

ОБЧИСЛЕННЯ НАЙБІЛЬШОГО ТИСКУ ТА КООРДИНАТ МАКСИМАЛЬНОЇ ТЕМПЕРАТУРИ НА РОБОЧІЙ ПОВЕРХНІ ДИСКОВОГО ГАЛЬМА

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CALCULATION OF THE HIGHEST PRESSURE AND COORDINATES OF A MAXIMUM TEMPERATURE ON A DISC BRAKE WORKING SURFACE

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Мета. Для вибраних шляхом математичного моделювання раціональних параметрів основних елементів дискового гальма шахтного локомотива з багатосекторним гальмовим диском обчислити координати максимальної температури та найбільший тиск на робочій поверхні.

Методика дослідження. Для знаходження координат максимальної температури і найбільшого тиску на робочій поверхні дискового гальма з багатосекторним гальмовим диском при вибраних раціональних параметрах було проведено математичне моделювання температури і тиску на поверхні тертя.

Результати дослідження. На основі математичного моделювання знайдені максимальна температура та її координати 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.

Наукова новизна. Розроблено математичну модель гальмування шахтного локомотива дисковим гальмом, що створює на осі колісної пари пульсуючий гальмовий момент, який залежить від її кутової координати, з урахуванням нелінійної залежності коефіцієнта зчеплення від відносного ковзання, на базі якої встановлені параметри гальмового моменту, що дозволяють поліпшити гальмові характеристики.

Практичне значення. Розроблено науково обґрунтовану інженерну методику вибору раціональних параметрів дискового гальма шахтного локомотива та визначення динамічних і кінематичних характеристик привода шахтного локомотива при гальмуванні дисковим гальмом із багатосекторним диском. Отримано аналітичний розв'язок задачі нестационарної теплопровідності про знаходження температурного поля, що виникає в гальмовому диску та фрикційних накладках дискового гальма шахтного локомотива при виконанні накладок у вигляді кільцевого сектора, на підставі якого знайдена залежність відносної температури на поверхні тертя гальмового диска від часу при циклічному гальмуванні.

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

Introduction. 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 overheating 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.

Problem Statement. Calculate the coordinates of the maximum temperature and maximum pressure on the working surface for the rational parameters of the main elements of the disc brake of a mine locomotive with a multi-sector brake disc selected by mathematical modeling.

Solving of the Problem. Let's 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'_{tmax} 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

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

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]:

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

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; $F_3 = \psi_3(S_3)m_l g/8, F_4 = \psi_4(S_4)m_l g/8$ – forces of adhesion of the corresponding wheels; $\psi_3 = k_1 \left[th(k_2 S_3) - k_3 S_3 + k_4 S_3^3 \right], \psi_4 = k_1 \left[th(k_2 S_4) - k_3 S_4 + k_4 S_4^3 \right]$ – 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; $S_3 = (\dot{\phi}_3 r - \dot{y}_3)/\dot{y}_3, S_4 = (\dot{\phi}_4 r - \dot{y}_4)/\dot{y}_4$ – relative slidings of the corresponding wheels; $\ddot{\phi}_2, \ddot{\phi}_3, \ddot{\phi}_4$ – 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 rigid 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

$$M'_{tmax} = M'_{0max} \left(1 + \frac{\mu_1 - \mu_2}{\mu_1 + \mu_2} \right) \quad (\mu_1 > \mu_2).$$

At the chosen materials of sectors of a brake disk and frictional pads $M'_{tmax} = 867$ 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'_{tmax}

$$p_{max} = \frac{M'_{tmax}}{2\mu_{naim} R_e F} = [p],$$

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; $R_e = \frac{2(R_2^3 - R_1^3)\alpha}{3(R_2^2 - R_1^2)\sqrt{2(1 - \cos \alpha)}}$ – equivalent radius of friction [2];

