ГІРНИЧА ЕЛЕКТРОМЕХАНІКА

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LONG BRAKING OF THE MINE LOCOMOTIVE ON BIAS AT THE PULSING BRAKE MOMENT

Анотація. *Ціль статті – розробити математичну модель і дослідити процес гальмування шахтного локомотива дисковим гальмом на затяжному уклоні при пульсуючому гальмовому моменту. На основі розробленої математичної моделі гальмування шахтного локомотива на затяжному уклоні дисковим гальмом, що створює на осі колісної пари пульсуючий синусоїдальний гальмовий момент, установлено максимальне абсолютне значення уклону колії, при якому температура робочої поверхні гальмового диска не перевищить припустиме значення під час затяжного спуску. Виконано порівняльний аналіз гальмування шахтного локомотива на затяжному уклоні дисковим гальмом, що створює на осі колісної пари постійний і пульсуючий синусоїдальний гальмові моменти.*
Ключові слова: *фрикційна пара, коефіцієнт зчеплення, дискове гальмо, гальмовий момент, колісна пара, рейкова колія.*

Аннотация. *Цель статьи – разработать математическую модель и исследовать процесс торможения шахтного локомотива дисковым тормозом на затяжном уклоне при пульсирующем тормозном моменте. На основе разработанной математической модели торможения шахтного локомотива на затяжном уклоне дисковым тормозом, создающим на оси колесной пары пульсирующий синусоидальный тормозной момент, установлено максимальное абсолютное значение уклона пути, при котором температура рабочей поверхности тормозного диска не превысит допустимое значение во время затяжного спуска. Выполнен сравнительный анализ торможения шахтного локомотива на затяжном уклоне дисковым тормозом, создающим на оси колесной пары постоянный и пульсирующий синусоидальный тормозные моменты.*
Ключевые слова: *фрикционная пара, коэффициент сцепления, дисковый тормоз, тормозной момент, колесная пара, рельсовый путь.*

Abstract. *Article purpose - to develop mathematical model and to investigate process of braking of the mine locomotive a disk brake on a long bias at the pulsing brake moment. On the basis of the developed mathematical model of braking of the mine locomotive on a long bias the disk brake creating the pulsing sinusoidal brake moment on an axis of wheel couple established the maximum absolute value of a bias of a way at which temperature of a working surface of a brake disk will not exceed admissible value during long descent. Comparative analysis of braking of the mine locomotive on a long bias is made by the disk brake creating the constant and pulsing sinusoidal brake moments on an axis of wheel couple.*
Keywords: *frictional couple, coupling coefficient, disk brake, brake moment, wheel couple, railway line*

The implementable force of braking and the loudspeaker of the drive of the mine locomotive when braking are defined first of all by coupling of wheels with rails. Brake force under the influence of the static and dynamic factors taking place when braking the locomotive has a statistical property and results from frictional interaction of a wheel and a rail [1].

From the moment of emergence of the first locomotives the numerous researches directed to studying of coupling of a wheel with a rail as the physical phenomenon were conducted, hypotheses of forming of force of adhesion in different operational conditions were entered. Taking into account influence on the size of force of adhesion of a condition of surfaces of frictional couple, physical properties of materials, speeds of the movement, geometry of a bandage of wheels and a profile of a way, normal loading were defined mean values of coefficient of coupling which were used for operational calculations. Influence of parameters of a running gear and properties of the suspender of the mine locomotive on the force of adhesion and brake characteristics was studied in detail [2].

In work [3] it is offered to improve traction and brake characteristics of the mine locomotive due to application of the elastic axle-box node including rubber-metal elements. In work [4] the mathematical model of braking of the mine locomotive a disk brake on a straight horizontal section of a railway line describing forced oscillations of elements of the wheel and motor block taking into account the nonlinear characteristic of interaction of frictional couple a wheel rail is developed and approved. In work [5] research of process of braking of the mine locomotive on a horizontal railway line by the disk brake creating the pulsing brake moment on an axis of wheel couple for the purpose of implementation of the greatest possible coefficient of coupling of wheels with rails is described, recommendations about the analytical choice of the brake moment for different conditions of a railway line are made, constructive conceptual solutions on production of the disk brake creating the pulsing brake moment are proposed.

Article purpose – to develop mathematical model and to investigate process of braking of the mine locomotive a disk brake on a long bias at the pulsing brake moment.

