Study of the influence of the brake shoe temperature and wheel tread on braking effectiveness

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Abstract. This article considers the coefficient of friction between the brake shoe and the wheel of the railway rolling stock. It conducts a review of the works on the study of the interaction of the brake shoe and wheel, considers a method for obtaining the calculated value of the friction coefficient used in the Russian Federation and in many countries, and indicates a drawback of this method that leads to errors in the calculations. The paper specially focuses on the study of heating of brake shoes during braking as a factor in changing the coefficient of friction between the wheel and the brake shoe. The FEM model of heating the brake shoes during braking is constructed taking into account changes in the mass and shape of the brake shoes during wear. Brake shoe wear is considered as a factor in changing the effectiveness of train brakes.

1. Introduction

Transport is one of the key industries of any state, the most important component of the effective functioning of the economy. The development of the transport system of the Russian Federation is aimed at more fully providing the needs of the economy and the population of the country with transport services.

At the same time, the velocity of trains is limited by the possibility of its safe stopping. Thus, in railway transport the effectiveness of train auto-brakes is the most important factor determining the efficiency and cost of the transportation process. This dependence is well seen in the Rules for maintenance of brake equipment and brake control of railway rolling stock. It should be noted that with an increase in braking effectiveness, there is not only an improvement in quality indicators, but also an increase in the safety of freight and passenger traffic. Also today, the urgent issue is the exact modeling of train movement processes to create diagnostic systems and automatic train control to increase labor productivity. All of this should ultimately increase the average and maximum train speeds.

The permissible train velocity is confined to its ability to stop at a safe distance in case of emergency. The stopping distance is determined by the ratio of the total braking force developed by the brake shoes and the mass of the train. The relationship between the total braking force developed by the brake shoes and the mass of the train is called the braking efficiency, see expression (1)

$$\theta = \frac{\sum T}{M},$$

where $\sum T$ is the total force of all brake shoes that press the wheels, kN;
$M$ is the mass of the train, kg.

In the computational models of the train’s movement, the train stopping distance is considered a value that depends on the brake shoe pressure on the wheels, the train mass and the train velocity. However, experimental operation of trains shows that trains with the same braking effectiveness, the same braking pressure and braking velocity, have different train stopping distances.

Considering that the types of brake shoes and wheels for the trains under consideration are the same, and the same is the pressing of the brake shoes on the wheel, we can conclude that the "shoe-wheel" coefficient of friction depends on an additional factor. So, study [1] considers the effect of thermoplastic deformations in the process of tribological interaction in the process of braking. Studies [1, 2] deal with the influence of the emergence of a third body in the "shoe-wheel" area of friction due to particles being separated during wear. But the vast majority of authors study the temperature dynamics in the "shoe-wheel" or "shoe-disk" system [3, 4, 5, 6, 7]. Most often they study the temperature dynamics as a factor affecting the destruction of materials. Works [8, 9] explore the direct influence of the brake shoe temperature on the adhesion force for a disc brake and consider the temperature distribution on the disk. Work [10] consider the experimental studies of the braking process of freight trains of the Italian Railways for various types of brake shoes. The use of shoe brakes affects the temperature of the wheel taping line. Changing the taping line temperature also affects the adhesion of the wheel to the rail. A number of works [11, 12, 13, 14] deal with the study of this issue.

From the studies considered above, we can unambiguously conclude that the friction coefficient significantly depends on temperature in the area of the "shoe-wheel" contact, and therefore can affect the train stopping distance. The temperature especially affects the braking force during prolonged braking on slopes to control or maintain velocity.

The coefficients used today in calculating the stopping distance of trains are determined experimentally. This method of determination includes all factors, including the temperature of the brake shoes, which changes during braking. But then why does discrepancy occur between the calculated braking and braking in actual use?

In order to determine how the temperature is taken into account in the coefficient of friction obtained by this method, it is necessary to consider this method in more detail.

According to the method, the braking force is determined by the formula:

$$B_{st} = \varphi \cdot T,$$

where $\varphi$ is the coefficient of friction;

$T$ is the force of the brake shoe pressing to the wheel tread, kN.

From this formula, it follows that the determining coefficient of braking effectiveness is the coefficient of friction. A number of fundamental works on the determination of "shoe-wheel" friction coefficients of railway rolling stock and the study of train braking features are presented in [15].

