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THE EFFECT OF STIFFNESS OF SHOE BRAKE ELEMENTS ON THE DISTRIBUTION OF CONTACT PRESSURES

Existing methods for the mine hoisting machines shoe brakes calculating are unreasonably based on the hypothesis in which the brake rim and the brake beam are assumed to be absolutely rigid.

Purpose. To develop recommendations for reducing the maximum contact stresses when the brake lining interacts with the drum of a mine hoisting machine. The tasks of this paper are to determine the applicability of the hypothesis on the absolute rigidity of the brake beam and to determine the dependence of the contact pressures distribution nature on the ratio of the brake lining transverse rigidity to the brake beam bending stiffness.

Methodology. The Gauss method for the sequential elimination of unknown variables; Euler's method for solving systems of differential equations; Newton's method for determining the numerical values of the roots of a differential equation; the finite element method for optimizing the design of the brake beam.

Findings. An analytical model of the brake beam as a circular beam of a constant section on the Winkler base is developed, whose rigidity depends on the stiffness properties of the composite lining. From the analysis of the stress-strain state, a dimensionless factor determining the nature of the contact pressure distribution is revealed, namely, the relative rigidity. To clarify the effect of this factor, computational experiments for a beam of constant section and a real brake pad for various design solutions for varying the relative rigidity were carried out in the SolidWorks Simulation software. A method for determining the distribution of contact pressures depending on the ratio of the brake lining transverse rigidity to the brake beam bending stiffness is developed. A comparison of the results of various design approaches to achieve an even distribution of the contact pressure along the brake beam is presented.

Originality. For the first time it has been analytically proved that the nature of the distribution of the contact pressure in the shoe brake of mine hoisting machines depends on the ratio of the brake lining transverse rigidity to the brake beam bending stiffness, and with its decrease the character of the distribution tends to be sinusoidal.

Practical value. The application of these recommendations will allow reducing the maximum contact pressure in the shoe brakes of mine hoisting machines.

Keywords: *bending stiffness of the brake beam, pressure distribution, shoe brake, SolidWorks Simulation*

Introduction. Increasing the volume of mining requires the increase in the efficiency and reliability of mine winders. It is known that the braking system have the highest value in protecting the hoisting installation from an accident [1, 2].

The reduction of the shoe brake contact pressure is the actual technical problem. For the solution of this problem it is necessary to investigate the effect of the ratio of the lining transverse rigidity to beam bending stiffness on the character of the contact pressure distribution [3–5].

Analysis of the recent research and publications. Many well-known scientists have been involved in the development of braking devices for mine winders: V. I. Vasiliev, B. L. Davydov, Z. M. Fedorova, N. S. Karpyshev, V. I. Belobrov, V. F. Abramovsky, V. I. Samusya, Z. Barecki, S. F. Scieszka, Yuan Mao Huang, J. S. Shyr.

The basic technique for calculating mine winder braking devices is described in the works of B. L. Davydov, Z. M. Fedorova and N. S. Karpyshev. This method is based on the hypothesis that the brake rim and brake beam are assumed to be absolutely rigid.

In further numerous works on improving the calculation technique of brakes the dynamic and thermal processes taking place in the elements of the braking system were taken into account. For example, the works of V. I. Belobrov, V. F. Abramovsky, V. I. Samusya and

V. I. Vasiliev [1] considered the dynamics of the lifting system during working and safety braking. Most importantly, the hypothesis about beam's and rim's rigidity was not substantiated.

Unresolved aspects of the problem. The results of calculating the mine winder brake stress-strain state have some discrepancy with those described in the literature. Thus, for example, the nature of the contact pressure distribution along the brake beam is not sinusoidal, with peak values at the center of the shoe, but on the contrary, it has the so-called U-shaped character with maximum on the edges. Therefore, an urgent scientific task is to determine the factors that affect the distribution of contact pressure and determine the range of applicability of the hypothesis of an absolutely rigid beam.

Presentation of the main research. To study the influence of factors, a physical model of the brake shoe was created – a circular beam of constant cross-section on an elastic foundation loaded with two horizontal forces (Fig. 1). The following symbols are used in this figure: h is the thickness of the lining, m; N is braking force acting on the shoe, N; R is brake rim radius, m; γ is half of arc of contact, deg.; φ is current angular coordinate, deg.

