# ТРАНСПОРТ ТА ГІРНИЧА МЕХАНІКА

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## МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ ПРОЦЕСУ ГАЛЬМУВАННЯ ШАХТНОГО ЛОКОМОТИВА ДИСКОВИМ ГАЛЬМОМ З ІМПУЛЬСНИМ ТОРМОЗНИМ МОМЕНТОМ ДЛЯ ВИЗНАЧЕННЯ ХАРАКТЕРИСТИК ЙОГО ПРИВОДА

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## MATHEMATICAL MODELING OF SHAFT LOCOMOTIVE BRAKING WITH A PULSING BRAKE MOMENT FOR DETERMINING THE CHARACTERISTICS OF ITS DRIVE

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**Мета**. розробка, розв'язок і аналіз математичної моделі, що дозволяє визначити динамічні та кінематичні характеристики приводу шахтного локомотива при гальмуванні пульсуючим гальмовим моментом.

**Методи дослідження** засновані на основі математичного моделювання процесу гальмування шахтного локомотива дисковим гальмом.

**Результати дослідження**. Показано, що на початку гальмування кутова швидкість колеса може збільшуватися протягом коротких проміжків часу.

**Наукова новизна**. На основі математичного моделювання процесу гальмування шахтного локомотива дисковим гальмом, що створюють пульсуючий гальмовий момент, установлені динамічні й кінематичні характеристики його приводу при реальних умовах роботи.

**Практичне значення**. Отримано графіки лінійної і кутової швидкостей ланок колісно-моторного блоку, відносного ковзання, коефіцієнта та сили зчеплення коліс з рейками, моменту гальмування, створюваного на вихідному валу редуктора, і гальмового моменту на колесі.

***Ключові слова:*** *локомотивне відкочування, гальмова сила, гальмовий момент, коефіцієнт зчеплення, дискове гальмо, фрикційна пара.*

The performance of underground mining depends on the performance of vehicles. One of the main modes of transport in coal and mine mines is locomotive haulage. In mine trains, only loco-motifs are equipped with braking means. Therefore, the braking capabilities of the train in specific mine conditions are determined by the braking force, which is realized by the locomotive.

The main problem that arises during braking of mine locomotives with a wheel-block brake is the instability of the braking force generated by the brake shoe while reducing the locomotive's speed [1]. Impulse brake pressing used when braking the rolling stock of the main and industrial railway transport is a promising direction for improving the braking performance, but significantly complicates the design of the brake system. The use of a pulsating braking torque due to the use of a disc brake with a multi-sector disc is a simple solution that improves the braking characteristics of mine shaft locomotives.

The book [2] provides a methodology for choosing a constant braking torque applied to the axis of a wheel pair. In order to prevent clutch disruption and wheel use for mine electric locomotives, it is recommended to realize 80% of the maximum possible braking torque. In [3, 4], a study was made of the braking process of a mine locomotive with a disk brake that creates a pulsating braking torque on the axis of the wheelset in order to realize the maximum possible coefficient of adhesion of wheels to rails. Recommendations are given on the analytical choice of braking torque for various rail conditions. Constructive conceptual solutions for the manufacture of a disk brake with a multi-sector disk that creates a pulsating braking torque are proposed.

***Article purpose*** – development, solution and analysis of a mathematical model to determine the dynamic and kinematic characteristics of a mine locomotive drive during braking with a pulsating braking torque.

Forced vibrations of a wheel-motor block of a mine locomotive during braking by a disk brake on a straight horizontal section of a rail track, taking into account the nonlinear characteristics of the interaction of the friction pair wheel-rail, can be described by a system of six second-order differential equations [3]

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

where $m_s$ – 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; $\beta_{y3}$, $\beta_{y4}$ – 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; $\beta$ – a tilt angle of a way (positive at the movement on rise and negative at the movement on descent); $F_3=\psi_3\left(S_3\right)\left(m_1g/8\right)\cos\beta$, $F_4=\psi_4\left(S_4\right)\left(m_1g/8\right)\cos\beta$ – forces of adhesion of the corresponding wheels; $\psi_3$, $\psi_4$ – coefficients of coupling of the corresponding wheels; $S_3$, $S_4$ – relative slidings of the corresponding wheels; $m_1$ – 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; $\varphi_2$, $\varphi_3$, $\varphi_4$ – 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).

