Methodology of thermal loading estimation of the brake mechanism of a truck

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Abstract. The paper presents the results, conclusions of the assessment of friction pairs in the aspect of the phenomenon of fading. As you know, most of the kinetic energy of a car with ABS is extinguished due to the work of friction in the brake mechanism. Overheating the brake mechanism, namely its friction pairs, leads to the appearance of the phenomenon of critical fading, accompanied by a sharp decrease in the braking torque. When the metal friction elements of braking devices operate in a repeated-short-term mode, it is advisable to assess their operating parameters at a steady-state thermal regime, i.e. under the most severe temperature conditions. The proposed technique makes it possible to obtain a high convergence of the calculated values with the experimental data. Evaluation of thermally loaded using this method can be carried out at any stage of the design and does not require large computing power. Visualization of the results allows you to identify areas of increased heating to assess the need to change the design of the brake mechanism.

1. Introduction
In order not to exceed the permissible values of the surface temperatures of the friction pairs, it is possible to vary the size of the heat transfer surface, the weight of the pair elements, and the heat transfer coefficient from the metal friction elements [1].

In practice, two main methods of diagnostics of brake systems are used – road and test bench. The main test requirements are set out in GOST R 41.90-99 (UNECE Regulation No. 90) "Uniform provisions concerning the approval of replacement brake lining assemblies, drum brake linings and discs and drums for powerdriven vehicles and their trailers" and UNECE Regulation No. 13 "Uniform provisions concerning the approval of vehicles of categories M, N and O with regard to braking".

The Type 0 tests are not considered in this procedure because of the single-time braking to test the thermal load.

The test procedure corresponds to the test scheme of the brake mechanism under friction heating according to RD 50-662-88 (methods for experimental estimation of the friction compatibility of materials of friction interfaces). The document established a set of methods for evaluating the friction compatibility of construction materials and lubricants used in friction interfaces. The particularity of these methods is to test the ability of this combination of materials to provide stable values of friction forces, wear intensity in a given or possibly larger range of temperatures, operating pressures and speeds.
2. Materials
For the calculation, we use the material of the brake pad lining, the friction compound 3554F from Ferrodo. The material is molded, asbestos-free with friction modifiers and reinforcing fibers in a phenolic matrix. It is characterized by stable friction characteristics, high wear resistance, providing a long service life of the brake drum in a variety of conditions.

Figure 1. Graph of dependence of the coefficient of friction on the temperature (thermal sensitivity) for the friction compound 3554F.

The brake drum material is antifriction cast iron SCH20. The thermal conductivity ranges from 54 to 45W/m·°C at a temperature range of 20-400 °C. To perform the calculation, we use the average value of the thermal conductivity coefficient of 50 W/m °C.

3. Calculation of thermal loads
Let us calculate the friction work performed per unit of time relatively the area of the brake drum for the application of thermal loads.

The approach for determining the heat flow is based on the braking work reduced to the brake drum, which can be determined through the ratio of the longitudinal reaction and the braking torque, in contrast to other approaches used in the design, previously described by the authors in [2]. Despite the conventionality of this ratio, this is enough for a relatively accurate job search. Taking into account the reaction, we obtain an expression for the braking force in the form:

$$P_b = 2 \frac{R_s \cdot \varphi_{max} \cdot \xi_{abr} \cdot r_{disc}}{r_{disc}} \tag{1}$$

To determine $\xi_{abr}$, the value of the longitudinal reaction, we will use the degree of using the maximum coefficient of adhesion $\xi_{abr}$ in the operation of the anti-lock braking system. The value is determined from the requirements of standards for the braking performance of the car with ABS. So, for dry asphalt, it should be not below the level of the braking skid $\xi_{abr} = 0.9$, wet and compact snow $\xi_{abr} = 0.85$ [19]. Introduction to the calculation of this indicator greatly simplifies the calculation process and eliminates the consideration of the dependencies of the algorithm of a particular anti-lock system, since the algorithm and the matrix of values of the control unit of the anti-lock system are often trade secret manufacturers of brake systems.

