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Stabilization of operating parameters of friction pairs in brake devices

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Abstract. Problem. *Theoretical and experimental studies of the dynamic and thermal loading of friction pairs of brake devices from the point of view of assessing the stability and stability of their operating parameters allowed us to propose the following. The efficiency of the friction unit of band-pad brakes by stabilizing the load of the friction unit based on modeling its stress-strain state and wear in the modes of lowering the drill string and drilling, as well as substantiation of methods for calculating the design and operational parameters of friction pairs of brakes. However, nothing was said about the local problem of stabilizing the operating parameters of friction pairs of brakes.* ***Goal.*** *The purpose of the development is to substantiate the method of generalizing the operating parameters of brake friction pairs for preliminary prediction of fluctuations in their stabilization and stability values. Methodology. Analytical methods were used to study the dynamic and energy load of friction pairs in disc-pad brakes. The efficiency of the brakes and the stabilization of their operating parameters were determined.* ***Results.*** *Based on the elements of the theory of thermal similarity, a method of basic generalized parameters of friction pairs is proposed, relating to their average, maximum and minimum values regarding their stabilization, oscillations and stability, a clear sequence and relationship of the operating parameters of friction pairs of braking devices is determined, temperature gradients of metal friction elements are estimated and their permissible values for calculating heating and forced cooling rates are determined.* ***Originality.*** *The proposed method of generalized operating parameters of brake friction pairs allows predicting them and establishing their oscillations, stability, and elasticity.* ***Practical value.*** *Using this method will increase the efficiency of friction pairs of disc-pad brake devices, as well as improve their wear resistance and frictional properties during cyclic braking.*

Keywords: *brake devices, friction pairs, operating parameters, vibrations, stability.*

Introduction and Analysis of publications

The assessment of the energy levels of the contact spots of macroprotrusions of brake friction pairs and the stabilization of their operational parameters should be considered in separate areas of frictional interaction. In this case, the method of generalized operational parameters, which include two or more components, will be used. It should be borne in mind that the more components are included in the generalized parameter, the more unstable it is. This is influenced by the sequence of determining the parameters in the full-scale and computational experiment, as well as their relationship.

Modern research on rolling stock brake mechanisms is divided into several areas. It is important to highlight the achievements of scientists in the field of determining the design parameters of the working surfaces of brake discs and drums. The main design parameters are the moment of inertia and the outer diameter, as well as the average radius of the friction zone of the metal friction element. At the same time, the main operational parameters are the surface temperature, the dynamic coefficient of friction and the braking torque of the brake friction pairs [1]. The proposed relationships in dependence [1] are incorrect, since they relate to different operating conditions of brake discs.

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In [2] it is stated that the heat transfer from the rough surface of the brake disc in comparison with the polished surface of the friction belt will be higher, because in the process of flowing around the irregularities of the macroprotrusions, the air flows will have shorter turbulization path lengths than on a polished surface. To estimate the aerodynamic and thermal losses in the friction pair, the dependence

$$\Pi = \frac{Nu_m \varepsilon_p}{Nu_p \varepsilon_m}, \qquad (1)$$

where Nu_m, Nu_p – Nusselt criteria for matte and polished surfaces of the ventilation effect of the brake disc, respectively ε_m, ε_z – local drag coefficients for matte and polished brake disc ventilation surfaces.

A number of generalized operational parameters were proposed in [3] and it was noted that a significant change in the braking torque of friction pairs relative to its average value leads to variability in the braking time of other indicators. At the same time, the generalized parameter was not considered separately.

In addition, the magnitude of the braking torque is due to a significant spread of its values due to the dispersion of the main dimensions and power characteristics of the brake elements, as well as the lack of simple and sufficiently accurate means of controlling the braking torque by operators. The efficiency of the friction unit of the band-pad brakes by stabilizing the load on the friction unit based on modeling its stress-strain state and wear in the modes of lowering the drill string and drilling, as well as substantiation of the methods for calculating the design and operational parameters of the brake friction pairs. However, nothing was said about the local problem of stabilizing the operational parameters of the brake friction pairs.

Purpose and Tasks

The purpose of this development is to substantiate a method for generalizing the operational parameters of brake friction pairs for preliminary prediction of fluctuations in their stabilization and stability values.

To achieve the set goal, it is necessary to analyze the dynamic and energy load of brake friction pairs and their efficiency, as well as the stabilization of the operational parameters of brake friction pairs.