$F = \alpha(R_2^2 - R_1^2)/2$ – 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:

$$R_2 = \sqrt[3]{\frac{6 M'_{tmax} \sqrt{2(1 - \cos \alpha)}}{\mu_{naim} \alpha [p] \pi} + R_1^3}.$$

We accept $\mu_{naim} = 0,38$ (taking into account the dependences given in work [7]); $[p] = 8,29 \cdot 10^5$ N/m² [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]

$$\theta_{1,2}(\rho, 0, Fo) = \frac{2\pi Bi_{1,2}}{Bi_{1,2}^2 + 1} \sum_{n=1}^{\infty} \frac{V_{0,1,2}(v_n \rho) (2 + \pi \rho_1 V_{0,1,2}(\rho_1 v_n))}{v_n (4 - \pi^2 \rho_1^2 V_{0,1,2}(\rho_1 v_n))} \times \\ \times \int_0^{Fo} Ki(Fo - \tau) \varphi_{1,2}(v_n, \tau) d\tau,$$

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; $\rho = r/R_2$; $\rho_1 = R_1/R_2$; $Fo = a_1 t / R_2^2$ – 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; $V_{01,2}(v_n \rho) = (Bi_{1,2} Y_0(v_n) - v_n Y_1(v_n)) J_0(v_n \rho) + (v_n J_1(v_n) - Bi_{1,2} J_0(v_n)) Y_0(v_n \rho)$ – 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

$$(v_n J_1(v_n \rho_1) + Bi_{1,2} J_0(v_n \rho_1)) (v_n Y_1(v_n) - Bi_{1,2} Y_0(v_n)) - \\ - (v_n J_1(v_n) - Bi_{1,2} J_0(v_n)) (v_n Y_1(v_n \rho_1) + Bi_{1,2} Y_0(v_n \rho_1)) = 0;$$

$Ki = q(t) R_2 / (T_d - T_n) \lambda_1$ – 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;

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

$\alpha_{tp} = \sqrt{\lambda_1 c_1 \gamma_1} / (\sqrt{\lambda_1 c_1 \gamma_1} + \sqrt{\lambda_2 c_2 \gamma_2})$ – the coefficient of distribution of thermal flows showing what part of heat generated at friction is taken away in a brake disk; $\kappa = \alpha / 2\pi$, $\operatorname{erfc} x = \frac{2}{\sqrt{\pi}} \int_x^{\infty} e^{-\tau^2} d\tau = 1 - \operatorname{erf} x$;

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

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

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

$$\theta_{1,2}(\rho, 0, Fo) = \frac{2\pi^2}{Bi_{1,2}^2 + 1} \sum_{n=1}^{\infty} \frac{V_{0,1,2}(v_n \rho)}{4 - \pi^2 \rho_1^2 V_{0,1,2}(v_n \rho_1)} \int_0^{Fo} \eta_{1,2}(v_n, \tau) d\tau + U_{1,2}, \\ \eta_{1,2} = c_{1,2} \exp \left[- (d_{1,2} - Bi_{1,2}^2) \right] \left(\frac{\exp(-d_{1,2} Fo)}{\sqrt{\pi Fo}} - \sqrt{d_{1,2}} \exp(d_{1,2} Fo) - 1 \right),$$

where $c_1 = U_1 Bi_1$; $c_2 = \sqrt{a} U_2 Bi_2$; $U_{1,2} = (T_{k1,2} - T_n) / (T_d - T_n)$. 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. $T_{k1,2}$ – 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