To find the full operation of friction in the brake mechanism, it is necessary to know the friction path in the ‘brake disc – pad’ pair. The theoretical background to determine the path of friction of individual wheels of the vehicle in braking mode is given in [20].

The rotation of each individual wheel of the vehicle with ABS when braking is characterized by a different combination of speed modes due to the operation of the brake system and the contact of the tire with the road surface, which is shown in Fig. 3.

At the same time, the following characteristic periods can be distinguished: braking, disinhibition, blocking or using the wheel, free rolling and quasi-uniform movement in the area $\omega_s = 0$.

Each period is characterized by a different combination of speed and load performance. Taking into account the short duration of the ABS cycle, we assume that each period is characterized by a steady slowdown (acceleration) $J_{const}$ and an average speed.
To find the general friction path in all areas, let us sum the above dependencies:

\[
L_{\text{pf}} = \sum_{i=1}^{I} L_{\text{pfTi}} + \sum_{j=1}^{J} L_{\text{pfPj}} + \sum_{n=1}^{N} L_{\text{pfDn}}
\]

(2)

where: \( L_{\text{pf}} \) – friction path of the lining of the drum; \( I \) – number of braking areas; \( J \) – number of areas of disinhibition; \( N \) – number of areas of constant rolling of the wheel;

\( L_{\text{pfTi}} \) – friction path for a pair of "disc-pad" on the braking area;

\( L_{\text{pfPj}} \) - way friction pair "disc pad" on the j-th section of the release;

\( L_{\text{pfDn}} \) - friction path for the pair "disc-pad" on the n-th section of the wheel constant rolling braking.

Finally, the friction work brought to the brake disc is determined from the expression:

\[
W_{\text{fr}} = L_{\text{pf}} P_x.
\]

(3)

**Table 1.** Initial data of the thermal model of friction pairs.

| Parameter                           | Units measuring | Value     |
|-------------------------------------|-----------------|-----------|
| Load                                | N               | 400       |
| Coefficient of friction             |                 | 0.5       |
| Friction linear velocity            | m/s             | 1.84      |
| Surface area of the rolling sample  | m³              | 6.782*10^4 |
| Power supplied to the rolling sample| W               | 368       |
| Heat flow                           | W/m²            | 542581    |

When carrying out the design calculation, the following ratio is used. In contrast to the above, this formula is simplified by using the geometric radius of the wheel, instead of the dynamic radius. Let's refine the formula in accordance with the above relations:

\[
N = R_k \cdot \varphi_{max} \cdot \frac{V_{a, max}}{a \cdot t}.
\]

(4)

\( r_k \) – rolling radius of the wheel;

\( a \) – average deceleration, acceleration at the maximum braking torque;

\( V_{a, \text{max}} \) – starting speed during braking with the maximum braking torque, type 0. For the test type 0: \( V_{a, \text{max}} = 90 \frac{\text{km}}{\text{h}} \)

The calculation result for the Type 0 tests is shown in Figure 2.

**Figure 2.** Relationship of the allocated power from the time to the moment of stopping.
Based on the obtained data, we calculate the supplied heat power or power per unit area (Table 1). Based on a number of sources, the part of the total heat power generated in the drum-pad friction pair is supplied to the drum. Moreover, 69% is brought to the drum on the basis of the article [3].

To determine the picture of the temperature fields of the brake assembly when applying thermal loads to it, the finite element method included in the SOLIDWORKS Simulation package was used. We focus on the fact that the designer for the preliminary assessment uses packages of a level similar to Simulation, which is part of the general PLM (Product Lifecycle Management) system, which, when using this technique, allows for an operational assessment of the thermal load, without resorting to higher-level packages.

![Figure 3. Model of the brake mechanism with applied thermal loads.](image)

Thermal loads and heat losses on various surfaces of the model are given, taking into account the given initial data and restrictions, for the assembly of the brake mechanism, as opposed to a single part of the brake drum in other calculation methods. We applied heat flow power to the brake drum, released between the drum and the pad.