The calculation by N. S. Karpyshev's method reduces to the following formulas

$$M_t = \frac{3PR}{n};$$

$$p_{\max} = \frac{M_T}{2fBR^2 \sin(\gamma)};$$

$$N = p_{\max} BR(\gamma + 0.5 \sin(\gamma)),$$

where M_T is braking torque, N · m; $p_{\max}$ is maximum contact pressure, Pa.

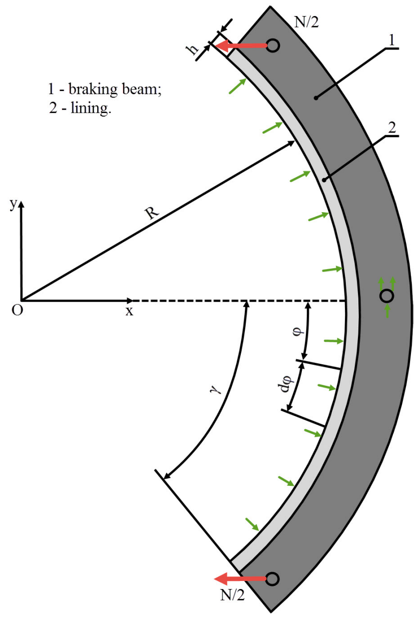

Fig. 1. Calculation model of the brake shoe: 1 — braking beam; 2 — lining

As an example, let us consider the brake shoe of a lifting machine CR-5 × 3.2/0.85 with the following parameters: R = 2480 mm; B = 400 mm is the brake rim width; $\gamma = 50°$; H = 400 mm is the brake beam height; h = = 80 mm; $E = 2.1 \times 10^{11}$ Pa is the modulus of elasticity of the beam material; $E_H = 3 \times 10^8$ Pa is the modulus of elasticity of the molding material of the lining 143–63; $P = 2.06 \times 10^5$ N is the difference in the static tension of the ropes; n = 2 is the number of brake beams; f = 0.3 is the coefficient of friction.

For this machine, using the N. S. Karpyshev's formulas, it follows that M_T = 772 kN · m, $p_{\max}$ = 0.68 MPa, N = 924.8 kN.

The following boundary conditions were used for modeling (Fig. 1):

- to the axes on the brake beam ends horizontal force $N/2$ are attached;
- the lining is fixed and allow no movements in the radial direction;
- the central axis of the beam is fixed and allow no movements in the vertical direction.

The analytical solution of this problem was obtained proceeding from the problem of the strength of a bicycle wheel, studied by F. V. Feodosiev and A. G. Zhukovskii.

Equilibrium equations used to construct a mathematical model are

$$\frac{dT}{d\varphi} + Q + \tau R = 0; \quad (1)$$

$$\frac{dQ}{d\varphi} - T - qR = 0; \quad (2)$$

$$\frac{dM}{d\varphi} + QR = 0, \quad (3)$$

where T is the longitudinal force in the beam, N; Q is the shear force, N; M is the bending moment, N·m; q is the distributed contact force, N/m; τ is the distributed frictional force, N/m.

The equation of distributed contact force is

$$q = -kw, \quad (4)$$

where w is the deflection of the beam, m; k is transverse rigidity of the lining, N/m^2

$$k = E_n \frac{B}{h}. \quad (5)$$

The equation of distributed frictional force is

$$\tau = fkw. \quad (6)$$

Hooke's law

$$M = -\frac{EI}{R}\frac{d\theta}{d\varphi}, \quad (7)$$

where EI – bending stiffness of the beam, N · m^2.

The kinematic dependence is

$$\theta = \frac{1}{R}\left(\frac{dw}{d\varphi} + v\right), \quad (8)$$

where v is the tangential displacement, m.

The inextensibility condition is

$$w = \frac{dv}{d\varphi}. \quad (9)$$

Substituting expressions (4—9) into equations (1—3), we obtain

$$\frac{EI}{R^3}\left(\frac{d^5w}{d\varphi^5}+\frac{d^3w}{d\varphi^3}\right)+kR\frac{dw}{d\varphi}+\frac{EI}{R^3}\left(\frac{d^3w}{d\varphi^3}+\frac{dw}{d\varphi}\right)+$$
$$+fkRw=\frac{EI}{R^3}\left(\frac{d^6v}{d\varphi^6}+\frac{d^4v}{d\varphi^4}\right)+kR\frac{d^2v}{d\varphi^2}+ \quad (10)$$
$$+\frac{EI}{R^3}\left(\frac{d^4v}{d\varphi^4}+\frac{d^2v}{d\varphi^2}\right)+fkR\frac{dv}{d\varphi}=0.$$

