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MATHEMATICAL SUBSTANTIATION OF MINE LOCOMOTIVE BRAKING EFFICIENCY UNDER PULSITING BRAKING TORQUE

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МАТЕМАТИЧНЕ ОБҐРУНТУВАННЯ ЕФЕКТИВНОСТІ ГАЛЬМУВАННЯ ШАХТНОГО ЛОКОМОТИВА ПРИ ПУЛЬСУЮЧОМУ ГАЛЬМІВНОМУ МОМЕНТІ

Purpose. Determination of braking torque parameters that provide high braking performance of a mine locomotive by mathematical modeling of the braking of a mine locomotive by a disc brake that creates a pulsating braking torque.

The methods. Forced vibrations of the elements of the wheel-motor unit (WMU) of a mine locomotive in the process of braking with a disc brake are studied by the methods of differential calculus and mathematical modeling. Integration of the system of differential equations is performed by the Runge-Kutt method.

Findings. A comparative analysis of a mine locomotive braking by a disc brake, which creates a constant and pulsating sinusoidal braking torque with a different number of sinusoid periods per revolution of the wheel pair, is carried out. It has been proved that the pulsating sinusoidal braking moment generated on the axle of the wheel pair equal to the sum of the constant component and the amplitude of the oscillations of the variable component multiplied by the sine of the product of the number of sinusoid periods per revolution of the wheel pair by its angular coordinate provides higher braking characteristics than the constant braking moment. It is shown that pulsating braking torque reduces the braking time and braking distance of the mine locomotive. The values of the amplitude of oscillations, which depend on the average value of the braking torque, and the number of periods of the sinusoid per one revolution of the wheelset, at which the greatest effect is achieved, are determined.

The originality. For the first time, a mathematical model has been developed for braking a mine locomotive by a disc brake, which creates a pulsating braking torque on the axle of the wheel pair, depending on its angular coordinate, taking into account the nonlinear dependence of the adhesion coefficient on relative slip, on the basis of which the efficiency of a disc brake with a multi-sector disc is shown in comparison with a disc brake with a homogeneous disk under various conditions of the rail track.

Practical implementation. The research results make it possible to determine the parameters of a disc brake with a multi-sector disc located on the wheelset axle and on the engine shaft, which provide the highest braking efficiency.

Keywords: *mine locomotive, disk brake, frictional pair, relative sliding, clutch coefficient.*

Introduction. An increase in the productivity of mine rail transport is possible only with high reliability of the brake systems of locomotives. The main characteristics that determine the efficient operation of a mine locomotive include the realizable traction and braking forces, reliability and energy consumption.

Great attention is paid to the study of the process of realizing the maximum possible adhesion force of the wheels of the locomotive with the rails. This force depends both on the state of the track and on the conditions of interaction of the wheel-rail friction pair [1]. The main parameter characterizing the force of adhesion of wheels to rails is the clutch coefficient [2].

In the work [3] a method for choosing a constant braking torque applied to the axle of the wheelset is given. In order to prevent slipping of the clutch and skidding of the wheels, it is recommended to implement 80% of the maximum possible braking torque for mining electric locomotives.

In the paper [4] examples of useful application of vibration, which are based on phenomena related to vibration features in non-linear mechanical systems are present, a general approach to the study and use of vibration is outlines. In particular, attention is paid to the study of the coefficient of sliding friction during vibration.

In the works [5, 6] recommendations are given on the analytical choice of the braking torque applied to the axle of the wheel pair of a mine locomotive in order to achieve the most effective braking for various states of the track. Constructive conceptual solutions are proposed for the manufacture of a disc brake that creates a pulsating braking torque.

The braking torque created on the wheel by a wheel-shoe brake depends on the speed of the mine locomotive, the condition of the rail track and the heating of the brake shoe, which does not allow to fully realize the possible coefficient of adhesion. Disc brakes used in transport systems do not have this disadvantage. Research aimed at determining the rational parameters of the disc brake of a mine locomotive, studying the dynamics of the drive during braking will contribute to improving the safety of movement, increasing the throughput of intra-mine transport, and developing mining engineering in Ukraine.

