Application of Modern Software Complexes in the Analysis of the Parameters of Friction Pairs of the Spring-Friction Apparatus of a Freight Car for their Efficiency

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Abstract: This study presents comprehensive analytical and numerical modeling of the force interactions in friction pairs, specifically between the “clamping beam – friction wedge” and “friction wedge – friction rod,” using the freight car model 18-9597 as a representative case. By integrating both analytical derivations and finite element simulations, the research identifies rational design parameters that ensure stable, efficient, and long-term performance of the spring-friction damping unit in freight car bogies. The study not only enhances the understanding of contact mechanics and frictional behavior under dynamic loading but also provides practical insights into the geometric optimization of friction elements. The derived formulas for coupling reactions as functions of friction pair geometry offer a valuable tool for engineers in designing more reliable systems. The findings are applicable to both the modernization of existing railway rolling stock and the design of new-generation freight wagons, potentially contributing to improved safety, durability, and maintenance efficiency across the rail industry.

1 INTRODUCTION

Analysis of the operation of friction wedges in the truck of the freight car model 18-9597. The main task of friction wedges is to transform the oscillatory movements occurring in the vertical and longitudinal directions into a friction force that dissipates the energy of these vibrations. At the same time, the geometry of the mating parts and the pre-compression force of the springs are of great importance, ensuring constant contact of the wedge with the friction plate and the support beam.

In freight wagons, to dampen vibrations from track irregularities and to transfer vertical and longitudinal loads. The friction wedges, interacting with the friction bar, dampen vibrations from the unevenness of the path. The study of the interaction between the “superstructure beam and the friction wedge” and between the “friction wedge and the friction bar” is necessary to increase the reliability and efficiency of the construction of wagons. Today, there are many ways of engineering analysis. For analysis, we will use the analytical method and numerical modeling to determine the reaction of bonds depending on the geometric pairs of friction, to find rational values. The analytical approach is based on classical equations of statics, dynamics, and friction theory.

To study the mechanics of interaction, an analytical method based on solving a system of equilibrium equations taking into account friction forces was used. Both normal and tangential forces acting on the wedge from both sides are taken into account. Parameters have been introduced: the angle of inclination of the working surface of the wedge, friction coefficients, spring stiffness, weight of the support beam and external dynamic loads. In parallel, numerical modeling using the finite element method (FEM) was carried out, in which a three-dimensional model of the node was constructed. Real contact conditions are applied, taking into account dry friction and nonlinear deformation of the elements. This made it possible to track the distribution of loads in various operating modes: starting, steady movement, braking, and moving over uneven tracks.

The simulation results confirmed that the vibration damping efficiency significantly depends on:

- the angle of inclination of the working surface of the wedge (optimally - 18-22°);
- uniformity of force from springs;
- conditions of rubbing surfaces (wear, contamination);
- values of the coefficient of friction (in the range 0.25–0.35 is preferable).

Thus, the combination of analytical and numerical approaches makes it possible to determine rational design parameters that ensure reliable and long-term operation of the spring-friction unit. The data obtained can be used in the modernization of existing trolleys and the development of new models of freight wagons.

Structurally, the spring-friction set of the truck of the freight car model 18-9597 is designed so that its friction wedges 2 and 3 are in contact with only three solid elements – a spring carrier 1, a frictiobar 4 (or 5) and double pins 7 (or 8) (Fig. 1).

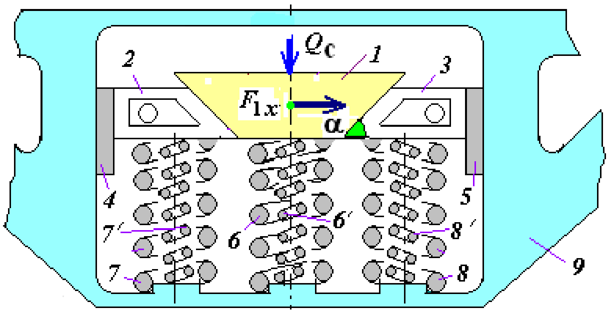


Figure 1: Spring-friction set of a bogie loaded with the pressure force of the car body with cargo.