Forced oscillations of the wheel and motor block of the mine locomotive in the course of braking on the straight section of a railway line having a bias a disk brake taking into account the nonlinear characteristic of interaction of frictional couple a wheel rail can be described by system of six differential equations of the second order

$$\left.\begin{aligned}
&(m_c/4-m_3-m_4)\ddot{y}=-\left[C_{y3}(y-y_3)+\beta_{y3}(\dot{y}-\dot{y}_3)+\right.\\
&\qquad\left.+C_{y4}(y-y_4)+\beta_{y4}(\dot{y}-\dot{y}_4)+(m_c/4-m_3-m_4)g\sin\beta\right],\\
&m_3\ddot{y}_3=C_{y3}(y-y_3)+\beta_{y3}(\dot{y}-\dot{y}_3)+F_3(S_3)-m_3 g\sin\beta\ ,\\
&m_4\ddot{y}_4=C_{y4}(y-y_4)+\beta_{y4}(\dot{y}-\dot{y}_4)+F_4(S_4)-m_4 g\sin\beta\ ,\\
&I_3\ddot{\varphi}_3=-\left[C_{\varphi3}(\varphi_3-\varphi_2)+\beta_{\varphi3}(\dot{\varphi}_3-\dot{\varphi}_2)+rF_3(S_3)\right]\ ,\\
&I_4\ddot{\varphi}_4=-\left[C_{\varphi4}(\varphi_4-\varphi_2)+\beta_{\varphi4}(\dot{\varphi}_4-\dot{\varphi}_2)+rF_4(S_4)\right]\ ,\\
&I_2\ddot{\varphi}_2=C_{\varphi3}(\varphi_3-\varphi_2)+\beta_{\varphi3}(\dot{\varphi}_3-\dot{\varphi}_2)+C_{\varphi4}(\varphi_4-\varphi_2)+\\
&\qquad+\beta_{\varphi4}(\dot{\varphi}_4-\dot{\varphi}_2)-u M'_t/2\ ,
\end{aligned}\right\}\tag{1}$$

where m_c – mass of structure; m_3, m_4 – the specified mass of the corresponding wheels; 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; C_{y3}, C_{y4}, – coefficients of rigidity of the corresponding elastic elements; β_{y3}, β_{y4} – coefficients of viscous internal resistance of the corresponding elastic elements; $C_{\varphi3}$, $C_{\varphi4}$, – coefficients of rigidity of the corresponding half shafts of the wheel and motor block; $\beta_{\varphi3}$, $\beta_{\varphi4}$ – coefficients of viscous internal resistance of the corresponding half shafts of the wheel and motor block; β – a tilt angle of a way (positive at the movement on rise and negative at the movement on descent); $F_3=\psi_3(S_3)(m_l g/8)\cos\beta$, $F_4=\psi_4(S_4)(m_l g/8)\cos\beta$ – forces of adhesion of the corresponding wheels; ψ_3, ψ_4 – coefficients of coupling of the corresponding wheels; S_3, S_4 – relative slidings of the corresponding wheels; m_l – mass of the locomotive; g – acceleration of gravity; I_2 – the given moment of inertia of a reducer, a disk brake and the engine concerning an axis of wheel couple corresponding to one wheel couple (depends on the location of a disk brake); I_3, I_4 – the given moments of inertia of the corresponding wheels concerning an axis of wheel couple; φ_2, φ_3, φ_4 – angular data of an output shaft of a reducer and corresponding wheels; $\dot{\varphi}_2$, $\dot{\varphi}_3$, $\dot{\varphi}_4$ – angular speeds; $\ddot{\varphi}_2$, $\ddot{\varphi}_3$, $\ddot{\varphi}_4$ – angular accelerations; r – radius of a circle of driving of wheels; M_t – the braking moment on day off to a reducer shaft (in case of an arrangement of a disk brake on an engine shaft $M_t=u M'_t/2$, where u – a gear ratio of a reducer; M'_t – the braking moment on an engine shaft).