We take the number of sectors of the brake disc, made in turn from steel 45 HB 415 and gray cast iron SCh 15-32 HB 200, equal to eight. Brake pads in the form of an annular sector with a central angle of friction material 6KX-1 (cold forming press material) [3]. The friction coefficients for these pairs of disk materials and friction linings are respectively 0.535 and 0.41 [6].

For the selected number of sectors of the brake disc and the shape of the friction linings, the dependence of the pulsating braking torque on the motor shaft on the angular coordinate of the motor shaft can be described with a sufficient degree of accuracy by the expression [7]

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

where, $M_0$, $M_0'$ – the constant components of the braking moments, respectively, on the axis of the wheelset and on the motor shaft; $n$, $n'$ – the number of periods of a sinusoid per revolution, respectively, of the axis of the wheel pair and the motor shaft; $A$, $A'$ – the amplitude of the oscillations of the variable components of the braking moments on the axis of the wheelset and on the motor shaft; $A^*=A'/M_0'=\left(\mu_1-\mu_2\right)/\left(\mu_1+\mu_2\right)$; $\mu_1$, $\mu_2$ – friction coefficients for two pairs of disc materials and friction linings.

Based on the selected parameters of a disk brake with a multi-sector disk, we set the values of the coefficients $n'$ and $A^*$. In our case $n'=4$ ; $A^*=0{,}132$ . Further, integrating the system of differential equations (1), taking into account formula (2), we determine the maximum value of the constant component of the braking torque on the engine shaft $M'_{0max}$, corresponding to the initial data (the state of the rail track, the mass of the train, the initial speed of the locomotive) [7] . Substitute in the formula (2) $M'_0 = 0{,}8M'_{0max}$. Integrating the system of differential equations (1) taking into account formula (2) for given initial data, we obtain graphs of the linear and angular velocities of the links of the wheel-motor block, relative slip, coefficient and force of adhesion of the wheels to rails, and the braking moment created on the output shaft of the gearbox, and the braking torque on the wheel, the forces in the rubber mounts of the wheel pair suspension, the speed and the path of the locomotive from time to time, we determine the braking distance and the braking time. In numerical calculations we will use the geometric, weight, elastic-dissipative and stiffness characteristics of the elements of the mine electric locomotive E10. Some of the graphs obtained for a rail track sprinkled with sand crushed as a result of a previous trip, with the mass of the composition $m_s = 5\cdot10^4$ kg and the initial locomotive speed $v_0 = 4{,}5$ m/s are shown in Fig. 1-5.

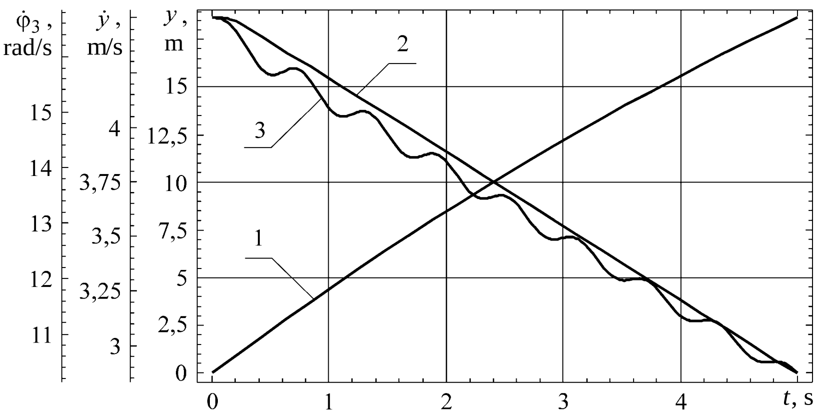


**Fig. 1. Dependences of the path and speed of the locomotive, the angular velocity of one of its wheels on time: 1 − dependence of the path of the locomotive; 2 − locomotive speed dependence; 3 − dependence of the angular velocity of one of the wheels**