Dynamic load of brake friction pairs

The effectiveness of the friction pairs of brake devices depends on the stabilization and oscillation (fluctuation) of operating parameters during aperiodic and long-term braking modes. Table 1 shows the generalized operating parameters of the friction pairs of brake devices.

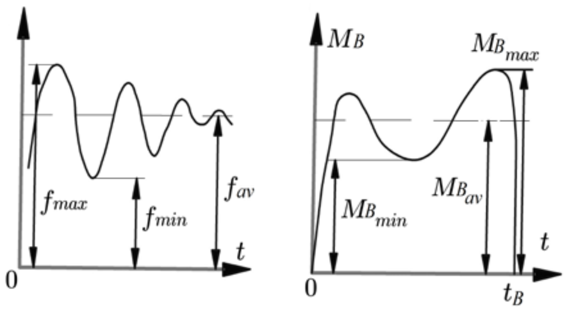
Fig. 1 shows the wave (damping) and increasing nature of the main operating parameters of brakes.

Table 1. Generalized operational parameters of friction pairs of brake devices

Coefficients	stability / oscillation
	impulse compressive normal forces
	$\alpha_{st} = N_{cp}/N_{max}$; (2) / $\alpha_k = N_{min}/N_{max}$; (3)
	dynamic friction coefficients
	$\alpha_{st} = f_{av}/f_{max}$; (4) / $\alpha_k = f_{min}/f_{max}$; (5)
	time of aperiodic and long braking
	$\alpha_{st} = \tau_{av}/\tau_{max}$; (6) / $\alpha_k = \gamma_{min}/\gamma_{max}$; (7)
	surface temperatures
	$\alpha_{st} = \vartheta_{av}/\vartheta_{max}$; (8) / $\alpha_k = \vartheta_{min}/\vartheta_{max}$; (9)
	braking torques:
	$\alpha_{st} = \frac{M_{av}}{M_{max}}$; (10) / $\alpha_k = \frac{M_{min}}{M_{max}}$; (11)
	linear wear
	$\alpha_{st} = h_{av}/h_{max}$; (12) / $\alpha_k = h_{min}/h_{max}$; (13)

where: indices mean average, maximum and minimum operating parameter

Fig. 2 shows a diagram for assessing the dynamic loading of friction pairs of brake devices.



a b

Fig. 1. Wave nature of the change: *a* - dynamic friction coefficient from the surface temperature of friction pairs; *b* - braking torque from braking time

Let us dwell on the definition of the terms of stabilization and fluctuation of the operational parameters of brake friction pairs.

Stabilization is the temporary maintenance of the constancy of the operational parameter of brake friction pairs at a certain energy level.

Oscillation - movement/change of an operational parameter at a given energy level with the possibility of its repetition at another energy level.

Stability refers to the stability of the operating parameters of different types of friction pairs during their oscillations.

Based on the initial data on the moment of inertia of metal friction elements, the friction force in friction pairs was determined, and then the operational parameters were given (Table 1), which allowed us to further construct graphical patterns (Fig. 3 and Fig. 4).

Analysis of the latter allowed us to establish the following:

- functional dependence $j = \mu(F_{fr}, \omega)$ indicates an increase in all parameters with an increase in the friction path;

- functional dependence $j = \mu(f, \omega)$ indicates an increase in all parameters in the case of an increase in impulse normal forces; in this case, the dynamic coefficient was not determined by calculation, but was specified.

Similarly, the functional dependence of the form was determined $M = \mu(N,f)$.