We call the ratio of the lining transverse rigidity to the bending stiffness of the beam as relative rigidity

$$\lambda = \frac{E_n BR^4}{hEI}.$$

Then equation (10) takes the form

$$\frac{d^6v}{d\varphi^6}+2\frac{d^4v}{d\varphi^4}+\left(1+\lambda\right)\frac{d^2v}{d\varphi^2}+f\lambda\frac{dv}{d\varphi}=0.$$

Its characteristic equation is

$$n\,[n^5+2n^3+(1+\lambda)n+f\lambda]=0.$$

Since the influence of friction is insignificant, we will neglect it in the future. Making a replacement $n^2 = m$, we get

$$m(m^2+2m+1+\lambda)=0.$$

For the considered machine, we obtain the following roots of the given equation

$$m=\begin{cases}0\\ -1+\sqrt{-\lambda}=-1+i\cdot 12.03\\ -1-\sqrt{-\lambda}=-1-i\cdot 12.03\end{cases},$$

then

$$n=\begin{cases}0\\ 0\\ 2.35+2.56i\\ -2.35-2.56i\\ 2.35-2.56i\\ -2.35+2.56i\end{cases}.$$

Force boundary conditions are

$$M(\gamma)=0;$$

$$Q\left(\gamma\right)=\frac{-N}{2}\cos\left(\gamma\right);$$

$$T\left(\gamma\right)=\frac{N}{2}\sin\left(\gamma\right).$$

Finally, the formula for the distribution of the contact pressure along the lining will take the form

$$q\left(\lambda,\varphi\right)=\frac{k}{B}\begin{pmatrix}C_0+\mathrm{sh}\left(\alpha\varphi\right)\sin\left(\beta\varphi\right)\left(\beta C_1-\alpha C_2\right)+\\ +\mathrm{ch}\left(\alpha\varphi\right)\cos\left(\beta\varphi\right)\left(\alpha C_1+\beta C_2\right)\end{pmatrix}, \quad (7)$$

where $C_0=\sqrt{\lambda}(\mathrm{ch}\left(\alpha\gamma\right)\cos\left(\beta\gamma\right)\left(\beta C_1-\alpha C_2\right)+\mathrm{sh}\,(\alpha\gamma)\times$ $\times\sin(\beta\gamma)\,(\alpha C_1+\beta C_2))$;

$$C_1=\frac{B_1A_{22}-B_2A_{12}}{B_0};$$

$$C_2=\frac{B_2A_{11}-B_1A_{21}}{B_0};$$

$$B_0=A_{11}A_{22}-A_{12}A_{21};$$

$$B_1=\frac{N}{2kR}\sin\left(\gamma\right);$$

$$B_2=\frac{-NR^3}{2\sqrt{\lambda}EI}\cos\left(\gamma\right);$$

$$A_{11}=\beta\,\mathrm{sh}(\alpha\varphi)\,\cos(\beta\varphi)+\alpha\,\mathrm{ch}\,(\alpha\varphi)\,\sin(\beta\varphi);$$

$$A_{12}=-\alpha\,\mathrm{sh}\,(\alpha\varphi)\,\cos(\beta\varphi)+\beta\,\mathrm{ch}\,(\alpha\varphi)\,\sin(\beta\varphi);$$

$$A_{21}=\mathrm{sh}\left(\alpha\gamma\right)\cos\left(\beta\gamma\right)\left(-\sqrt{\lambda}\right)+\mathrm{ch}\left(\alpha\gamma\right)\sin\left(\beta\gamma\right);$$

$$A_{22}=\mathrm{sh}\left(\alpha\gamma\right)\cos\left(\beta\gamma\right)\left(-1\right)+\mathrm{ch}\left(\alpha\gamma\right)\sin\left(\beta\gamma\right)\left(-\sqrt{\lambda}\right);$$

$$\alpha=\sqrt{0.5\cdot\left(-1+\sqrt{1+\lambda}\right)};$$

$$\beta=\sqrt{0.5\cdot\left(1+\sqrt{1+\lambda}\right)}.$$

We call the ratio of the contact pressure to the pressure caused by the same force N acting on an absolutely rigid range of the same area ($F = BR2\gamma$) as reduced pressure $\chi(\lambda, \varphi)$. Let us construct a graph of the dependence of the distribution of the reduced pressure on the relative rigidity (Fig. 2).