Main part. The forced oscillations of the wheel-motor block of a mine locomotive during braking by a disc brake, taking into account the nonlinear characteristics of the interaction of the wheel-rail friction pair, can be de-scribed by a system of six second-order differential equations

$$\left.\begin{aligned}
&(m_s/4 - m_3 - m_4)\ddot{y} = -\Big[C_{y3}(y - y_3) + \beta_{y3}(\dot{y} - \dot{y}_3) + \\
&\qquad\qquad + C_{y4}(y - y_4) + \beta_{y4}(\dot{y} - \dot{y}_4)\Big], \\
&m_3\ddot{y}_3 = C_{y3}(y - y_3) + \beta_{y3}(\dot{y} - \dot{y}_3) + F_3(S_3)\ , \\
&m_4\ddot{y}_4 = C_{y4}(y - y_4) + \beta_{y4}(\dot{y} - \dot{y}_4) + F_4(S_4)\ , \\
&I_3\ddot{\varphi}_3 = -\Big[C_{\varphi 3}(\varphi_3 - \varphi_2) + \beta_{\varphi 3}(\dot{\varphi}_3 - \dot{\varphi}_2) + rF_3(S_3)\Big]\ , \\
&I_4\ddot{\varphi}_4 = -\Big[C_{\varphi 4}(\varphi_4 - \varphi_2) + \beta_{\varphi 4}(\dot{\varphi}_4 - \dot{\varphi}_2) + rF_4(S_4)\Big]\ , \\
&I_2\ddot{\varphi}_2 = C_{\varphi 3}(\varphi_3 - \varphi_2) + \beta_{\varphi 3}(\dot{\varphi}_3 - \dot{\varphi}_2) + C_{\varphi 4}(\varphi_4 - \varphi_2) + \\
&\qquad\qquad + \beta_{\varphi 4}(\dot{\varphi}_4 - \dot{\varphi}_2) - M_t\ ,
\end{aligned}\right\}, \qquad (1)$$

where m_s – mass of the train; 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; $F_3 = \psi_3(S_3)(m_l\, g/8)\cos\beta$, $F_4 = \psi_4(S_4)(m_l\, g/8)\cos\beta$ – 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_1 – mass of the locomotive; g – acceleration of gravity; I_2 – the reductioned 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 – reductioned moments of inertia of the corresponding wheels concerning an axis of wheel couple; φ_2, φ_3, φ_4 – angular coordinates 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).

The clutch coefficients of wheels with rails ψ_3 and ψ_4 – functions of the relative slips of the corresponding wheels and are found by the formula [7]

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

Where k_1, k_2, k_3, k_4 – numerical coefficients of the mechanical characteristics of the wheel-rail friction pair (depend on the condition of the track); S – relative sliding of the wheel along the rail (negative in the braking mode and positive in the acceleration mode).

Relative slips at any given time can be divided by formulas

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

The integration of the system of differential equations (1), taking into account formulas (2) and (3), was performed by the Runge-Kutt method for four states of the rails (sprinkled with sand; covered with sand crushed as a result of a previous drive; wet, clean; covered with liquid coal mud).

In numerical calculations geometric, weight, elastic-dissipative and stiffness characteristics of the elements of the mine electric locomotive E10 were used. The mass of the composition was taken equal to $5 \cdot 10^4$ kg. The initial speed of the locomotive differed in different states of the track. The braking torque created by a disc brake with a multi-sector disc on the wheelset axle was determined by the formula

$$M_t = M_0 - A\sin\left(\alpha\varphi_2\right) = M_0\left(1 - A^*\sin\left(\alpha\varphi_2\right)\right),$$

where M_0 – a constant component of the braking torque on the wheelset axle; A – an amplitude of fluctuations of the variable component of the braking torque on the wheel pair axle; α – the number of periods of the sinusoid in one revolution of the wheelset; φ_2 – angular coordinate of the axis of the wheel pair axle; $A^{*}= A/M_0$.