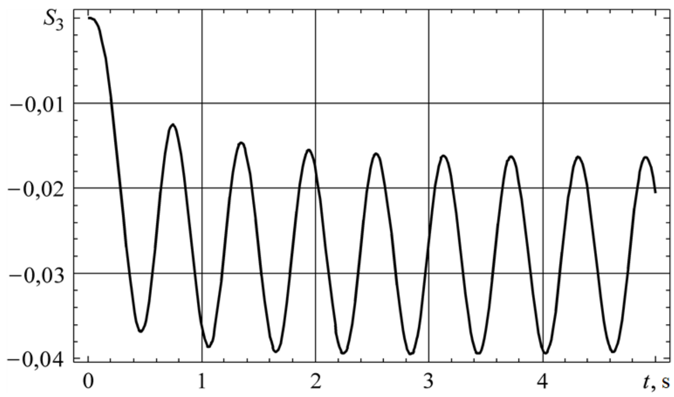


**Fig. 2. The dependence of the relative slip of one of the wheels of the locomotive on time**

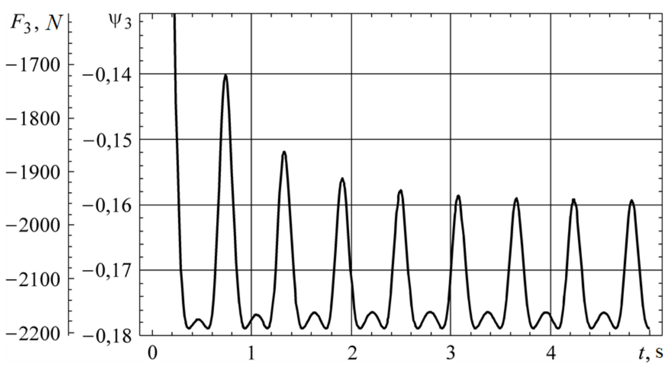


**Fig. 3. Dependences of the coefficient of adhesion and the adhesion force of one of the wheels of a locomotive with a rail on time**