Analyzing the above graph, we can conclude that the reduced pressure distribution character, depending on the relative rigidity, can be divided into two main types:

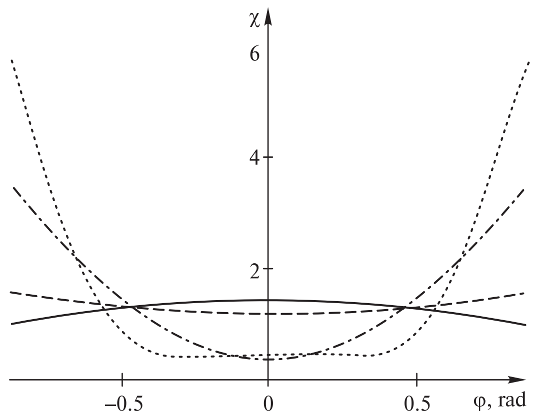

Fig. 2. Dependence of the distribution of the reduced pressure on the relative rigidity λ:
——— — λ = 1; – – – – — λ = 10; –·–·– — λ = 100; ······ — λ = 1000

1. The nature of the brake beam reduced pressure has U-shaped form with maximum values arise on the edges.

2. The distribution of the reduced pressure has a sinusoidal character with a peak in the center of the brake beam.

It follows that between these two states there is an optimum value of the relative rigidity at which the values of the reduced pressure on the edges and in the middle of the brake beam are equal to each other. To find this optimum pressure, we plot the dependence of the reduced pressure along the edges and in the middle of the brake beam on the relative rigidity (Fig. 3).

From graphs 2 and 3 it follows that for values of relative rigidity λ less than 5.3, the distribution of reduced pressure has a sinusoidal character, whereas in excess of that value the distribution has a U-shaped form. The most uniform distribution corresponds to the optimal rigidity $\lambda = 5.3$.

For the given machine from the analytical model calculation it follows

$$M_T = fR^2 \int_{-\gamma}^{\gamma} q(\varphi) d\varphi \; ;$$

$$p(\varphi) = \frac{q(\varphi)}{B};$$

$$N_x = R \int_{-\gamma}^{\gamma} q(\varphi) \cos(\varphi) d\varphi.$$

The maximum pressure p is reached at an angle equal to γ and is 1.58 MPa.

From the above formulas we obtain the following values of the parameters: M_T = 772 kN · m; $p_{\max}$ = = 1.58 MPa; N_x = 828.8 kN.

The graph (Fig. 4) compares the contact pressures between the brake rim and the lining, obtained by the method of N. S. Karpyshev and from the proposed analytical model, under the condition of the same braking torque. It is shown that N. S. Karpyshev's model assumes increased horizontal forces by 10.3 % and reduced contact pressures by 2.32 times. As follows from Fig. 2 that the range of applicability of the N. S. Karpyshev model corresponds to brake blocks with a relative rigidity less than 1.45.

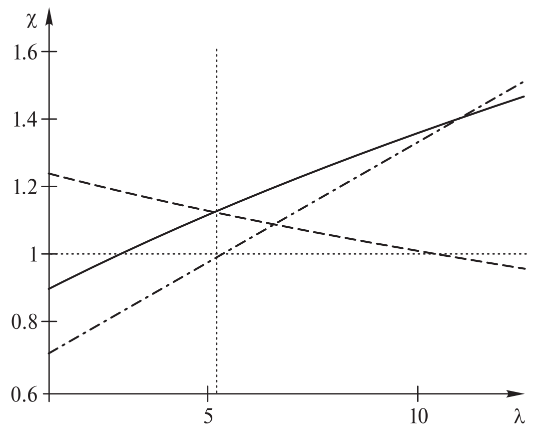

Fig. 3. Dependence of the reduced pressure on the relative rigidity:

—— — on the edge of the brake beam; ----- — in the middle; -·-·-· — their ratio

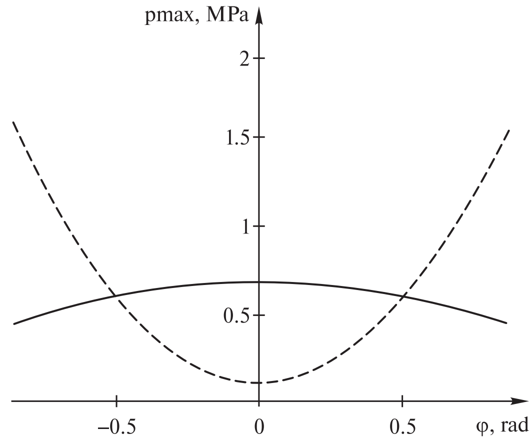

Fig. 4. Contact pressure distribution:

—— — according to N. S. Karpyshev's model; ----- — according to the developed analytical model

Fig. 5 presents a comparison between the distribution of the reduced pressure of the analytical model (curve *1*) corresponding to the relative rigidity $\lambda = 5.3$ and the distribution obtained from the computational experiment (curve *2*) for which the λ value is above the optimal one. Decreasing the relative rigidity, it was obtained that the uniform distribution, with the exception of the edge effect zones, is achieved at a value of relative rigidity $\lambda = 3.5$. The error in determining the optimal rigidity of the analytical model in comparison with the results of the computational experiment in SolidWorks Simulation does not exceed 33 %. This discrepancy in the results is due to the fact that the beam used in the analytical model was of a constant cross-sectional type, in contrast to the real beam design.

The obtained results make it possible to describe the technique for developing a brake shoe design with the

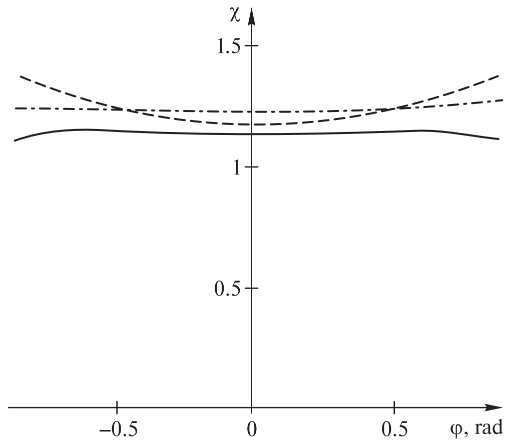

Fig. 5. Comparison of the results of the analytical solution and the computational experiment:

—— — the curve of the analytical solution for λ = 5.3; ----- — the curve of the computational experiment for λ = 5.3; -·-·-· — the curve of the computational experiment for λ = 3.5

most uniform distribution of contact stresses, consisting of the following steps:

1. Conducting a computational experiment in the SolidWorks Simulation software for a real beam design with determination of deflection and contact pressures.
2. Determining the corresponding optimal value of the relative rigidity and the coefficient *j* of its variation to obtain a uniform pressure distribution based on Fig. 3.
3. Decreasing the modulus of elasticity of the lining material by a factor of *j*.
4. Determining the corresponding distribution of the contact pressure by means of a computational experiment.
5. Developing the design of the brake shoe providing the obtained value of the relative rigidity.
6. Checking the distribution of pressure for the developed brake pad design using a computational experiment in SolidWorks Simulation and, if necessary, refinement of the value of the coefficient *j* by the iterative method for equalizing the values of contact pressures and deflections.

From the results of a computational experiment for a brake beam of a real design (Fig. 6, curve *1*) we determine that the coefficient *l*, corresponding to the ratio of the contact pressure value at the edge of the beam to the contact pressure value in the middle of the beam, is 13.65. Further from the graph shown in Fig. 3, we find that for a given value of *l* there corresponds a value of relative rigidity $\lambda = 149$. As it was indicated earlier the optimal value of the relative rigidity $\lambda = 5.3$, so for an optimal distribution of contact pressures, it is necessary to reduce the relative rigidity in $j = 28$ times. To this end, the modulus of elasticity of the lining material was reduced by a factor of 28 in SolidWorks Simulation, and then a computational experiment was repeated (Fig. 6, curve *2*). After the next iteration (Fig. 6, curve *3*), the distribution of contact pressures with a deviation from uniform no more than 5 % was achieved.

There are two fundamentally different ways to constructively reduce the relative rigidity: 1) to increase the beam bending stiffness; 2) to decrease the lining transverse rigidity.

As the first method, the thickness of the brake beam was increased along the edges and radial reinforcements were added along its sides. The second method can include the addition of a gasket from the elastic material between the beam and the lining.

For this example, to determine the optimum thickness of the beam a number of computational experiments were carried out, during which the design of the brake beam of a mine hoisting machine type CR-5 × 3.2/0.85 (Fig. 7, *a*) was optimized in the following way (Fig. 7, *b*):

- radial reinforcements were added in the form of transverse stiffeners, 20 mm thick;
- the height of the beam along the edges was increased and equaled 1.5 of the height in the middle of the beam;
- between the beam and the lining there was added a layer of rubber whose thickness is commensurate to the thickness of the lining.