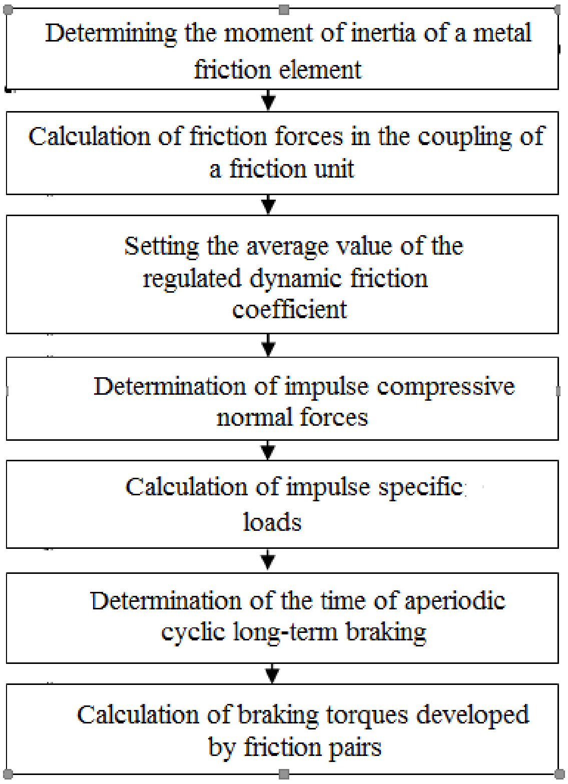


Fig. 2. Scheme for assessing the dynamic loading of friction pairs of brake devices

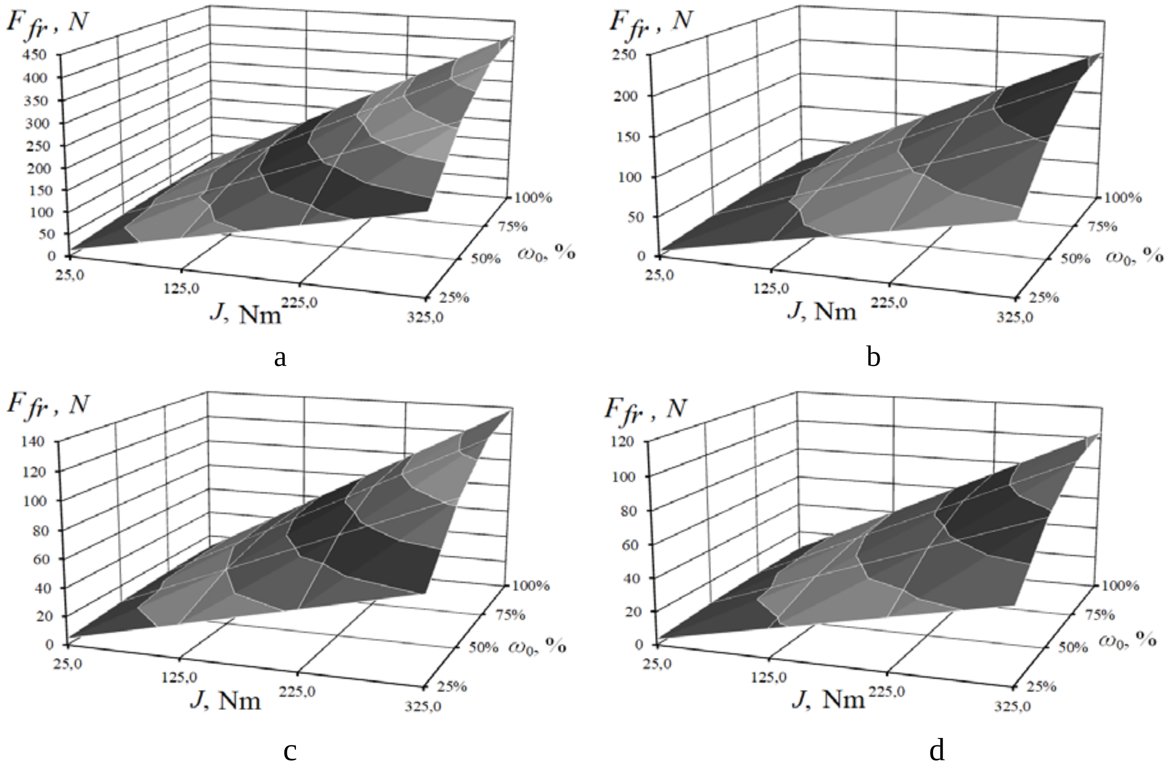


Fig. 3. Dependence of the friction force on the moment of inertia J and the decrease in the angular velocity of rotation ω_0 (% of the initial value ω) of a metal friction element at different values of the friction path S_0: $a - S_0 = 2.5$ m; $b - S_0 = 5.0$ m; $c - S_0 = 7.5$ m; $d - S_0 = 10.0$ m