Coefficients of coupling of wheels with rails ψ_3, ψ_4 are also ψ_4 functions of relative slidings of the corresponding wheels and are on a formula [6]

$$\psi=k_1\left[\ \mathrm{th}(k_2 S)-k_3 S+k_4 S^3\right]\ .\tag{2}$$

Relative slidings can be determined by formulas at any moment

$$S_3 = (\dot{\varphi}_3 r - \dot{y}_3)/\dot{y}_3 , \quad S_4 = (\dot{\varphi}_4 r - \dot{y}_4)/\dot{y}_4 . \tag{3}$$

At small tilt angles of a way ($|\beta| \le 5\,°$)

$$\sin\beta \approx \mathrm{tg}\,\beta = \frac{i}{1000},$$

where i – a bias of a way (positive at the movement on rise and negative at the movement on descent).

Let's find the maximum absolute value of a sine of the angle of an inclination of a way at which temperature of a working surface of a brake disk will not exceed admissible value T_d (under the terms of work of frictional couple) after the locomotive, moving on descent with the switched-off engines, passes a way y with a constant speed v with the set mass of structure. Let's believe that from brake means only the disk brake of each drive cart is involved.

At the first stage the task is reduced to definition of the best axial effort $N_{\max}$ at which through a period $t = y/v$ temperature of a surface of friction of the brake disk rotating with a constant angular speed $\omega = (v/r)u$ at the reference temperature $T_\mathrm{n} = 25$ of °C will not exceed T_d . For this purpose we will use a formula

$$T_{1,\,2} = \theta_{1,\,2}(T_\mathrm{d} - T_\mathrm{n}) + T_\mathrm{n}, \tag{4}$$

where $T_{1,\,2}$ – temperature on a friction surface (hereinafter the index 1 belongs to a disk, 2 – to frictional pads); $\theta_{1,2}$ – dimensionless temperature.

Dimensionless temperature in turn is on a formula [7]

$$\theta_{1,\,2}(\rho,\,0,\,Fo) = \frac{2\pi Bi_{1,\,2}}{Bi_{1,\,2}^2 + 1} \sum_{n=1}^{\infty} \frac{V_{0\;1,\,2}(v_n\rho)\left(2 + \pi\rho_1 V_{0\;1,\,2}(\rho_1 v_n)\right)}{v_n\left(4 - \pi^2\rho_1^2 V_{0\;1,\,2}(\rho_1 v_n)\right)} \times \int_0^{Fo} Ki(Fo - \tau)\,\varphi_{1,\,2}(v_n,\,\tau)\,d\tau\ , \tag{5}$$

where $\rho = r/R_2$ – dimensionless radius; r – current radius; R_2 – external radius of a disk; $Fo = a_1 t/R_2^2$ – Fourier's criterion (dimensionless time); $a_{1,\,2} = \lambda_{1,\,2}/c_{1,\,2}\gamma_{1,\,2}$ – heat diffusivity coefficients; λ_1, λ_2 – heat conductivity coefficients; c_1, c_2 – specific heat capacities; γ_1, γ_2 – density; t – current time; $Bi_{1,\,2} = (\sigma_{1,\,2}/\lambda_{1,\,2})\,R_2$ – Biot's criterion; σ_1, σ_2 – the heat emission coefficients considering specific conditions of process of heat emission; $V_{01,2}(v_n\rho) = \left(Bi_{1,2}Y_0(v_n) - v_n Y_1(v_n)\right) J_0(v_n\rho) + \left(v_n J_1(v_n) - Bi_{1,2}J_0(v_n)\right)Y_0(v_n\rho)$ – kernel of final integral transformation of Hankel; v_n – eigenvalues; J_0, Y_0 – Bessel functions according to the first and the second sort of an order zero; J_1, Y_1 – Bessel functions according to the first and the second sort of first order; $\rho_1 = R_1/R_2$; R_1 – internal radius of a disk; $Ki = \dfrac{q(t)\,R_2}{(T_\mathrm{d} - T_\mathrm{n})\,\lambda_1}$ – Kirpichev's criterion; $q(t) = \dfrac{M_\mathrm{t}\,\omega}{t_\mathrm{t}\,F}\displaystyle\int_0^t \left(1 - \frac{\tau}{t_\mathrm{t}}\right) d\tau$ – thermal flow; $M_\mathrm{t} = \mu N_{max} R_e$ – the brake moment arising in a disk brake; μ – friction coefficient for couple of materials of a disk and a frictional pad; $R_\mathrm{e} = \dfrac{2}{3}\dfrac{R_2^3 - R_1^3}{R_2^2 - R_1^2}\dfrac{\alpha}{\sqrt{2(1 - \cos\alpha)}}$ – equivalent radius of friction; α – central corner of ring sector of a frictional pad; t_t – braking time; $F = \alpha\left(R_2^2 - R_1^2\right)/2$ – area of contact; $\varphi_1 = \alpha_\mathrm{tp}\kappa e^{-v_n^2 Fo}\left(\dfrac{1}{\sqrt{\pi Fo}} - (1-\kappa)\,Bi_1 e^{\kappa^2 Bi_1^2 Fo}\,\mathrm{erfc}\left((1-\kappa)\,Bi_1\sqrt{Fo}\right)\right)$; $\varphi_2 = \dfrac{(1 - \alpha_\mathrm{tp})\sqrt{a}\ e^{-a v_n^2 Fo}}{\lambda\sqrt{\pi Fo}}$; $\alpha_\mathrm{tp} = \sqrt{\lambda_1 c_1\gamma_1}\Big/\left(\sqrt{\lambda_1 c_1 \gamma_1} + \sqrt{\lambda_2 c_2 \gamma_2}\right)$ – the coefficient of distribution of thermal flows showing what part of heat generated at friction is taken away in a brake disk; $\kappa = \alpha/2\pi$; $\operatorname{erfc} x = \frac{2}{\sqrt{\pi}}\int_x^{\infty} e^{-\tau^2} d\tau = 1 - \operatorname{erf} x$; $\operatorname{erf} x = \frac{2}{\sqrt{\pi}}\int_0^{x} e^{-\tau^2} d\tau$ – integral of probabilities; $a = a_2/a_1$; $\lambda = \lambda_2/\lambda_1$.