The addition of radial reinforcements does not have a significant effect on the distribution of contact pressures and in itself is impractical.

The addition of a gasket made of rubber, whose thickness is commensurate with the thickness of the lining, reduces the coefficient *l* by more than 2 times and is the most effective method for reducing contact pressures.

The results of the studies are given in Table.

Conclusions.

1. The nature of the contact pressures distribution depends on the relative rigidity λ and can vary from sinusoidal to U-shaped.
2. It is determined that for the mine hoisting machine CR-5 × 3.2/0.85 – the contact pressures distribu-

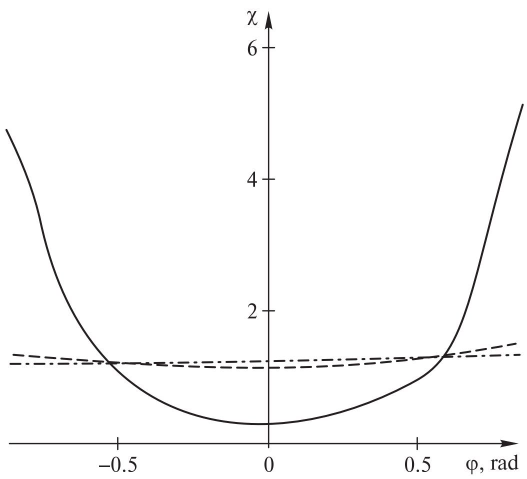

Fig. 6. Results of the computational experiment:

—— — beam of real construction $E_n = 300$ MPa, $\lambda = 149$; – – – · — beam of real construction $E_n = 10.65$ MPa, $\lambda = 7.74$; – · – · · — beam of real construction $E_n = 7.29$ MPa, $\lambda = 5.83$

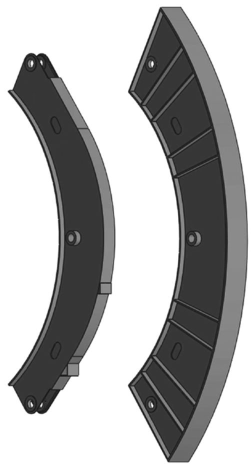

Fig. 7. Constructive solution to increase the bending stiffness of a real beam design:

a — a beam of real construction; b — a beam optimized for the structural rigidity criterion

Table

Results of simulation of constructive solutions to reduce relative rigidity

Model	Maximum contact pressure P_{max}, MPa	The ratio of the contact pressure values along the edge of the beam and in the middle of the beam *l*	Weight of the beam, tons
Real beam design	3.45	14.71	1.71
Radial reinforcement	3.06	12.82	2.23
Increase in the thickness of the beam	2.39	11.17	2.38
Adding a layer of rubber between the beam and the lining	2.02	6.75	1.71
All of the above	1.40	2.93	2.98

tion along the brake beam is U-shaped, while the calculation by Karpyshev's method shows a sinusoidal character, which understates the maximum contact pressure for this machine by 2.32 times.

3. The sinusoidal law of the contact pressures distribution is characteristic of braking devices whose value of relative rigidity is less than 1.45.

4. A technique, that makes it possible to achieve an even distribution of contact pressures between the brake rim and the lining, has been developed, the main stages of which are:

- conducting a computational experiment in the SolidWorks Simulation software for a real beam design with determination of deflection and contact pressures;
- determining the corresponding optimal value of the relative rigidity and the coefficient *j*;
- decreasing the modulus of elasticity of the lining material by a factor of *j*;
- determining the corresponding distribution of the contact pressure by means of a computational experiment;
- if necessary, refining the value of the coefficient *j* by the iterative method;
- developing the design of the brake shoe providing the obtained value of the relative rigidity.

5. To decrease the relative rigidity λ, the addition of a gasket made of an elastic material, whose thickness is commensurate with the thickness of the lining, is more effective than changing the shape of the beam.

The existing method for calculating the mine hoisting machine shoe brakes, that include the hypothesis of absolute stiffness of the brake beam and is described in the works of B. L. Davydov, N. S. Karpyshev and Z. M. Fedorova, gives a 2.32-fold lower value of the maximum contact pressures.