In Figure 1 the following are designated: 1 – tail section of the bolster; 2 and 3 – friction wedges; 4 and 5 – friction bar; 6 – sets of springs under the bolster; 7 and 8 – sets of sub-wedge springs; 9 – side frame.

In [1], the spring-friction complex of a truck model 18-9597 of a freight wagon was numerically modeled in the classical formulation of the problem. A final numerical formula has also been obtained for determining the reaction of the bonds of friction wedges to a spring beam, taking into account the design features of the wedges and manufacturing errors on inclined surfaces.

The proposed model was further investigated, and a method was developed for determining the coupling reactions of the spring-friction complex of the trolley model 18-9597, which takes into account the possible angular contact of the rubbing surfaces by individual faces [2].

When creating a mathematical model of a friction wedge 2 (or 3) as an object of analytical research, according to the principle of bond release [3], the contact surfaces from the side of the pressure beam 1 and the friction bar 4 (or 5), considered as external bonds, are replaced by two bond reactions represented as. , an alternative support beam, and , an alternative friction bar.

The formula for the actual reaction on an equivalent inclined surface is calculated when the friction wedge moves vertically downwards, taking into account that the friction wedge is overloaded due to the longitudinal force applied to the support beam, taking into account only the reaction of the spring springs. The defined formula does not contain the mass of the container of the wagon with the load and the reaction of the spring springs. The study [4] did not specify which of the friction wedges is overloaded and which is unloaded.

In previous studies, the results of analytical and numerical modeling of the force action of friction pairs "pressure beam - friction wedge" and "friction wedge – friction bar" of the spring-friction trolley set model 18-9597 freight wagon were obtained. The equations of equilibrium of the superstructure beam as a physical body were calculated using the analytical method.

Analytical formulas have been developed for determining bond reactions depending on geometric parameters, in particular, the angles of inclination of the contacting surfaces of friction pairs. However, the studies [5], [6] did not take into account the factor of the longitudinal force acting from the front carriage and applied to the suspension beam of the bogie model 18-9597.

Failure to account for this longitudinal force leads to an underestimation of the loads acting on the friction pairs, which, in turn, reduces the accuracy of forecasts of wear and durability of suspension units. This can lead to premature component failure and increased operating costs.

To increase the accuracy of the model, it is necessary to include the influence of the longitudinal force in the calculations, which will allow for more adequate estimates of bond reactions and provide realistic modeling of the dynamics of force interaction in the spring-friction complex. In particular, it is proposed to supplement the mathematical model taking into account the applied longitudinal force, which will create conditions for a more accurate wear analysis and the development of recommendations for design optimization.

2 MODEL DESCRIPTION AND ASSUMPTIONS

When modeling the asymmetric placement of a solid–state load relative to the axis of symmetry of the carriage, as shown in [5, 6], it should be noted the importance of analytical determination of reactions in friction pairs: "pressure beam - friction wedge" and "friction wedge – friction bar". These reactions depend on the longitudinal force exerted by the lead car on the support beam, as well as on the geometric parameters of the structural elements.

Let's assume that through the heel of the wagon frame, pressure forces from the wagon frame with a load are transmitted to the spring support of the suspension beam [5], [6]. The corresponding pressure forces (let's denote them as QC or QD) are transmitted to the sets of bogie springs of the freight car. In response, the elastic forces of these sets of springs exert pressure on the support beam and friction wedges, which, in turn, transfer the load to the side frames of the trolleys.

In this case, only one part of the longitudinal force F_x, designated as F_{1x}, is taken into account, which is applied to the suspension beam from the side of the coupling devices of the lead car. The remaining parts of the longitudinal force: F_{2x} – applied to the suspension beam of the rear bogie [7], and the fraction transmitted to the coupling devices of the driven wagons [3], [7] are not considered in this task. Let's model the design scheme of the mechanical system "pressure beam – friction wedges – friction bars" in accordance with the image on Figure 2. In this model:

F_6 – These are the reactions (elastic forces) of sets of spring springs, which create resistance to vertical movement of the support beam 1 downwards. The calculations take into account five such sets (see Figure 2a);

F_7 и $F8$ – These are the reactions (elastic forces) of sets of wedge springs that resist the lowering of friction wedges 2 and 3, respectively (see Figures 2b and 2d).