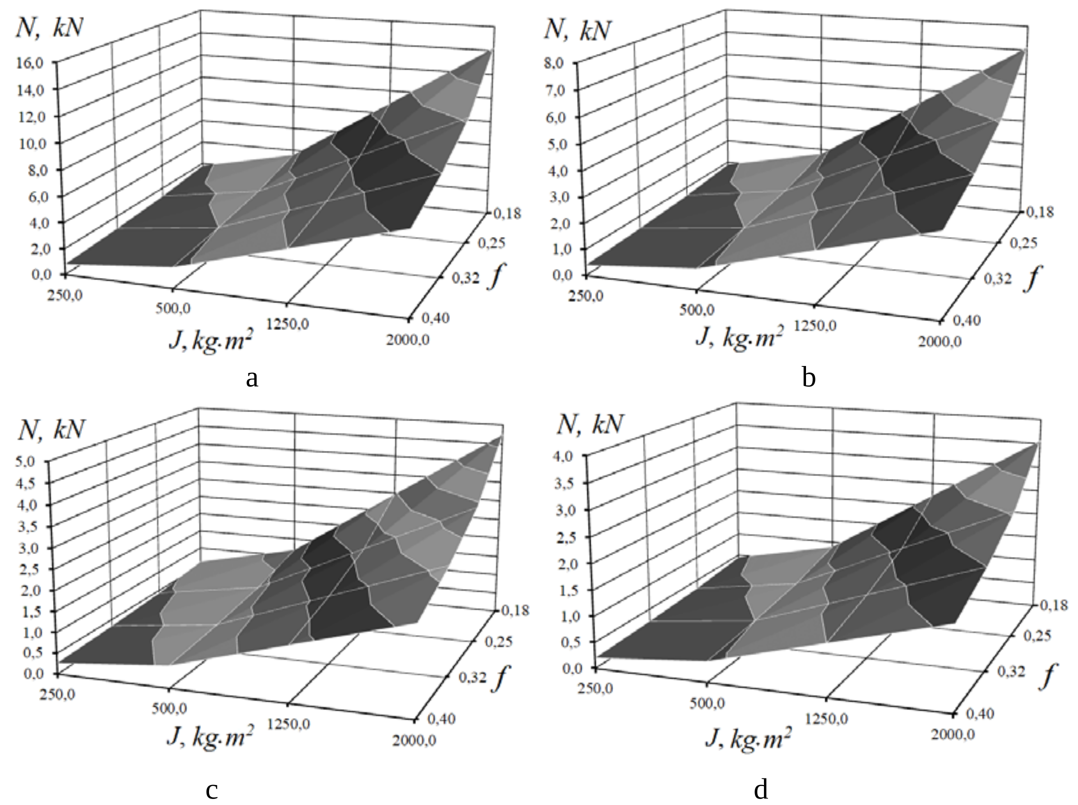


Fig. 4. Dependence of impulse normal forces N on the moment of inertia J of the metal element and the dynamic friction coefficient f of the drum-pad brake at different values of the friction path S_0: a – S_0 = 2.5 m; b – S_0 = 5.0 m; c – S_0 = 7.5 m; d – S_0 = 10.0 m

Energy load of brake friction pairs

The level of energy loading of metal-polymer friction pairs of tribosystems depends on their surface and volume temperature gradients. Let us dwell on the surface and volume temperature gradients of different types of brake discs, drums and pulleys according to Table 2. We will analyze the surface and volume temperature gradients of solid and self-ventilated brake discs during the preliminary stages of type *I* and *II* tests of bus disc-pad brakes BAZ A079.33. In the first type of tests, there was a pulsed supply of heat to the friction pairs through the interaction of the contact spots of their microprotrusions, and in the second type of tests, there was a long-term supply of heat.

The condition for the occurrence of temperature gradients in a brake disc of any type is the observance of inequality $\Delta t_S > \Delta t_V$ (the increase in surface temperature is higher than the bulk temperature). If this inequality is not observed, inversion of heat flows from the body of the brake disc to its working surfaces is possible. From Table 2 it follows that surface temperature gradients are always greater than volume temperature gradients in brake discs of different types [4].

This is especially noticeable when pulsed heat is supplied to the brake friction pairs [5].

Pulsed heat supply to the surface of a self-ventilated brake disc through its variable cross-section causes local heating of the friction raceway, which leads to the formation of microcracks on the surface. The latter are caused by aperiodic cycles of "heating (expansion) - forced cooling (compression) of the surface and subsurface layers on both sides of the brake disc. In addition, pulsed heat supply is a thermal shock, which contributes to the stabilization of the thermomechanical properties of the working surfaces of the brake pad linings.

