are sanded [6]. Let's receive that failure of coupling in the course of braking will happen at $M'_0 \geq 766$ N · m. Thus, the maximum value of a constant component of the moment of braking on an engine shaft $M'_{0max} = 766$ N · m. The maximum instantaneous value of the necessary brake moment on an engine shaft

$$M'_{\text{t}\,max} = M'_{0max}\left(1 + \frac{\mu_1 - \mu_2}{\mu_1 + \mu_2}\right) \quad (\mu_1 > \mu_2).$$

At the chosen materials of sectors of a brake disk and frictional pads $M'_{\text{t}\,max} = 867$ N · m.

Let's consider a disk brake with one brake disk. For constructive reasons we accept the internal radius of a working zone of a disk $R_1 = 9{,}3 \cdot 10^{-2}$ m. Let's determine the external radius of a working zone of a disk R_2. The maximum pressure upon friction surfaces arising during creation of the maximum brake moment $M'_{\text{t}\,max}$

$$p_{max} = \frac{M'_{\text{t}\,max}}{2\mu_{naim} R_{\text{e}} F} = [p],$$

where μ_{naim} – the smallest possible value of coefficient of friction for couple of materials of a disk and frictional pads under existing conditions works; $R_{\text{e}} = \dfrac{2\left(R_2^3 - R_1^3\right)\alpha}{3\left(R_2^2 - R_1^2\right)\sqrt{2(1 - cos\ \alpha)}}$ – equivalent radius of friction [2]; $F = \alpha\left(R_2^2 - R_1^2\right)/2$ – area of contact of a pad and disk; $[p]$ – the maximum admissible pressure in disk brakes for the considered frictional couple.

After transformations we will receive:

$$R_2 = \sqrt[3]{\frac{6 M'_{\text{t}\,max}\sqrt{2(1 - cos\ \alpha)}}{\mu_{naim}\,\alpha [p] \pi} + R_1^3}.$$

We accept $\mu_{naim} = 0{,}38$ (taking into account the dependences given in work [7]); $[p] = 8{,}29 \cdot 10^5$ N/m^2 [2]. Then $R_2 = 1{,}8 \cdot 10^{-1}$ m.

In work [8] it is shown that the maximum temperature on a surface of friction of a disk reached at the end of braking is stabilized, since the third cycle including braking to a full stop and dispersal. Let's expect temperature surfaces of friction of a brake disk at the end of the third braking to a full stop. We will determine dimensionless temperature on a friction surface in the course of heating by a formula [8]

$$\theta_{1,\,2}(\rho,\,0,\,Fo) = \frac{2\pi Bi_{1,\,2}}{Bi_{1,\,2}^2 + 1} \sum_{n=1}^{\infty} \frac{V_{0\,1,\,2}(\nu_n \rho)\left(2 + \pi\rho_1 V_{0\,1,\,2}(\rho_1 \nu_n)\right)}{\nu_n\left(4 - \pi^2\rho_1^2 V_{0\,1,\,2}(\rho_1 \nu_n)\right)} \times$$