Thus, this scheme allows us to take into account the operation of the system elements when transferring vertical loads through friction elements to freight carts.

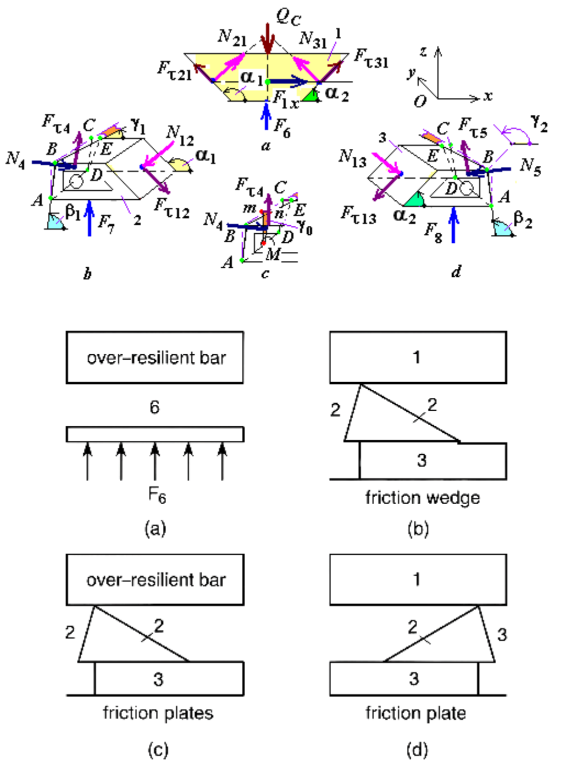


Figure 2: Calculated models of the pressure beam 1 and friction wedges 2 and 3.

In Figure 2 the following designations are indicated: N_{21}, N_{31} и $F_{\tau 21}$, $F_{\tau 31}$ – normal and tangential components of bond reactions (friction wedges) *2* и *3*; α_1 и α_2 – angles of inclination of the surfaces of the pressure beam to the horizon, rad. ($\alpha = 45^{\circ} + 1^{\circ}$); in Figure 2b, 2d: N_{12}, N_{13} и $F_{\tau 12}$, $F_{\tau 13}$ – the normal and tangential components of the coupling reaction 1 (tail section of the suspension beam) to wedges 2 and 3, N_4, N_5 и $F_{\tau 4}$, $F_{\tau 5}$ – the normal and tangential component of the bond reaction (friction strips), $\gamma_1 \leq \frac{\pi}{2}$ и $\gamma_2 \geq \frac{\pi}{2}$ – the angle of inclination of the back surface *ABED* friction wedges 2 and 3, in contact with the friction plates 4 and 5, rad ($\gamma_1 \approx 88^{\circ} - 1^{\circ}$, $\gamma_2 \approx 92^{\circ} + 1'$).

Let's analyze the equilibrium of the superstructure beam 1 (see Fig. 2a). The pressure beam 1 is affected by the reactions R_{21} and R_{31} of the friction wedges 2 and 3, which are decomposed into normal and tangential components N_{21}, N_{31} and F_{t21}, F_{t31}; the active force QC, equal to the reaction force occurring on the pressure beam 1, and the reactive force in the form of an equivalent reaction of sets There are 6 F_6 springs. At the same time, we assume that the inclined surfaces of the superstructure beam are made with disadvantages, i.e. $\alpha_1 \neq \alpha_2$, where α_1 and α_2 are the angles of inclination of the surfaces of the superstructure beam 1 to the horizon, rad. ($\alpha_1 \approx 134^\circ 30' + 1^\circ$, $\alpha_2 \approx 45^\circ 30' + 1^\circ$).

In the analytical study, as in [5], [6], we assume that the angles of inclination of the surfaces (a1 and a2) of the pressure beam 1, the friction wedges 2, 3 and the friction bars (β_1, β_2 and γ_1, γ_2) have different values ($\alpha_1 \neq \alpha_2$, $\beta_1 \neq \beta_2$, and $\gamma_1 \neq \gamma_2$), which correspond either to their manufacture with errors, or take into account the uneven wear of their surfaces.We also assume that the coefficients of sliding friction f between the contacting surfaces of the support beam (f_1 and f_2), friction wedges and slats (f_3 and f_4) have different values.