When designing different types of discs for vehicle categories, calculations must be carried out not only to determine the design and weight parameters, but also to take into account the energy consumption of their friction belts. A feature of the band-pad brake drilling winch is that no regulated tests have been developed for it, and therefore, based on selected data on the energy load of friction units, we will show the change in surface and volume temperature gradients of their elements. In pulsed and long-term modes of supplying heat to the contact zone of metal-polymer friction pairs of the band-pad brake, the heating rate of the surfaces causes a change in thermal gradients in their thickness as follows:

brake pulley rim (fig. 5 *a*, *b*):

- in pulsed mode, heat supply over time $(0{,}2 – 1{,}4)\cdot10^{-4}$ *s* and temperature differences (10 – 15 ºC)· and change a_p from $8{,}7\cdot10^{-6}$ to $1{,}08\cdot10^{-5}$ m/s^2 temperature gradients were equal ($1{,}4\cdot10^3$ - $2{,}4\cdot10^3$ *°C/sm*) heating varied from $16{,}706\cdot10^3$ to $11{,}933\cdot10^3$ *°C/s*, that is, it decreased with increasing time of action of the thermal pulse; at the same time, the heating rate of the pulley rim is the same in magnitude as the heating rate of the friction lining in the case when there is a discrete nature, that is, the interaction of microprotrusions of friction pairs;

Table 2. Brake discs, drums and pulleys of various types with their temperature gradients during heat application

Type of brake device	Condition of metal friction elements				
Disk-pad	ventilated			solid	
	at the beginning		at the end	at the beginning	at the end
	braking				
	with temperature gradients:				
		voluminous	superficial	voluminous	superficial
	*I**	$\frac{\partial t}{\partial\delta}\geq5.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq20.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq2.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq10.0\frac{°C}{mm}$
	*II**	$\frac{\partial t}{\partial}\geq2.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq10.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq1.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq5.0\frac{°C}{mm}$
Drum-pad	steel			cast iron	
	at the beginning		at the end	at the beginning	at the end
	braking				
	with temperature gradients:				
		voluminous	superficial	voluminous	superficial
	*I**	$\frac{\partial t}{\partial\delta}\geq8.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq30.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq5.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq18.0\frac{°C}{mm}$
	*II**	$\frac{\partial t}{\partial\delta}\geq4.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq21.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq3.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq9.5\frac{°C}{mm}$
Band-pad	steel 14KhG2NML			steel 60 G	
	at the beginning		at the end	at the beginning	at the end
	braking				
	with temperature gradients:				
		voluminous	superficial	voluminous	superficial
	*I**	$\frac{\partial t}{\partial\delta}\geq9.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq38.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq6.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq32.0\frac{°C}{mm}$
	*II**	$\frac{\partial t}{\partial\delta}\geq6.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq22.0\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq4.5\frac{°C}{mm}$	$\frac{\partial t}{\partial\delta}\geq18.0\frac{°C}{mm}$

*Note: I, II – pulsed and long-term heat supply to the brake disc body was carried out provided that the friction pairs reached a temperature of t_S = 100 ºC.

Friction lining (fig. 5 *c*, *d*):

- in pulsed mode, heat supply over time $(0{,}2 - 1{,}4)\cdot10^{-4}$ *s* and by temperature difference (10-15 °C) and change a_p from 6,0 to 2,0 · 10-7 m^2/s and from $16{,}706\cdot10^{-3}$ to $11{,}933\cdot10^{-3}$ *ºC/s*, that is, it decreased with increasing pulsed thermal current exposure time [6];

- with prolonged heat supply mode for a period of time (2,0-14,0) *s* with temperature difference (10-15 ºC) and change a_p from 6,0 to $2{,}0\cdot10^{-7}$ m^2/s temperature gradients are equal (53,0 – 170 *ºC/sm*), while the heating rate varies from 1,67 to 1,19 *°C/s*, that is, it remained almost quasi-stable, despite the fact that the time of action of the heat flux increased by a factor of 7.0;

- with prolonged heat supply for a period of time 2 – 14 *s* by temperature difference 10-15*ºC* and change a_p from $8{,}7\cdot10^{-6}$ to $1{,}08\cdot10^{5}$ m^2/s temperature gradients are equal 13,87 - 23,5 ºC/sm, while the heating rate varied from 1,67 to $1{,}08\ 10^{-5}$ m^2/s and became almost quasi-stable.