First of all, the value of the constant component of the braking torque M_{0max}, was determined, at which during the braking process the clutch breaks down and the wheel begins to move skidding. Then 80% of this value was found. The braking torque was set constant ($A = 0$) and pulsating ($A \neq 0$).

The distance (Fig. 1) passed by the locomotive during braking on the track sprinkled with sand for the initial speed of the locomotive $v_0 = 5$ m/s will be less with a pulsating braking moment. With a constant braking torque ($M_0 = 2480$ N m, $A = 0$) the braking time is 14.1 s, and the braking distance is 36 m. With a pulsating braking torque with a small value of the parameter α ($M_0 = 2650$ N m, $A = 340$ Н·м, $\alpha = 4$) the braking time and the braking distance decrease by 11%, which is respectively 12.6 s and 32 m. With a pulsating braking torque with a large value of the parameter α ($M_0 = 2730$ N m, $A = 360$ N m, $\alpha = 44$) the braking time and braking distance decrease by 19%, which is 11.4 s and 29 m, respectively.

It is interesting to note that the angular velocity $\dot{\varphi}_3$ of the wheel at the beginning of braking changes unevenly. It may even increase for short periods of time. This is due to the presence of an oscillatory process in the movement of the wheel, characterized by an increase and decrease in the slip of the wheel relative to the rail.

Similar calculations that were carried out for other rail track states led to the following results (everywhere M_0 is 80% of its maximum possible value). For the case when the rails are covered with sand crushed as a result of a previous drive (the initial speed of the locomotive $v_0 = 4.5$ m/s): with a constant braking torque ($M_0 = 1900$ N·m, $A = 0$) the braking time is 16.4 s, and the braking distance is 38 m; with a pulsating braking torque with a small value of the parameter α ($M_0 = 2020$ N·m, $A = 260$ N·m, $\alpha = 4$) the braking time and the braking distance are reduced by 11% and amount to 14.6 s and 34 m, respectively; with a pulsating braking torque with a large value of the parameter α ($M_0 = 2100$ N·m, $A = 270$ N·m, $\alpha = 44$) the braking time and the braking distance are reduced by 18%, which is 13.5 s and 31 m, respectively. For the case when the rails are wet, clean (the initial speed of the locomotive $v_0 = 3.5$ m/s): with a constant braking torque ($M_0 = 1410$ N·m, $A = 0$) the braking time is 16.7 s, and the braking distance is 30 m; with a pulsating braking torque with a small value of the parameter α ($M_0 = 1490$ N·m, $A = 190$ N·m, $\alpha = 4$) the braking time and the braking distance are reduced by 10% and are respectively 15 s and 27 m; with a pulsating braking torque with a large value of the parameter α ($M_0 = 1550$ N·m, $A = 215$ N·m, $\alpha = 44$) the braking time and the braking distance are reduced by 16%, which is 14 s and 25.2 m, respectively. For the case when the rails are covered with liquid coal mud (the initial speed of the locomotive $v_0 = 3$ m/s): with a constant braking torque ($M_0 = 830$ N·m, $A = 0$) the braking time is 23.6 s, and the braking distance is 36 m; with a pulsating braking torque with a small value of the parameter α ($M_0 = 880$ N·m, $A = 115$ N·m,

α = 4) the braking time and braking distance are reduced by 8% and are respectively 21.7 s and 33 m; with a pulsating braking torque with a large value of the parameter α (M_0 = 910 N·m, A = 120 N·m, α = 44) the braking time and braking distance are reduced by 14%, which is 20.3 s and 31 m, respectively.