To experimentally determine the coefficient of friction, a "Throwing" method is used. This method involves the forced uncoupling of the test unit, which in turn performs self-braking with a certain force of pressing of the brake shoe on the wheel (Figure 1). The next experiment of this kind is carried out with a different pressing force. Next, we construct a curve velocity versus time or velocity versus distance and differentiate it at given intervals with step 2 km/h.
Figure 1. The methodology for determining the coefficient of friction of the brake shoe experimentally.

At each interval, there is a change in velocity (3), the time of the interval (4), and then the deceleration acceleration in the interval (5)

\[ \Delta V_i = V_i - V_{i+1}, \]  

where \( V_{i+1} \) is the velocity at the end of the interval, km/h; \( V_i \) is the velocity at the beginning of the interval, km/h.

\[ \Delta t_i = t_{i+1} - t_i, \]  

where \( t_{i+1} \) is the time at the end of the interval, s; \( t_i \) is the time at the beginning of the interval, s.

\[ a_i = \frac{\Delta V_i}{\Delta t_i}, \]  

The braking force can be found by two formulas:

\[ B_{sti} = T \cdot \varphi_i, \]  

where \( T \) is the force of the brake shoe pressure on the wheel, \( T = \text{const}, \) kN; \( \varphi_i \) is the coefficient of friction on the \( i \)th interval.

\[ B_{sti} = m \cdot a_i, \]  

where \( m \) is the mass of the tested unit, t.

Therefore,

\[ \varphi_i = \frac{m a_i}{T} \]
Thus, the value of the coefficients obtained for each interval is approximated into a dependence, which is the formula for finding the coefficient.

The coefficient of friction determined by the experimental method for cast-iron and composite brake shoes can be represented as formulas (9, 10) or as graphs (Figure 2), which are also given in the Rules of Traction Calculations

\[
\varphi_{ci} = 0.6 \cdot \frac{1.6T + 100}{8T + 100} \cdot \frac{\theta + 100}{5\theta + 100},
\]

\[
\varphi_{com} = 0.44 \cdot \frac{0.17 + 20}{0.47 + 20} \cdot \frac{\theta + 150}{2\theta + 150}.
\]

![Figure 2. Friction coefficient dependence versus velocity of train and pressure force for the cast-iron brake shoe and composite brake shoe.](image)

After considering this method of experimentally obtaining the friction coefficient, one can make several conclusions:

1) the coefficient obtained reflects not only the dependence of the friction force between the shoe and the wheel, but also the inertia forces of the rotating masses stored in the wheelsets, which make the graph of the coefficient obtained smoother. In experiments conducted on benches whose inertia of rubbing elements wasn't high, the graph of the friction coefficient was sharper and more complex [8];

2) the temperature of the brake shoes is taken into account in this coefficient, but only for cases of braking, in which the velocity is constantly reduced, and the train tends to stop. It is impossible to use this coefficient with certainty to simulate long-term braking at a constant velocity, since the brake shoe temperature will begin to change and despite the constant value of pressing and velocity, the friction coefficient will begin to change significantly according to [8];

3) with increase in pressing force, the friction coefficient decreases. This is strange, because with increasing pressure, the degree of diffusion between the shoe and the wheel should go up. According to [16, 17], with an increase in the pressing force, the braking force builds up, which means that the mechanical braking power increases. During frictional braking, all mechanical power is converted to heat and, as a result, an increase in pressure leads to an increase in temperature. An increase in temperature leads to a decrease in the coefficient of friction between the shoe and the wheel;
4) to more accurately simulate the process of the train braking, it is necessary to conduct additional studies of the heating of the shoes and wheels during braking, depending on the braking duration. It is important to note that the heating rate of the shoe and the maximum temperature during braking will depend not only on the dissipated heat output, but also on the overall heat capacity of the shoe and on the area of heat exchange between the shoe and the environment. The shoe loses about 80% of the initial mass and 25% of the heat exchange area with the environment during wear, which significantly changes the energy balance and temperature in the friction zone, and hence, it changes the friction coefficient and braking force. Thus, it can be explained why trains with the same braking pressure and other equal braking conditions in operation have different train stopping distances.

2. Materials and Methods
In the framework of this article, we studied the dynamics of temperature, as well as the distribution of the temperature field at various values of heat capacity and heat transfer of the brake shoe. The heat capacity and heat transfer change due to changes in the geometry of the shoes during wear while in operation.