Application of the recommendations developed by the authors will allow reducing the maximum contact pressure in the shoe brakes of mine hoisting machines.

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Вплив жорсткості елементів колодкового гальма на розподілення контактного тиску

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Мета. Розробка рекомендацій щодо зменшення максимальних контактних напружень при взаємодії гальмівної накладки з барабаном шахтної підіймальної машини. Завданнями даної роботи є визначення можливості застосування гіпотези про абсолютну жорсткість гальмівної балки й знаходження залежності характеру розподілу контактного тиску від відношення поперечної жорсткості гальмівної накладки до згинальної жорсткості гальмівної балки.

Методика. Метод Гауса для послідовного виключення невідомих змінних; метод Ейлера для розв'язання систем диференційних рівнянь; метод Ньютона для визначення чисельних значень коренів диференціального рівняння; метод кінцевих елементів для оптимізації конструкції гальмівної балки.

Результати. Розроблена аналітична модель гальмівної балки як кругового брусу постійного поперечного перерізу на Вінклеровській основі, жорсткість якого залежить від характеристик жорсткості накладки. З аналізу напружено-деформованого стану виявлено безрозмірний фактор, що визначає характер розподілу контактного тиску – відносна жорсткість накладки. Для уточнення впливу цього фактора проведені обчислювальні експерименти у програмі SolidWorks Simulation для брусу постійного поперечного перерізу й реальної гальмівної колодки з різними конструктивними рішеннями щодо зміни відносної жорсткості. Розроблена методика визначення характеру розподілу контактних тисків у залежності від відношення поперечної жорсткості гальмівної накладки до згинальної жорсткості гальмівної балки. Представлено порівняння результатів різних конструктивних рішень задля досягненню рівномірного розподілу контактного тиску вздовж гальмівної балки.

Наукова новизна. Уперше аналітично доведено, що характер розподілу контактного тиску в колодковому гальмі шахтних підіймальних машин залежить від відношення поперечної жорсткості гальмівної накладки до згинальної жорсткості гальмівної балки, а при його зменшенні характер розподілу прагне до синусоїдального.

Практична значимість. Застосування цих рекомендацій дозволить зменшити максимальний контактний тиск у колодкових гальмах шахтних підіймальних машин.

Ключові слова: *згинальна жорсткість гальмівної балки, розподіл тиску, колодкове гальмо, SolidWorks Simulation*

Влияние жесткости элементов колодочного тормоза на распределение контактного давления

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Цель. Разработка рекомендаций по уменьшению максимальных контактных напряжений при взаимодействии тормозной накладки с барабаном шахтной подъемной машины. Задачами данной работы являются определение возможности применения гипотезы об абсолютной жесткости тормозной балки и нахождение зависимости характера распределения контактных давлений от отношения поперечной жесткости тормозной накладки к изгибной жесткости тормозной балки.

Методика. Метод Гаусса для последовательного исключения неизвестных переменных; метод Эйлера для решения систем дифференциальных уравнений; метод Ньютона для определения численных значений корней дифференциального уравнения; метод конечных элементов для оптимизации конструкции тормозной балки.

Результаты. Разработана аналитическая модель тормозной балки как кругового бруса постоянного сечения на Винклеровском основании, жесткость которого зависит от жесткостных свойств накладки. Из анализа напряженно-деформированного состояния выявлен безразмерный фактор, определяющий характер распределения контактного давления – относительная жесткость накладки. Для уточнения влияния этого фактора проведены вычислительные эксперименты в программе SolidWorks Simulation для бруса постоянного сечения и реальной тормозной колодки для различных конструктивных решений по изменению относительной жесткости. Разработана методика определения характера распределения контактных давлений в зависимости от отношения поперечной жесткости тормозной накладки к изгибной жесткости тормозной балки. Представлено сравнение результатов различных конструктивных подходов для достижения равномерного распределения контактного давления вдоль тормозной балки.

Научная новизна. Впервые аналитически доказано, что характер распределения контактного давления в колодочном тормозе шахтных подъемных машин зависит от отношения поперечной жесткости тормозной накладки к изгибной жесткости тормозной балки, а при его уменьшении характер распределения стремится к синусоидальному.

Практическая значимость. Применение этих рекомендаций позволит уменьшить максимальное контактное давление в колодочных тормозах шахтных подъемных машин.

Ключевые слова: *изгибная жесткость тормозной балки, распределение давления, колодочный тормоз, SolidWorks Simulation*

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