From the surfaces of the brake drum with a radiation coefficient of 0.8, heat is transferred to the environment with a temperature of 25°C by radiation.

A heated brake drum transfers a certain amount of heat to the environment. The amount of heat according to the Newton-Richman law depends on the emissivity of the body, which is determined by the radiation coefficient. In this case, the coefficient was assumed to be 0.8 for gray cast iron based on a number of reference data and such a parameter as the degree of blackness.

Heat transfer to the environment from the surfaces of the brake drum to the environment happens by means of forced convective heat transfer. Convective heat transfer coefficient is 300 W/m²·K.

From those surfaces through which thermal energy is dissipated by radiation, energy is also dissipated by forced convection.

The brake caliper transfers heat to the environment through natural convection. Part of the surface of the shank is located under the wheel hub to simplify the model, we take the average value of the convective heat transfer coefficient.

Heat transfer from the caliper, the flap and the non-working surface of the brake pad to the environment with an air temperature of 294 K occurs with a convective heat transfer of 100 W/m²·K.

Heat transfer to the environment by radiant energy from the surface of the brake caliper and the flap with an ambient temperature of 294 K. The radiation coefficient is 0.7.

The non-working side of the brake pad transfers some of the heat to the environment through a radiation mechanism. Based on the reference data and the shade of the parts, the radiation coefficient
is set to 0.7. It is less than the value of the radiation coefficient of the brake drum, which is made of cast iron and when the surface of the sample is thermally oxidized, its color has a darker radiation spectrum compared to the steel from which the base of the brake pad.

The software automatically creates a combined grid for the solid. The accuracy of the solution depends on the quality of the grid. In general, the smaller the grid, the higher the accuracy.

A finer mesh is especially important where there are changes in temperature and strain (change by an order of magnitude). At the same time, the constructor must clearly understand where the zone is located. A smaller mesh may be required at the junctions of several load-bearing elements, in holes, in corners, contact areas, and in areas with high voltage (Fig. 4).

![Figure 4. Thickening area of the finite element grid.](image)

The Simulation package includes automatic element shape validation by the Jacobian criterion and the overall grid quality. These two metrics together are exhaustive, as they take into account the location of the mid-nodes, skewness, warping, curvature, aspect ratio, and ultimately, the ability of the element to transmit data. After constructing the grid, it is recommended to control the value of the Jacobian criterion, which directly indicates the ability of the grid to transfer data from the element space to the real one. It is recommended to avoid the appearance of sharp corners in the CE, the greatest accuracy is given by elements with the same sides or close to the same. The ratio of the maximum element size to the minimum must not exceed 2, or the angle must not be less than 30°.

When creating computational models, it is recommended to avoid the appearance of sharp corners in the FE, the greatest accuracy is given by elements with the same sides or close to the same. The ratio of the maximum element size to the minimum must not exceed 2, or the angle must not be less than 30°.

The created assembly mesh has a cathet size of 1.8 mm, the mesh density coefficient is average, and the tetrahedron aspect ratio is 1.5.

| Table 2. Information about the grid |
|------------------------------------|
| Parameter                          | Units measuring | Value       |
| Load                               | N               | 400         |
| Coefficient of friction            |                 | 0.5         |
| Friction linear velocity           | m/s             | 1.84        |
| Surface area of the rolling sample | m²              | 6.782*10⁴   |
| Power supplied to the rolling sample | W         | 368         |
| Heat flow                          | W/m²            | 542581      |

4. Calculation of thermal loads
The result of the study was a plot of the temperature field of the assembly unit, which gives an exhaustive picture of the temperature of the parts. The main interest is the maximum surface temperature of the brake drum. Namely, the maximum temperature and the surface on which the
points with the maximum temperature are located. The extremum is located on the working surface of the drum, the area of contact with the pad [4]. The temperature of the remaining layers of the sample gradually decreases as the contact surface is removed. Moreover, there is a sharp decrease in the surface temperature of the caliper in comparison with the mobile sample.

![Figure 5. Thickening area of the finite element grid.](image)

The following conclusions can be drawn from the analysis of the simulation results.