To study these parameters, mathematical modeling of the braking process using the finite element method (FEM) was chosen. This method is the most acceptable for solving our problem, since the temperature measurement in the contact area of the wheel and the shoe is very complicated and might have significant instrumental errors.

Experimental studies are necessary, but for their optimal planning, one needs to start with FEM modeling. The FEM modeling will make it possible to determine the temperature distribution in the contact area, to place the sensors at the correct points and to avoid errors. For this reason, our research presented in this article will focus on the FEM model.

The closest study was conducted for the Serbian Railways [17]. In the present work, a FEM model of the wheel and shoes is constructed. The aim of the study was to determine the level of high temperatures leading to the destruction of the wheel. In the study, we obtained temperatures in the friction zone with various pressures of \( B_{st} = 20379 \text{N} \) and \( B_{st} = 37162 \text{N} \) for different braking velocities of 20, 40, 60 km / h. The coefficient of friction in this case was \( \mu = 0.115 \). However, in the above studies, we considered braking with new brake shoes, the geometry and mass of which did not change.

To study the change in contact temperature during wear of the brake shoes, we modeled three types of shoe geometry shown in Figure 3.

![Figure 3. Models of brake shoes in three states: new brake shoe a, worn brake shoe with uniform wear b, worn brake shoe with wedge-shaped wear c.](image)

The modeled brake shoes had the following parameters. The first shoe (Figure 3 a) has the geometry of a new shoe dimensioned 80 mm wide, 85 mm thick and 380 mm high. The second shoe has the geometry of a uniformly worn shoe, 16 mm thick with the same dimensions of height and width.
(Figure 3 b). The third shoe has a shoe geometry with uneven wear. The thickness is variable from the thin end of 25 mm from the thin end of 13 mm with the same dimensions of height and width (Figure 3 c).

Given the range of the coefficient of friction for different velocities and pressing, we take its average calculated value \( \varphi = 0.2 \). In the process of frictional interaction of the shoe with the wheel, heat is released in proportion to the mechanical power generated by the braking force. Its power can be found by the formula for kinematic systems:

\[
P = \frac{M \cdot n}{9.55 \varphi}
\]

where \( M \) is the braking torque, kN\( \cdot \)m; \( n \) is the number of the wheel turns, rpm

As can be seen from this dependence, at high wheel velocities the power that needs to be dissipated is higher than at low velocities. Respectively, in the zone of contact of the wheel with the shoe, the temperature becomes higher, and therefore the braking efficiency decreases due to a decrease in the coefficient of friction. Therefore, when the train set slows down, the dissipated thermal power decreases, and the temperature of the shoes in the immediate contact zone decreases. Therefore, the action of the friction force of adhesion becomes more effective.

Braking torque \( M \) is the product of braking force \( B_{st} \) by wheel radius \( r_w = 0.475 \) m:

\[
M = B_{st} \cdot r_w
\]

For each geometry of the shoes, three calculations were performed for different pressure on the shoe: \( T_1 = 17.15 \) kN, \( T_2 = 19.62 \) kN, \( T_3 = 39.2 \) kN. The indicated pressures corresponded to the braking forces \( B_{st1} = 3.43 \) kN, \( B_{st2} = 3.924 \) kN, \( B_{st2} = 7.84 \) kN. Braking was considered for a velocity of 60 km/h.

The density of the heat flux applied to the shoe is determined as the ratio of the heat power (expression (11)) allocated in the friction region to the doubled value of the area of interaction between the wheel and the shoe, since the heat flux is divided between the wheel and the shoe.

\[
q = \frac{p}{2S_s}
\]

where \( S_s \) is the area of the shoe surface, \( S_s = 0.0305 \) m\(^2\).

The shoe surface area \( S_s \) was determined by building a model of the shoe with its actual dimensions using the Autodesk Inventor software environment.

The computational model considers only the shoe, since the wheel temperature is not of interest at this stage. The finite element model of an unworn brake shoe is shown in Figure 4. A grid of Hex-type elements was used in this model; the solution was produced in the MSC Patran / Sinda environment.

![Figure 4. The Finite Elementary Model of an unworn brake shoe.](image)

Material properties are adopted according to the reference data for wheel steel and cast iron. The calculation was carried out in an unstable setting.