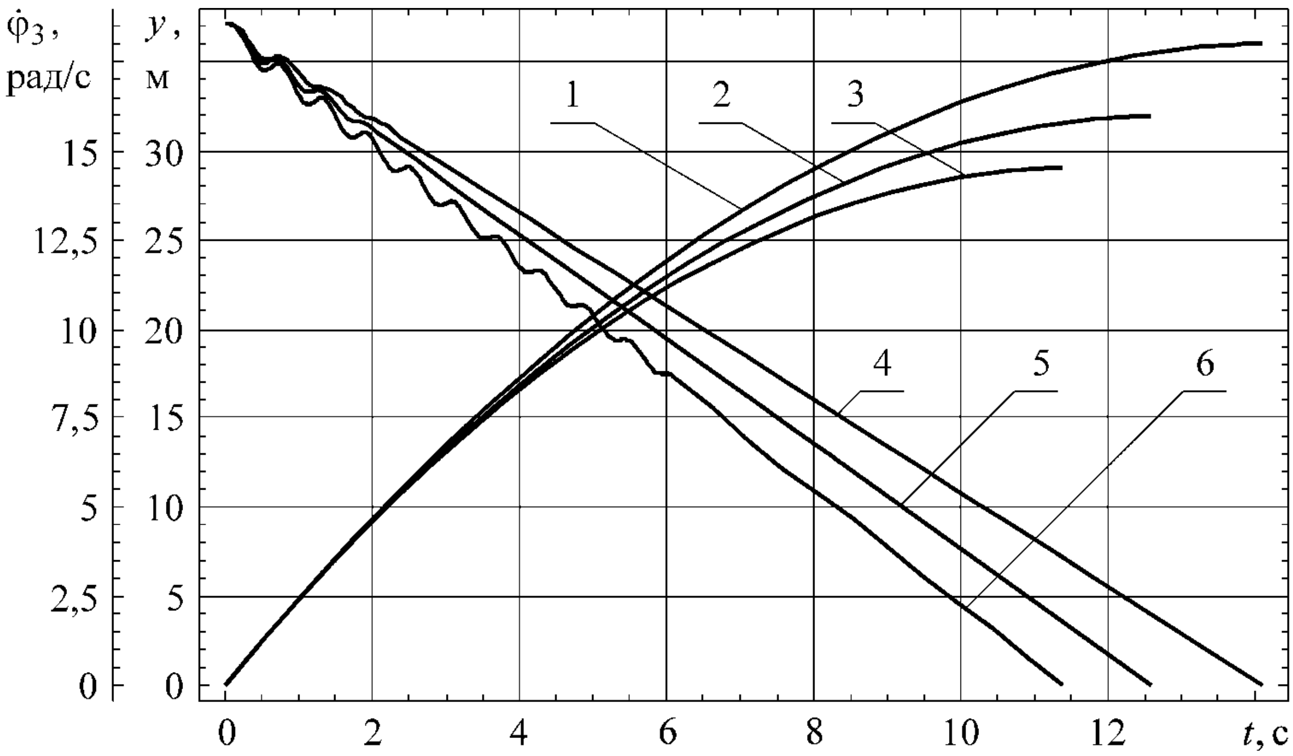

Fig. 1. Dependences of the distance passable by the locomotive and the angular velocity of one of its wheels on time: 1 – dependence of the distance with M_0 = 2480 N m, A = 0; 2 – dependence of the distance with M_0 = 2650 N m, A = 340 N m, α = 4; 3 – dependence of the distance with M_0 = 2730 N·м, A = 360 N·м, α = 44; 4 – dependence of the angular velocity with M_0 = 2480 N·м, A = 0; 5 – dependence of the angular velocity with M_0 = 2650 Н·м, A = 340 N·м, α = 4; 6 – dependence of the angular velocity with M_0 = 2730 N·м, A = 360 N·м, α = 44

Calculations have shown that with a pulsating braking torque to break the clutch, it is necessary to apply a greater value M_{0max} than with a constant braking torque for any of the considered states of the rails, as well as for different values of a train mass and the initial speed of the locomotive. This results in reduced braking time and braking distances. This effect is most pronounced if the amplitude of oscillations A is 10 ... 15% of the constant component of the braking torque M_0, i.e. with values A^*that are within from 0.1 to 0.15. The number of periods of the sinusoid in one revolution of the wheel pair α also plays a significant role. The best results can be obtained with α that is equal to 35…55 (Fig. 2). The use of a pulsating braking torque on slippery rails gives a significant effect, although less in percentage terms than on dry rails.