Further, we will substitute the value of the brake moment M_t' corresponding $N_{\max}$ in system of differential equations (1). For a disk brake with a multisector disk we use a formula

$$M_t' = \frac{2}{u}\left(M_0 - A\sin\left(\alpha\varphi_2\right)\right) = M_0' - A'\sin\left(\alpha'\varphi_1\right) =$$
$$= M_0'\left(1 - A^*\sin\left(\alpha'\varphi_1\right)\right) = M_0'\left(1 - \frac{\mu_1-\mu_2}{\mu_1+\mu_2}\sin\left(\alpha'\varphi_1\right)\right) \quad \left(\mu_1 > \mu_2\right), \tag{6}$$

where M_0 – a constant component of the moment of braking on an axis of wheel couple; A – amplitude of fluctuations of a variable component of the moment of braking on an axis of wheel couple; α – number of the periods of a sinusoid for one turn of wheel couple; $M_0' = 2M_0/u$ – a constant component of the moment of braking on an engine shaft; $A' = 2A/u$ – amplitude of fluctuations of a variable component of the moment of braking on an engine shaft; $\alpha' = \alpha/u$ – number of the periods of a sinusoid for one turn of a shaft of the engine; μ_1, μ_2 – friction coefficients for two couples of materials of a disk and frictional pads.

The system of differential equations (1) taking into account formulas (2), (3), (6) is nonlinear as unknown functions are included into it not linearly. She represents mathematical model of braking of the mine locomotive a disk brake on a long bias at the nonlinear brake moment. Having integrated system (1) taking into account formulas (2), (3), (6), we will find value of a sine of the angle of an inclination of a way β at which the speed of the locomotive will remain to a constant.

We will carry out calculations at the parameters of disk brakes specified below with homogeneous and multisector brake disks. A disk brake with a homogeneous disk: disk material – steel 45 HB 415; material of frictional pads – 6KH-1; internal radius of a working zone of a disk $R_1 = 9{,}3 \cdot 10^{-2}$ m; external radius of a working zone of a disk $R_2 = 1{,}7 \cdot 10^{-1}$ m; disk thickness $2b_1 = 2{,}5 \cdot 10^{-2}$ m; a form of frictional pads – in the form of ring sector with the central corner $\alpha = \pi/4$; thickness of frictional pads $2b_2 = 1{,}1 \cdot 10^{-2}$ m. A disk brake with a multisector disk: quantity of the sectors of a brake disk made in turn of steel 45 HB 415 and the SCh 15-32 NV 200 gray cast iron, – eight; material of frictional pads – 6KH-1; internal radius of a working zone of a disk $R_1 = 9{,}3 \cdot 10^{-2}$ m; external radius of a working zone of a disk $R_2 = 1{,}8 \cdot 10^{-1}$ m; disk thickness $2b_1 = 2{,}5 \cdot 10^{-2}$ m; a form of frictional pads – in the form of ring sector with the central corner $\alpha = \pi/4$; thickness of frictional pads $2b_2 = 1{,}1 \cdot 10^{-2}$ m.