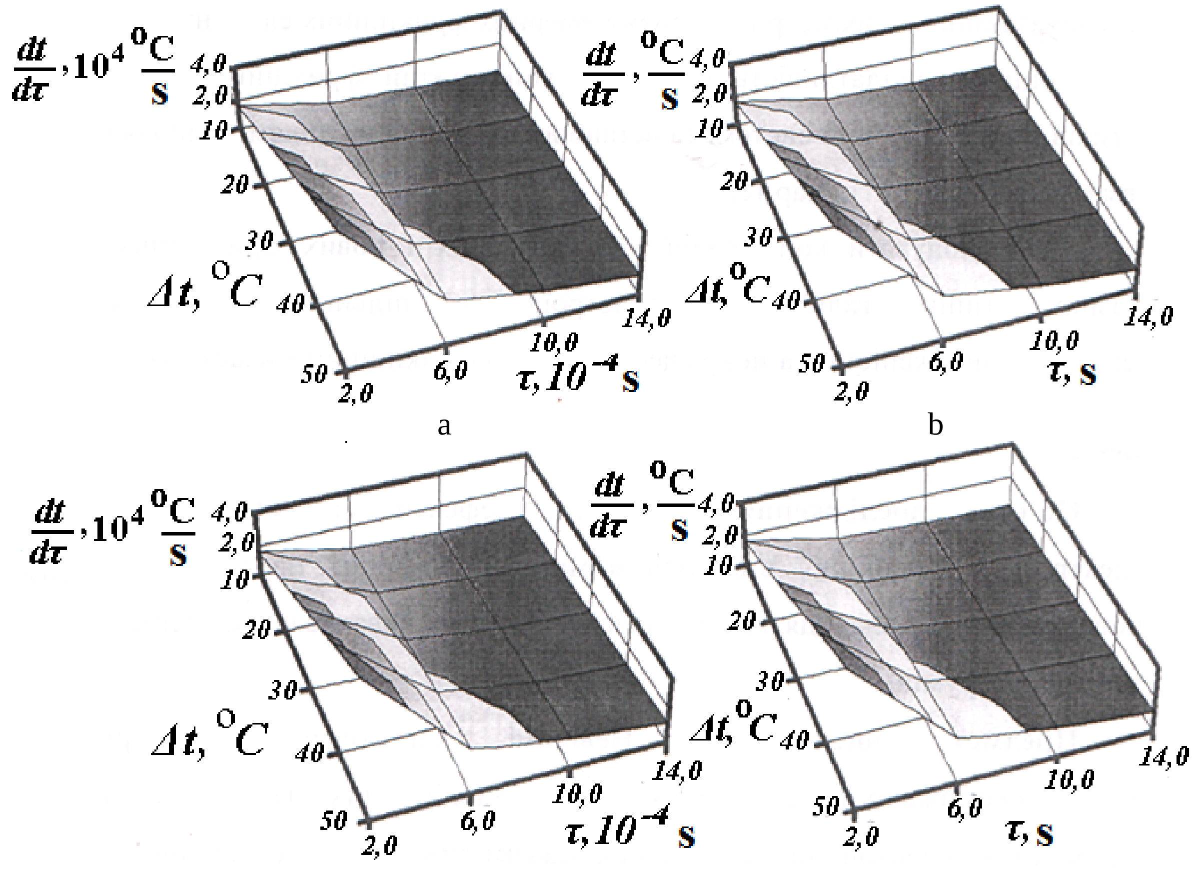


Fig. 5. Regularities of the heating rate change - materials of the pulley rim (*a*, *b*) and the friction lining (*c*, *d*) depending on the temperature difference (Δt) and the time of heat supply in the modes: a, c - pulse [τ = (2.0 - 14.0) $\cdot 10^{-4}$, s]; b, d - long-term [τ = (2.0 - 14.0), s] friction

Based on the calculated and experimental data on the energy load of non-solid and massive metal friction elements of brakes, a number of parameters in the established intervals of their change are proposed for practical application: temperature higher than the bulk temperature. In case of non-observance of this inequality, inversion of heat flows from the body of the brake disc to its working surfaces is possible. From Table 2 it follows that surface temperature gradients are always greater than bulk temperature gradients in brake discs of different types [5].