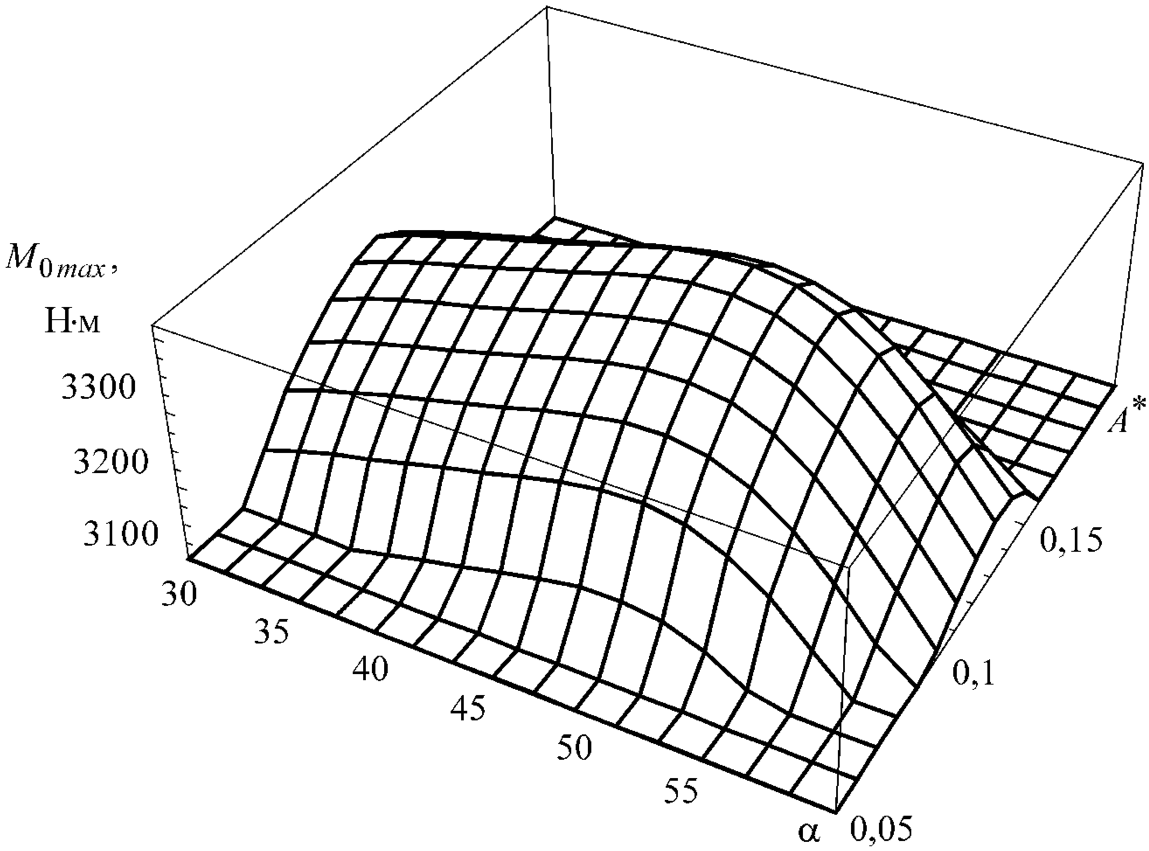

Fig. 2. Dependence of the maximum value of the constant component of the braking torque on the coefficients α and A^*(rails are covered with sand, $m_s = 5 \cdot 10^4$ kg, v_0 = 5 m/s)

The decrease in braking time and braking distance is due to the fact that with an increase in the maximum possible value M_0 with a pulsating braking torque the relative slip during braking takes values closer to the extremum point of the function $\psi = \psi(S)$ [7]. This, in turn, results to the fact that the value of the clutch coefficient of wheels to rails fluctuates around a value closer to the maximum possible for each of the considered states of the rail track.