$$\times \int_0^{Fo} Ki(Fo-\tau)\,\varphi_{1,2}(v_n,\tau)\,d\tau,$$

where $\theta_{1,2}=(T_{1,2}-T_n)/(T_d-T_n)$ – dimensionless temperature (hereinafter the index 1 belongs to a disk, 2 – to frictional pads); $T_{1,2}$ – temperature; T_n – reference temperature of a disk and pads; T_d – admissible temperature on a friction surface; $\rho=r/R_2$; $\rho_1=R_1/R_2$; $Fo=a_1t/R_2^2$ – Fourier's criterion (dimensionless time); $a_{1,2}=\lambda_{1,2}/c_{1,2}\gamma_{1,2}$ – coefficients of heat diffusivity of a disk and frictional pads respectively; $\lambda_{1,2}$ – heat conductivity coefficients; $c_{1,2}$ – specific heat capacities; $\gamma_{1,2}$ – density; t – time; $Bi_{1,2}=\sigma_{1,2}R_2/\lambda_{1,2}$ – Biot's criterion; $\sigma_{1,2}$ – heat emission coefficients; $V_{01,2}(v_n\rho)=(Bi_{1,2}Y_0(v_n)-v_nY_1(v_n))J_0(v_n\rho)+(v_nJ_1(v_n)-Bi_{1,2}J_0(v_n))\,Y_0(v_n\rho)$ – a kernel of final integral transformation of Hankel on a variable ρ; Y_0, Y_1, J_0, J_1 – Bessel functions; v_n – the eigenvalues defined from the equation

$$\left(v_nJ_1(v_n\rho_1)+Bi_{1,2}J_0(v_n\rho_1)\right)\left(v_nY_1(v_n)-Bi_{1,2}Y_0(v_n)\right)-$$
$$-\left(v_nJ_1(v_n)-Bi_{1,2}J_0(v_n)\right)\left(v_nY_1(v_n\rho_1)+Bi_{1,2}Y_0(v_n\rho_1)\right)=0;$$

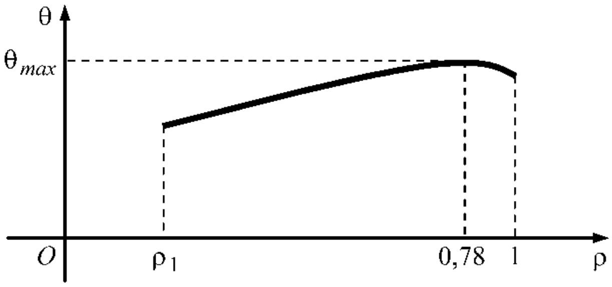
$Ki=q(t)\,R_2/(T_d-T_n)\lambda_1$ – Kirpichev's criterion; $q(t)=\frac{M_t'\,\omega_n}{t_t\,F}\int_0^t\left(1-\frac{\tau}{t_t}\right)d\tau$ – thermal flow; ω_n – the angular speed of a disk in an initial instant; t_t – braking time;

$$\varphi_1=\alpha_{tp}\,\kappa e^{-v_n^2Fo}\left(\frac{1}{\sqrt{\pi Fo}}-(1-\kappa)\,Bi_1e^{\kappa^2Bi_1^2Fo}erfc\left((1-\kappa)Bi_1\sqrt{Fo}\right)\right);\ \varphi_2=\frac{(1-\alpha_{tp})\sqrt{a}\;e^{-a\,v_n^2Fo}}{\lambda\sqrt{\pi Fo}};$$

$\alpha_{tp}=\sqrt{\lambda_1c_1\gamma_1}/\left(\sqrt{\lambda_1c_1\gamma_1}+\sqrt{\lambda_2c_2\gamma_2}\right)$ – the coefficient of distribution of thermal flows showing what part of heat generated at friction is taken away in a brake disk; $\kappa=\alpha/2\pi$, $erfc\,x=\frac{2}{\sqrt{\pi}}\int_x^\infty e^{-\tau^2}d\tau=1-erf\,x$;

$erf\,x=\frac{2}{\sqrt{\pi}}\int_0^x e^{-\tau^2}d\tau$ – integral of probabilities $a=a_2/a_1$; $\lambda=\lambda_2/\lambda_1$.

Apparently from drawing, in a final instant dependence of dimensionless temperature on a surface of friction of a brake disk from dimensionless radius has a maximum in a point $\rho_0=0{,}78$.