1. The uneven distribution of the temperature of the brake mechanism parts over the depth and surface leads to the occurrence of temperature stresses in the parts that are cyclical in nature, and as practice shows, cause deformations of the parts.

2. The factor that has a decisive influence on the nature of the formation of the temperature field of the brake parts is the convective heat transfer that occurs when the parts forming the friction pair are relatively sliding.

3. In brake designs with incomplete overlap of the brake drum with the brake pad, there is an intermittent supply of heat to the brake drum. This causes temperature fluctuations on the surface of the brake drum, the range of which reaches several tens of degrees. The range of these vibrations quickly attenuates at a distance from the surface of the drum and at a depth of more than 2 mm is minimal.

4. Temperature fluctuations on the surface of the drum with a span of several tens of degrees lead to stress fluctuations in the surface layers of the brake drum with a span of hundreds of megapascals. High-frequency voltage fluctuations increase the load on the brake parts.

5. **Experimental approach**

As a rotating sample, we used disks with a diameter of 36 mm and a thickness of 6 mm obtained from the original brake disc of a car by waterjet and turning. This made it possible to obtain the roughness and accuracy of the test surface comparable to the working surface of the original brake disc and drum.

One of the factors that confirms the adequacy of the tests carried out is what are the materials of friction pairs. A chemical analysis of the data of brake discs was carried out, which showed full compliance of the chemical composition with the requirements of the regulatory documentation for this brand of cast iron, all parameters were in the middle of the allowable ranges (Fig. 6).
A movable sample is a disk that rotates at a constant speed of 1000 rpm (16.6 s⁻¹) [5]. A sample fixed in the carriage is pressed against it, which is a rectangular segment of the brake pad. In the experiment, two manufacturers of brake pads were used slightly below the average (Riginal company) and upper price segment with verified authentication (Ferrodo company).

The heat load of the friction pairs during the test was estimated using the SAT HotFind-L [5] thermal imager. The surface temperature of the moving sample is of great interest, then the imager was tuned specifically for it. The emissivity correction coefficient is chosen to be 0.7, which, according to the documentation for the thermal imager, corresponds to processed gray cast iron (Fig. 7).

A comparison of the experimental and calculated values of the maximum temperatures of the moving sample indicates a minimum deviation of the two values. A more detailed comparison of the experimental data with the calculated ones is presented [5]. It is worth paying attention to the fact that the region with the maximum temperature is on the same surfaces in the calculation model and the experimental sample.

Figure 6. A frame of thermal video received by the SAT HotFind-L thermal imager: 1 - temperature at the center point of the video detector; 4 - maximum temperature in the captured area.

Figure 7. The maximum temperature values of moving samples during 5 minutes for a combination of different moving and fixed samples.
6. Conclusion
The given method of assessing the heat load helps to quickly assess the temperature field in the
friction pairs and take it into account when selecting materials and their heat treatment in the design
process. In addition, having a curve of the coefficient of friction of the friction mixture of the brake
lining, you can compare its values with the results of modeling.

References
[1] Chichinadze A V 1970 *Thermal dynamics of friction* (Moscow: Mashinostroenie)
[2] Voloaca S, Fratila G 2012 Concerns regarding temperature distribution obtained by experiments
and finite element analyses for types of brake discs *U.P.B. Sci. Bull., Series D* 74(3) 33
[3] Pershin V K, Fishbejn L A 2005 Simulation of thermal conditions in the friction interaction of
the wheel and brake pad *Trasport Urala* 4 34–44
[4] Pinca-Bretotean C, Bhandari R, Sharma C, Dhakad S, Cosmin P, Kumar A 2021 An
investigation of thermal behaviour of brake disk pad assembly with Ansys *Materials Today:
Proceedings* 5 58
[5] Dygal V G, Zhukov I S 2021 Experimental and Theoretical Approach for Evaluation of
Thermal Loading of Car Brake Discs *Proceedings of the 6th International Conference on
Industrial Engineering (ICIE 2020). ICIE 2021. Lecture Notes in Mechanical Engineering* (Cham: Springer) pp 61–68