When using a multi-sector disc made of materials that have different friction coefficients with the material of the brake pad linings made in the form of an annular sector [5, 6], the braking torque will change according to a dependence very close to sinusoidal. If the brake pad lining is made in the form of an annular sector with a central angle of 45° from friction material 6KH-1 (cold forming press material), and the brake disc is divided into eight sectors, made alternately from steel 45 HB 415 and gray cast iron SCH 15-32 HB 200 (the friction coefficients between these materials are respectively equal to 0.535 and 0.41 [8]), then the oscillation amplitude will be

$$\frac{0{,}535-0{,}41}{2} \div \frac{0{,}535+0{,}41}{2} \cdot 100\% = 13{,}2\%$$

of M_0. When the brake disc is placed on the axle of the wheel pair, the number of periods of the sinusoid for one of its revolutions α will be equal to four. Placing a brake disc on the motor shaft with a gear ratio of 10.875, we get α that is equal to 43.5.

Conclusions. A comparative analysis of the braking of a mine locomotive by a disc brake, which creates a constant and pulsating sinusoidal braking torques with a different number of sinusoidal periods per wheel pair revolution, is performed. It is shown that by creating a pulsating braking torque it is possible to reduce the braking time and braking distance of a mine locomotive.

It has been ascertained that the greatest effect from the use of pulsating braking torque is achieved if the oscillation amplitude is 10 ... 15% of the average value of the braking torque, and the number of sinusoid periods per one revolution of the wheel pair is between 35 and 55.

Further studies of the dynamics of braking of a mine locomotive by a disc brake should be directed at studying the influence on the braking distance and braking time of values of the stiffness and damping coefficients of rubber-metal elements as well as values of the stiffness and damping coefficients of the semi-axles of the wheel-motor unit.

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АНОТАЦІЯ

Мета. Встановлення параметрів гальмівного моменту, що забезпечують високі гальмівні характеристики шахтного локомотива шляхом математичного моделювання гальмування шахтного локомотива дисковим гальмом, яке створює пульсуючий гальмівний момент.

Методика. Вимушені коливання елементів колісно-моторного блоку (КМБ) шахтного локомотива в процесі гальмування дисковим гальмом досліджені методами диференціального числення та математичного моделювання. Інтегрування системи диференціальних рівнянь виконано методом Рунґе-Кутта.

Результати. Виконано порівняльний аналіз гальмування шахтного локомотива дисковим гальмом, що створює на осі колісної пари постійний та пульсуючий синусоїдальний з різним числом періодів синусоїди за один оберт колісної пари гальмівні моменти. Доведено, що пульсуючий синусоїдальний гальмівний момент, створюваний на осі колісної пари, який дорівнює сумі сталої складової та амплітуди коливань змінної складової, помноженій на синус добутку числа періодів синусоїди за один оберт колісної пари на її кутову координату, забезпечує більш високі гальмівні характеристики шахтного локомотива, ніж сталий гальмівний момент. Показано, що пульсуючий гальмівний момент забезпечує зменшення часу гальмування та гальмівного шляху шахтного локомотива. Встановлено значення амплітуди коливань, що залежить від середнього значення моменту гальмування, та кількості періодів синусоїди за один оберт колісної пари, при яких досягається найбільший ефект.

Наукова новизна. Вперше розроблено математичну модель гальмування шахтного локомотива дисковим гальмом, що створює на осі колісної пари пульсуючий гальмівний момент, який залежить від її кутової координати, з урахуванням нелінійної залежності коефіцієнта зчеплення від відносного ковзання, на базі якої показано ефективність дискового гальма з багатосекторним диском у порівнянні з дисковим гальмом з однорідним диском на при різних станах колії.

Практична значимість. Результати досліджень дозволяють встановити параметри дискового гальма з багатосекторним диском, розміщеним на осі колісної пари та на валу двигуна, що забезпечують найбільшу ефективність гальмування.

Ключові слова: *шахтний локомотив, дискове гальмо, фрикційна пара, відносне ковзання, коефіцієнт зчеплення.*