3 METHODS

To solve this applied problem, we use the kinetostatics method from the course of theoretical mechanics [3]. It is based on an engineering calculation method based on the fact that inertia forces are reduced to an equivalent system of forces, while taking into account external loads in equilibrium conditions.

4 RESEARCH RESULTS

Let's calculate two equations of equilibrium of a plane system of forces, equating the sum of the projections of all forces on the x and z axes to zero:

$$N_{21}\cos(\alpha_1 - \frac{\pi}{2}) + F_{\tau 21}\cos(\alpha_1) + N_{31}\cos(\alpha_2 + \frac{\pi}{2}) + F_{\tau 31}\cos(\alpha_2) + F_{1x} = 0 \quad (1)$$

$$N_{21}\sin(\alpha_1 - \frac{\pi}{2}) + F_{\tau 21}\sin(\alpha_1) + N_{31}\sin(\alpha_2 + \frac{\pi}{2}) + F_{\tau 31}\sin(\alpha_2) - Q_C + F_6 = 0 \quad (2)$$

Here, as you can see, there are two independent equilibrium equations, and four unknown ones: N_{21}, N_{31} and F_{t21}, F_{t31}.To solve the problem, it is sufficient to add (1) and (2) the equation following from Coulomb's law

$$F_\tau \leq fN, \quad (3)$$

where f – is the coefficient of sliding friction between the contacting surfaces of the bolster 1 and the friction wedges 2, 3, as well as between the friction wedges 2, 3 and the friction bars.

Substituting equalities (3) into (1) and (2), after transformations we have

$$aN_{21} + bN_{31} = -F_{1x}, \quad cN_{21} + dN_{31} = Q_C - F_6, \quad (4)$$

where a, b, c and d – are constant coefficients:

$$a = cos(\alpha_1 - \frac{\pi}{2}) + f_1 cos(\alpha_1);$$
$$b = cos(\alpha_2 + \frac{\pi}{2}) + f_2 cos(\alpha_2);$$
$$c = sin(\alpha_1 - \frac{\pi}{2}) + f_1 sin(\alpha_1);$$
$$d = \sin(\alpha_2 + \frac{\pi}{2}) + f_2 \sin(\alpha_2) \quad (5)$$

According to Kramer's rule [8], from system (4) we find the normal components of the reaction of the connections (friction wedges 2 and 3) in the execution of inclined surfaces of the support beam 1 with errors:

$$N_{21} = -\frac{1}{ad - bc}\left[F_{1x}\left(\sin(\alpha_2 + \frac{\pi}{2}) + f_2\sin(\alpha_2)\right) + (Q_C - F_6)\left(\cos(\alpha_2 + \frac{\pi}{2}) + f_2\cos(\alpha_2)\right)\right] \quad (6)$$

$$N_{31} = \frac{1}{ad - bc}\left[(Q_C - F_6)\left(\cos(\alpha_1 - \frac{\pi}{2}) + f_1\cos(\alpha_1)\right) + F_{1x}\left(\sin(\alpha_1 - \frac{\pi}{2}) + f_1\sin(\alpha_1)\right)\right] \quad (7)$$

The matrix, composed of the coefficients of the unknowns of system (4), and calculated symbolically [9], [10] is equal to:

$$\begin{aligned} ad - bc &= \sin(\alpha_1)\cos(\alpha_2) + \sin(\alpha_2) f_2 \sin(\alpha_1) + \\ &+ f_1\cos(\alpha_1)\cos(\alpha_2) + f_1\cos(\alpha_1) f_2\sin(\alpha_2) + \\ &+ \sin(\alpha_2)\cos(\alpha_1)(-1) + \sin(\alpha_2) f_1 \sin(\alpha_1) + \\ &+ f_2\cos(\alpha_1)\cos(\alpha_2) - f_2\cos(\alpha_2) f_1\sin(\alpha_1). \end{aligned} \quad (8)$$