$$T_{1,2} = \theta_{1,2} (T_d - T_n) + T_n.$$

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: $\omega_n = 201,39$ rad/s (corresponds to the linear speed of the engine $\dot{y} = 5$ m/s); braking time $t_t = 21$ s; dispersal time $t_p = 29$ s; $T_n = 25$ °C; $T_d = 240$ °C. For the disk made of steel 45 HB 415, $a_1 = 1,3 \cdot 10^{-5}$ m²/s; $a_2 = 6,7 \cdot 10^{-8}$ m²/s; $\lambda_1 = 4,5 \cdot 10^1$ Wt/(m·°C); $\lambda_2 = 5,1 \cdot 10^{-1}$ Wt/(m·°C); $c_1 = 461$ J/(kg·°C); $c_2 = 963$ J/(kg·°C); $\sigma_1 = 44$ Wt/(m·°C); $\sigma_2 = 8$ Wt/m·°C. For the disk made of the СЧ 15-32 HB 200 gray cast iron, $a_1 = 1,7 \cdot 10^{-5}$ m²/s; $\lambda_1 = 6,3 \cdot 10^1$ Wt/(m·°C); $c_1 = 502$ J/(kg·°C); $\sigma_1 = 44$ Wt/(m·°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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АННОТАЦИЯ

Цель. Для выбранных путем математического моделирования рациональных параметров основных элементов дискового тормоза шахтного локомотива с многосекторным тормозным диском вычислить координаты максимальной температуры и наибольшее давление на рабочей поверхности.

Методика исследования. Для нахождения координат максимальной температуры и наибольшего давления на рабочей поверхности дискового тормоза с многосекторным тормозным диском при выбранных оптимальных параметрах было проведено математическое моделирование температуры и давления на поверхности трения.

Результаты исследования. На основе математического моделирования найдены максимальная температура и её координаты 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.

Научная новизна. Разработана математическая модель торможения шахтного локомотива дисковым тормозом, создающим на оси колесной пары пульсирующий тормозной момент, который зависит от ее угловой координаты, с учетом нелинейной зависимости коэффициента сцепления от относительного скольжения, на базе которой установлены параметры тормозного момента, позволяющие улучшить тормозные характеристики.

Практическое значение. Разработана научно обоснованная инженерная методика выбора рациональных параметров дискового тормоза шахтного локомотива и определения динамических и кинематических характеристик привода шахтного локомотива при торможении дисковым тормозом с многосекторным диском. Получено аналитическое решение задачи нестационарной теплопроводности о нахождении температурного поля, возникающего в тормозном диске и фрикционных накладках дискового тормоза шахтного локомотива при выполнении накладок в виде кольцевого сектора, на основании которого найдена зависимость относительной температуры на поверхности трения тормозного диска от времени при циклическом торможении.

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

ABSTRACT

Objective. Calculate the coordinates of the maximum temperature and the maximum pressure on the working surface for the rational parameters of the main elements of the disc brake of a mine locomotive with a multi-sector brake disc selected by mathematical modeling.

Research Methodology. To find the coordinates of the maximum temperature and maximum pressure on the working surface of a disc brake with a multi-sector brake disc at the selected optimal parameters, mathematical modeling of temperature and pressure on the friction surface was carried out.

Research Results. On the basis of mathematical modeling, the maximum temperature and its coordinates and the highest pressure on the working surface of a disc brake with a multi-sector brake disc are found. It is shown that the maximum temperature on the friction surface of the main elements of the disc brake with the selected parameters in specific mine conditions under the most unfavorable operating conditions will not exceed the permissible value.

Scientific Novelty. A mathematical model has been developed for braking a mine locomotive with a disc brake, which creates a pulsating braking moment on the wheelset axis, which depends on its angular coordinate, taking into account the nonlinear dependence of the adhesion coefficient on relative slip, on the basis of which the parameters of the braking torque are specifications.

Practical Significance. A scientifically substantiated engineering method has been developed for selecting rational parameters of a disc brake for a mine locomotive and determining the dynamic and kinematic characteristics of a mine locomotive drive when braking with a disc brake with a multi-sector disc. An analytical solution to the problem of unsteady thermal conductivity is obtained on finding the temperature field arising in the brake disc and friction linings of a disc brake of a mine locomotive when linings are made in the form of an annular sector, on the basis of which the dependence of the relative temperature on the friction surface of the brake disc on time is found. with cyclic braking.

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

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