3. Results
As a result of the calculation, it turned out that the geometric dimensions of the shoes significantly influence the steady-state temperature in the zone of the wheel-rail contact. It also turned out that the heating rate of a worn shoe is several times greater than that of a new one. In this case, there is a significant temperature difference between the friction surface and the back surface of the shoe. To graphically visualize the thermal field inside the shoe, Figure 5 shows longitudinal sections of thermal models.

**Figure 5.** Distribution of temperature fields in the longitudinal section of shoes without wear a, with uniform wear b, and with wedge-shaped wear c.

As can be seen from Figure 5, the maximum temperature of the shoes occurs on the outermost edges of the shoes for all cases of wear. An unworn shoe has a large mass, which allows one to distribute more heat in a cold volume, providing heat removal from the friction surface, which reduces the maximum temperature in the "shoe-wheel" system. Shoes with high wear and low mass with the same value of the heat flux heat up much more. This can be seen in the image of thermal fields (figure 5 a, 5 b). For a comparative analysis, the average temperatures in the "shoe-wheel" contact zone are summarized in table 1.

| Pressing, kN | Temperature, °C |
|-------------|----------------|
|             | New | Worn out | Wedge-shaped wear |
| 17.185      | 155 | 158       | 167               |
| 24.55       | 192 | 231       | 243               |
| 39.28       | 243 | 683       | 730               |

The obtained temperature values are approximately consistent with the results obtained in [17]. The difference is due to the different geometric parameters of the brake shoes, the difference in pressing force and the different friction coefficients.

Also, the heating process slows down in the new shoe due to the large volume of material. Worn shoes, due to their low mass, heat up much more intensively than new ones. At the same time, shoes with wedge-shaped wear gain a temperature at the thin end higher than shoes with uniform wear. To comparatively analyze the heating rate of shoes, depending on the degree of wear and pressure, the results of the FEM calculation are presented in the form of graphs in Figure 6.
The graph of temperature changes with the force of shoe pressure on the wheel $T_1 = 17.15$ kN, which corresponds to pressing service braking, is shown in Figure 6 a. This braking can be used to control the velocity of the train. In this case, a small heat flux is released in the friction region. The temperature for all wear conditions of the shoes reaches approximately the same level, but its rise time differs by more than 2.5 times. Figure 6 b shows the result of the calculation of the braking process for pressing force $T_2 = 19.62$ kN. Such pressing force is characteristic of service braking, resulting in a slow stop of the train. In this case, the heat flux increases. The worn-out shape of the shoe causes low heat transfer, which does not cope with the incoming heat flow. As a result of this, the temperature starts rising. With this braking, the steady-state temperatures begin to differ more significantly (about 20%). Figure 6 c shows a graph for calculating the braking temperature with pressing $T_3 = 39.2$ kN. This pressing is applied during full service or emergency braking. Such braking causes the maximum heat flux, which leads to a sharp difference in the steady-state temperature in the friction zone of the shoe and the wheel. The difference in the steady-state temperature between the shoes with different degrees of wear reaches 2.5-3 times. The heating time up to a steady state also varies by 2.5 times.

4. Discussion
Based on the obtained simulation results from the three points of force obtained for the shoe pressure on the wheel, we can construct generalized graphs of the change in the steady-state maximum value for shoes with varying degrees of wear (Figure 7).
The graphs show a significant increase of the difference between the steady-state temperatures in the "shoe-wheel" friction zone for blocks with various degrees of wear.

5. Conclusions
In the final part of the article, we can draw a number of conclusions:

- the degree of wear affects the heat capacity and heat transfer of the brake shoes;
- the temperature of the brake shoes varies significantly with the degree of wear;
- a change in temperature leads to a change in the coefficient of friction;
- trains with the same parameters and braking conditions having brake shoes with different degrees of wear can stop after completely different distance;
- due to varying degrees of wear the difference in temperature of the brake shoes becomes more pronounced during braking with the high pressing of the brake shoes.
  - worn brake shoes heat up 2.5 times faster than new ones;
  - for short braking, up to 150 seconds, the low heat capacity of worn brake shoes leads to an increase in temperature dispersion.

The conclusions made can be useful for specialists who determine the modes of brake control for trains, and designers of brake systems for railway cars and locomotives.

The results can be used to simulate the movement of the train and to clarify the equation of the train movement, making it possible to determine the velocity of movement and coordinates of the train at any given time.

The prospects of this research are the development and creation of an experimental unit that will clarify the dynamics of the change in the friction factor in the braking process and clarify the dependence of brake force on the wear of brake shoes and other braking parameters.

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