We will carry out calculation of the best axial effort $N_{\max}$ for a multisector brake disk in the assumption that the disk is not broken into sectors and it is made either of steel 45 HB 415, or of the SCh 15-32 NV 200 gray cast iron. As rated we will accept smaller of the received values.

At the chosen parameters of a disk brake with a homogeneous disk, the mass of structure $m_c = 5 \cdot 10^4$ kg, a way $y = 1000$ m, speed $v = 3$ m/s and rails covered with liquid coal dirt, we will receive $\left|\sin\beta\right| = 0{,}0139$ that corresponds $|i| \approx 13{,}9$ ‰. At the chosen parameters of a disk brake with a multisector disk and the same input datas $\left|\sin\beta\right| = 0{,}0152$ that corresponds $|i| \approx 15{,}2$ ‰.

It should be noted that at other conditions of a railway line we will receive approximately the same values of a bias of a way, but relative sliding of wheels on rails will differ.

Conclusions

1. On the basis of the developed mathematical model comparative analysis of braking of the mine locomotive on a long bias is made by the disk brake creating the constant and pulsing sinusoidal brake moments on an axis of wheel couple.

2. It is established that at the chosen parameters of a disk brake with a multisector disk the maximum absolute value of a bias of a way at which temperature of a working surface of a brake disk will not exceed admissible value during long descent for set lengths of a way, speeds and the mass of structure will be more, than at the chosen parameters of a disk brake with a homogeneous disk. For example, with a length of way 1000 m,

the speed of the engine 3 m/s and mass of structure $5 \cdot 10^4$ kg the difference of biases is equal to 1,3‰ that makes 9,35% of the maximum absolute value of the bias corresponding to a disk brake with a homogeneous disk.

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АНАЛИЗ РЕЗУЛЬТАТОВ МОДЕЛИРОВАНИЯ ЭЛЕКТРОМЕХАНИЧЕСКОЙ СИСТЕМЫ «ЭЛЕКТРИЧЕСКАЯ СЕТЬ – ПРИВОД – КОМПРЕССОР – ПНЕВМОСЕТЬ»

Анотація. *В роботі проведено аналіз результатів моделювання математичної моделі, що дозволяє визначати оптимальний режим роботи системи виробництва і розподілу стислого повітря, що складається з наступних елементів "електрична мережа - привод - компресор - пневмосети". Проведений аналіз показав, що запропонований варіант регулювання з «плаваючим» верхнім рівнем тиску забезпечує скорочення витрат електричної енергії, споживаної розглянутою електромеханічною системою. У зіставленні з класичним двохпозиційним регулюванням, економія досягається в межах 1...13% в залежності від значень витрати стислого повітря, споживаного пневмоприймачами..*

Ключові слова: *електропривод, регулювання, компресор, електромеханічна система.*

Аннотация. *В работе проведен анализ результатов моделирования математической модели, позволяющей определять оптимальный режим работы системы производства и распределения сжатого воздуха, состоящей из следующих элементов “электрическая сеть – привод – компрессор – пневмосеть”. Проведенный анализ показал, что предлагаемый вариант регулирования с «плавающим» верхним уровнем давления обеспечивает сокращение расхода электрической энергии, потребляемой рассматриваемой электромеханической системой. В сопоставлении с классическим двухпозиционным регулированием, экономия достигается в пределах 1…13 % в зависимости от значений расхода сжатого воздуха, потребляемого пневмоприемниками.*

Ключевые слова: *электропривод, регулирование, компрессор, электромеханическая система.*

Abstract. *The paper analyzes the results of modeling a mathematical model that allows to determine the optimal mode of operation of the production and distribution of compressed air, consisting of the following elements “electrical network - drive - compressor - pneumatic network”. The analysis showed that the proposed control option with a “floating” upper pressure level ensures a reduction in the consumption of electrical energy consumed by the electromechanical system under consideration. In comparison with the classical two-stage regulation, savings are achieved within 1...13%, depending on the values of the compressed air consumption consumed by the pneumatic receivers.*

Keywords: *electric drive, control, compressor, electromechanical system.*