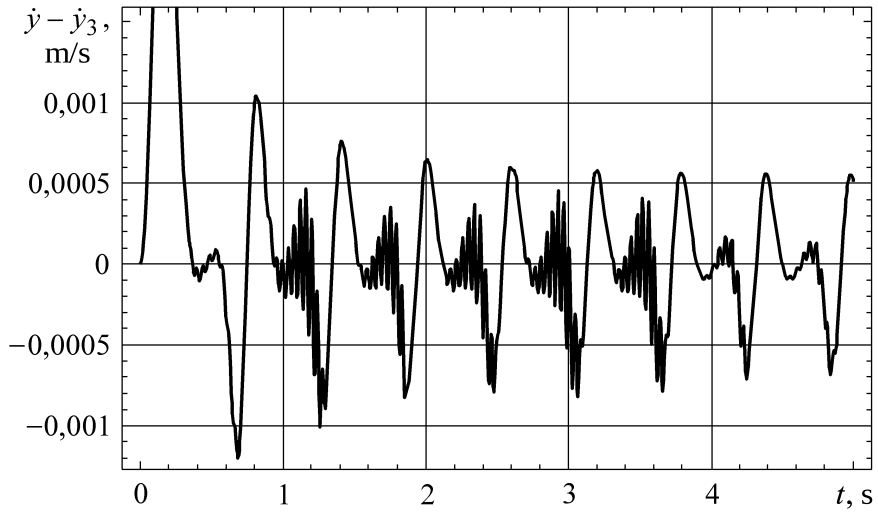


**Fig. 4. The dependence of the difference between the linear speeds of the locomotive and one of its wheels on time**

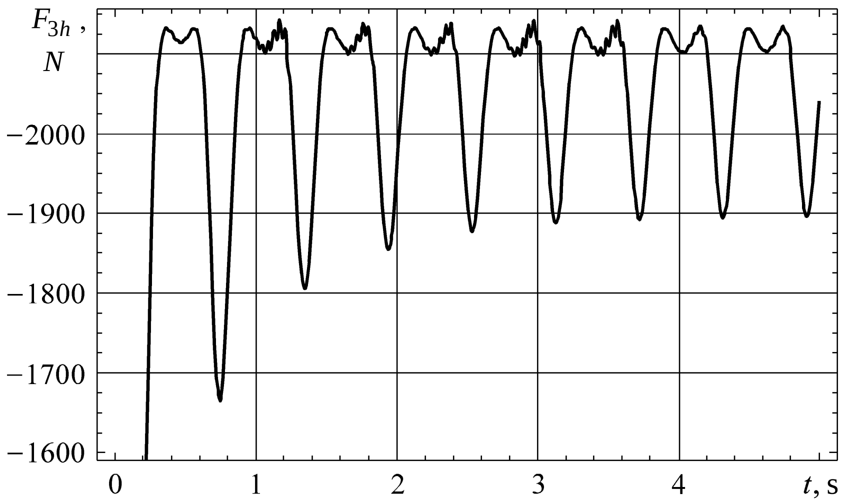


**Fig. 5. The dependence of the longitudinal force in one of the rubber-thallic hinges of the wheel pair suspension on time**

As can be seen from fig. 1 the angular speed of the wheel at the beginning of braking varies unevenly. For short periods of time, it can even increase. This is due to the presence of an oscillatory process in the wheel movement, characterized by an increase and decrease in the amount of wheel slippage relative to the rail.

## Conclusions

1. Based on mathematical modeling of the braking process of the E10 mine locomotive with a disk brake with a multi-sector disk, the dynamic and kinematic characteristics of its drive are determined for given initial data;
2. It has been established that due to the oscillatory process, characterized by an increase and decrease in the value of wheel slippage relative to the rail, at the beginning of braking, the angular velocity of the wheel can increase over short periods of time.

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## АННОТАЦИЯ

**Цель**. Разработка, решение и анализ математической модели, позволяющей определить динамические и кинемачеcкие характеристики привода шахтного локомотива при торможении пульсирующим тормозным моментом.

**Методы исследования** основаны на основе математического моделирования процесса торможения шахтного локомотива дисковым тормозом.

**Результаты исследования**. Показано, что в начале торможения угловая скорость колеса может увеличиваться в течение коротких промежутков времени.

**Научная новизна**. На основе математического моделирования процесса торможения шахтного локомотива дисковым тормозом, создают пульсирующий тормозной момент, установлены динамические и кинематические характеристики его привода при реальных условиях работы.

**Практическое значение**. олучены графики линейной и угловой скоростей звеньев колесно-моторного блока, относительного скольжения, коэффициента и силы сцепления колес с рельсами, момента торможения, создаваемого на выходном валу редуктора, и тормозного момента на колесе.

***Ключевые слова:*** *локомотивный откат, тормозная сила, тормозной момент, коэффициент сцепления, дисковый тормоз, фрикционная пара.*

## ABSTRACT

**Objective.** development, solution and analysis of a mathematical model that allows to determine the dynamic and kinematic characteristics of the drive of the mine locomotive during braking by pulsating braking torque.

**The research methodology** are based on mathematical modeling of the process of braking a mine locomotive with a disc brake.

**Research results**. It is shown that at the beginning of braking the angular speed of the wheel can increase for short periods of time.

**Scientific novelty.** On the basis of mathematical modeling of the process of braking a mine locomotive with a disc brake, which creates a pulsating braking moment, the dynamic and kinematic characteristics of its drive under real operating conditions are established.

**Practical meaning.** Graphs of linear and angular velocities of the wheel-motor unit, relative slip, coefficient and force of traction of wheels with rails, braking moment created on the output shaft of the gearbox, and braking torque on the wheel are obtained.

***Keywords****: locomotive rolling out, braking force, braking moment, coupling coefficient, disc brake, friction pair.*