[10,0 < q < 500,0], kW/m^2; [25,0 < α < 500,0], W/(m$^2 \cdot$ºC); [0,01 < δ < 0,05], m; [30,5 < K < 135,0], W/(m$^2 \cdot$ºC);

From the given values of the parameters it follows that they depend on the density of the heat flux penetrating the rims of the pulley and drum, as well as the body of the disk.

Brake friction pair efficiency

Stability in a metal friction element is achieved when the heat conduction process is stable. In this case, the main parameter for stabilizing the state of the tribosystem is the temperature gradient on the working surface of the metal element and along its thickness. The first is characterized by electrothermomechanical friction friction, when the heat generated is equal to the amount of heat that is removed to the environment. In this case, a state of the set temperature is observed on the friction surfaces. The second gradient corresponds to the minimum value along the thickness of the metal friction element and causes its thermal stabilization state.

Analysis of the values of generalized operational parameters of friction pairs of brake devices (Table 3) showed:

- small-sized brakes have less stability of braking torque and its fluctuations due to a shorter interval of their change than large-sized ones;

- regarding braking efficiency, it depends on the time of frictional interaction of friction pairs and is maximum in emergency mode;

Table 3. Values of generalized operational parameters of friction pairs of brake devices

<table>
<tr><td colspan="2" rowspan="3">Calculated dependencies</td><td colspan="6">Braking devices:</td></tr>
<tr><td colspan="2">band-</td><td colspan="2">drum-</td><td colspan="2">disk -</td></tr>
<tr><td colspan="6">block</td></tr>
<tr><td rowspan="3">Coefficients</td><td>Braking torque stability $\alpha_{st} = M_{av} / M_{\max}$</td><td colspan="2">0.9</td><td colspan="2">0.835</td><td colspan="2">0.67</td></tr>
<tr><td>Braking torque fluctuations
$\alpha_{fl} = M_{\min} / M_{\max}$</td><td colspan="2">0.75</td><td colspan="2">0.6</td><td colspan="2">0.36</td></tr>
<tr><td>Braking efficiency
$\beta_{ef} = k \frac{\alpha_{st}}{\tau^2}$, where $k = 0.95\tau^{1.97}$
when braking:
emergency $\tau = 1.0$ s
short-term, $\tau = 3.0$ s
long-term, $\tau = 10.0$ s</td><td colspan="2">

0.86
0.83
0.80</td><td colspan="2">

0.79
0.77
0.74</td><td colspan="2">

0.64
0.62
0.59</td></tr>
<tr><td rowspan="2">Coefficients</td><td rowspan="2">Total efficiency of friction
pairs $\chi_{ef} = \frac{\beta_{ef}}{0{,}58h^{0{,}244}} = \frac{k \cdot \alpha_{st}}{0{,}58h^{0{,}244} \cdot \tau^2}$
when braking:
emergency $\tau = 1.0$ s
short-term, $\tau = 3.0$ s
long-term, $\tau = 10.0$ s</td><td colspan="6">Linear wear, h, mm</td></tr>
<tr><td>1.0

1.474
1.426
1.376</td><td>10.0

0.841
0.813
0.784</td><td>1.0

1.368
1.323
1.276</td><td>10.0

0.780
0.755
0.728</td><td>1.0

1.084
1.062
1.024</td><td>10.0

0.626
0.605
0.584</td></tr>
</table>

Table 4. Permissible values of operating parameters of different types of braking devices

No. p/p	Parameter names	Symbol designation	Units of measurement	Braking devices: band-block	drum-block	disk -block
1	Mass of the metal friction element	m	kg	429.0	86.6	12.3
2	Moment of inertia	I	Nm^2	150.0-850.0	80.0-400.0	25.0-180.0
3	Dynamic coefficient of friction	f	-	0.20-0.50	0.18-0.45	0.30-0.05
4	Braking torque of friction pairs	M_B	N·m	63.75-170.0	150.0-370.0	339.0-936.0
5	Surface temperatures of friction pairs	t_s	°C	350/200		
6	Temperature gradients superficial volumetric	grad$t$	°C/mm	32.0 6.5	18.0 5.5	10.0 2.5
7	Equivalent stress gradients	gradσ	MPa/mm	4.61-35.5	9.8-40.6	12.09-46.09
8	Pace: $\frac{\text{heating}}{\text{cooling}}$	$\Delta t/\Delta\tau$	°C/c	$\frac{0.05-0.280}{0.045-0.023}$;	$\frac{0.08-0.65}{0.15-0.6}$;	$\frac{0.37-0.85}{0.35-0.78}$;

- regarding braking efficiency, it depends on the time of frictional interaction of friction pairs and is maximum in emergency mode;

- the given efficiency of brake friction pairs depends on the linear wear of friction linings and time, and is maximum in emergency braking mode.