Dependence of dimensionless temperature on a surface brake disk from dimensionless radius at the end of braking

During cooling we will determine dimensionless temperature on a surface of friction from a ratio which conclusion is similar provided in the monograph [4]:

$$\theta_{1,\,2}\left(\rho,\,0,\,Fo\right)=\frac{2\,\pi^{2}}{Bi_{1,\,2}^{2}+1}\sum_{n=1}^{\infty}\frac{V_{0\,1,\,2}\left(v_{n}\rho\right)}{4-\pi^{2}\rho_{1}^{2}\,V_{0\,1,\,2}\left(v_{n}\rho_{1}\right)}\int_{0}^{Fo}\eta_{1,\,2}\left(v_{n},\,\tau\right)d\tau+U_{1,\,2}\,,$$

$$\eta_{1,\,2}=c_{1,\,2}\;exp\;\left[\;-\left(d_{1,\,2}-Bi_{1,\,2}^{2}\right)\right]\left(\frac{exp\;\left(-d_{1,\,2}Fo\right)}{\sqrt{\pi Fo}}-\sqrt{d_{1,\,2}}\,exp\;\left(d_{1,\,2}Fo\right)-1\right),$$

where $c_1=U_1\,Bi_1$; $c_2=\sqrt{a}\;U_2\,Bi_2$; $U_{1,\,2}=\left(T_{\text{k}\,1,\,2}-T_{\text{n}}\right)\big/\left(T_{\text{d}}-T_{\text{n}}\right)$. At repeated heating instead of T_{n} it is necessary to substitute the maximum temperature on a surface of friction of a disk at the end of the cooling period. $T_{\text{k}\,1,\,2}$ – the maximum temperature at the end of the heating period on a friction surface; $d_1=v_n^2$; $d_2=a\,v_n^2$. Temperature is defined from a ratio

$$T_{1,\,2}=\theta_{1,\,2}\left(T_{\text{d}}-T_{n}\right)+T_{\text{n}}\;.$$

Calculation is feasible in the assumption that the disk is not broken into sectors and is made either of steel 45 HB 415, or of the СЧ 15-32 HB 200 gray cast iron at the following input datas: $\omega_{\text{n}}=201{,}39$ rad/s (corresponds to the linear speed of the engine $\dot{y}=5$ m/s); braking time $t_{\text{t}}=21$ s; dispersal time $t_{\text{p}}=29$ s; $T_{\text{n}}=25$ °C; $T_{d}=240$ °C. For the disk made of steel 45 HB 415, $a_1=1{,}3\cdot10^{-5}$ m^2/s; $a_2=6{,}7\cdot10^{-8}$ m^2/s; $\lambda_1=4{,}5\cdot10^{1}$ Wt/(м·°C); $\lambda_2=5{,}1\cdot10^{-1}$ Wt/(м·°C); $c_1=461$ J/(kg·°C); $c_2=963$ J/(kg·°C); $\sigma_1=44$ Wt/(м·°C); $\sigma_2=8$ Wt/m·°C. For the disk made of the СЧ 15-32 HB 200 gray cast iron, $a_1=1{,}7\cdot10^{-5}$ m^2/s; $\lambda_1=6{,}3\cdot10^{1}$ Wt/(м·°C); $c_1=502$ J/(kg·°C); $\sigma_1=44$ Wt/(м·°C). Then the maximum temperature at the end of the third braking on a surface of friction of a brake disk from steel 45 HB 415 $T_1=198$ °C, and on a surface of friction of a brake disk from the СЧ 15-32 HB 200 gray cast iron – $T_1=206$ °C. Thus, taking into account the dependences given in work [7] in specific mine conditions at the end of the third braking to a full stop the maximum temperature on a surface of friction of a multisector disk will not exceed admissible value.

Conclusions

1. On the basis of mathematical modeling of rational parameters of basic elements of a disk brake of the mine locomotive with a multisector brake disk the maximum temperature and the largest pressure on its working surface are determined.
2. It is established that at the chosen parameters of a disk brake with a multisector disk in specific mine conditions at the end of the third braking to a full stop the maximum temperature on a surface of friction will make no more than 206 °C, i.e. will not exceed admissible value.

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