Stabilization of operating parameters of brake friction pairs

Analysis of the performed field research works [7-14] on the assessment of the operational parameters of friction pairs of small-sized and large-sized brake devices of lifting and transport equipment showed that the implementation of a significant durability resource is associated with their oscillations and stabilization. For this, it is necessary to know the permissible values of the operational parameters of different types of brake devices (see Table 4).

Analysis of the permissible values of the operational parameters of friction pairs of different types of brakes allowed us to establish the following:

- the larger the interval of change of a particular parameter, the higher the stability fluctuations;
- surface temperature (in the numerator) refers to small-sized brakes, and in the denominator - to overall brakes;
- stabilizers for metal friction elements are the heating rates, both surface and through their thickness.

When the energy load of the brake friction pairs changes, an excess entropy (dQ/dt) appears. The value of entropy must be positive for all perturbations of the operational parameters of the brake friction pairs. In addition, it is necessary to take into account the fact that in the field of linear thermodynamics of irreversible processes, the conditions of quasi-stability of the operational parameters of the brake friction pairs are fulfilled.

Discussion of results

Theoretical and experimental studies of dynamic and thermal loading of friction pairs of brake devices in terms of assessing the stability and stability of their operational parameters allowed us to propose the following:

- the method of basic generalized parameters of friction pairs, relating to their average, maximum and minimum values in terms of their stabilization, fluctuations and stability, was implemented on the basis of elements of the theory of thermal similarity;
- a clear sequence and relationship of operational parameters of friction pairs of brake devices and ensured their stability;
- to estimate the temperature gradients of metal friction elements and determine their permissible values for calculating heating rates and forced cooling.

Conclusions

Thus, according to the proposed method of generalized operational parameters of brake friction pairs, it is possible to predict them and establish their oscillations, stability and resilience.

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Стабілізація робочих параметрів пар тертя в гальмівних пристроях

Анотація. Проблема. *Теоретичні та експериментальні дослідження динамічного та теплового навантаження пар тертя гальмівних пристроїв з точки зору оцінки стійкості та стійкості їх робочих параметрів дозволили запропонувати наступне. Ефективність роботи вузла тертя стрічково-колодкових гальм шляхом стабілізації навантаження вузла тертя на основі моделювання його напружено-деформованого стану та зносу в режимах спуску бурильної колони та буріння, а також обґрунтування методів розрахунку конструктивних та експлуатаційних параметрів пар тертя гальм. Однак нічого не було сказано про локальну проблему стабілізації робочих параметрів пар тертя гальм.* ***Мета***. *Метою розробки є обґрунтування методу узагальнення робочих параметрів пар тертя гальм для попереднього прогнозування коливань їх значень стабілізації та стійкості. Методологія. Для дослідження динамічного та енергетичного навантаження пар тертя в дисково-колодкових гальмах використовувалися аналітичні методи. Визначено ефективність роботи гальм та стабілізацію їх робочих параметрів.* ***Результати***. *На основі елементів теорії теплової подібності запропоновано метод основних узагальнених параметрів пар тертя, що стосуються їх середніх, максимальних та мінімальних значень щодо їх стабілізації, коливань та стійкості, визначено чітку послідовність та взаємозв'язок робочих параметрів пар тертя гальмівних пристроїв, оцінено градієнти температур металевих елементів тертя та визначено їх допустимі значення для розрахунку швидкостей нагрівання та примусового охолодження.* ***Оригінальність****. Запропонований метод узагальнених робочих параметрів пар тертя гальм дає змогу прогнозувати їх та встановлювати їх коливання, стійкість та пружність.* ***Практична цінність****. Використання цього методу підвищить ефективність пар тертя дисково-колодкових гальмівних пристроїв, а також покращить їх зносостійкість та фрикційні властивості під час циклічного гальмування*

Ключові слова: *гальмові пристрої, пари тертя, експлуатаційні параметри, коливання, стійкість*

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