Assume that if the angle is negative, then the function is reduced to the function of the positive angle by the formulas $\sin(\alpha - \frac{\pi}{2}) = -\sin(\frac{\pi}{2} - \alpha) = -\cos(\alpha)$ and $\cos(\alpha - \frac{\pi}{2}) = \cos(-(\frac{\pi}{2} - \alpha) = \sin(\alpha)$ [8], relations (6), taking into account expression (8), are written as

$$N_{21} = -\frac{1}{\begin{aligned}&\sin(\alpha_1)\cos(\alpha_2) + \sin(\alpha_2) f_2 \sin(\alpha_1) + \\ &+ f_1 \cos(\alpha_1)\cos(\alpha_2) + f_1 \cos(\alpha_1) f_2 \sin(\alpha_2) + \\ &+ \sin(\alpha_2)\cos(\alpha_1)(-1) + \sin(\alpha_2) f_1 \sin(\alpha_1) + \\ &+ f_2 \cos(\alpha_1)\cos(\alpha_2) - f_2 \cos(\alpha_2) f_1 \sin(\alpha_1)\end{aligned}} \times \begin{bmatrix} F_{1x}\left(\cos(\alpha_2) + f_2 \sin(\alpha_2)\right) - \\ -(Q_C - F_6)\left(\sin(\alpha_2) - f_2 \cos(\alpha_2)\right) \end{bmatrix} \quad (9)$$

$$N_{31} = \frac{1}{\begin{aligned}&\sin(\alpha_1)\cos(\alpha_2) + \sin(\alpha_2) f_2 \sin(\alpha_1) + \\ &+ f_1 \cos(\alpha_1)\cos(\alpha_2) + f_1 \cos(\alpha_1) f_2 \sin(\alpha_2) + \\ &+ \sin(\alpha_2)\cos(\alpha_1)(-1) + \sin(\alpha_2) f_1 \sin(\alpha_1) + \\ &+ f_2 \cos(\alpha_1)\cos(\alpha_2) - f_2 \cos(\alpha_2) f_1 \sin(\alpha_1)\end{aligned}} \times \begin{bmatrix} (Q_C - F_6)\left(\sin(\alpha_1) + f_1 \cos(\alpha_1)\right) - \\ - F_{1x}\left(\cos(\alpha_1) - f_1 \sin(\alpha_1)\right) \end{bmatrix}. \quad (10)$$

Sometimes in the case when $f_1 = f_2$ and $\alpha_1 = \alpha_2 + \frac{\pi}{2}$, which correspond to the design of the inclined surfaces of the support beam 1 without errors, expressions (9) and (10) will take the form:

$$N_{21} = -\frac{1}{1 - f_1^2} \begin{bmatrix} F_{1x}\left(\cos(\alpha_2) + f_1 \sin(\alpha_2)\right) - \\ -(Q_C - F_6)\left(\sin(\alpha_2) - f_1 \cos(\alpha_2)\right) \end{bmatrix} \quad (11)$$

$$N_{31} = \frac{1}{1 - f_1^2} \begin{bmatrix} (Q_C - F_6)\left(\cos(\alpha_2) - f_1 \sin(\alpha_2)\right) + \\ + F_{1x}\left(\sin(\alpha_2) + f_1 \cos(\alpha_2)\right) \end{bmatrix}. \quad (12)$$

that is, $N_{31} = N_{21}$ at $F_{1x} = 0$, which is consistent with the results of [1], confirming the correctness of the obtained analytical expressions.

The nature of the change in the reaction of the connections (friction wedges 2 and 3) depending on the variation of the angles of inclination (α_1 and α_2) of the inclined surfaces of the friction pairs "pressure beam - friction bars" without taking into account and taking into account the errors in their manufacture in accordance with formulas (9) and (10) are shown in Figure. 3 a – Figure 3d.

The analysis of the $N_{21}(\alpha_1)$ shows that with an increase in the angle α_1 (which is equivalent to a decrease in the angle α_2), the magnitude of the bond reaction increases, whereas with a decrease in α_1, on the contrary, it decreases. At the same time, taking into account the manufacturing errors of inclined surfaces in friction pairs leads to a decrease in the bond reaction compared with calculations without taking them into account.

In turn, the analysis of the dependence of N_{31} (α_2) indicates that with an increase in the angle α2 (which corresponds to a decrease in α_1), the bond reaction decreases, and with a decrease in α_2, it increases. When taking into account the manufacturing errors of inclined surfaces in friction pairs, an increase in the bond reaction is observed compared to calculations without taking them into account.

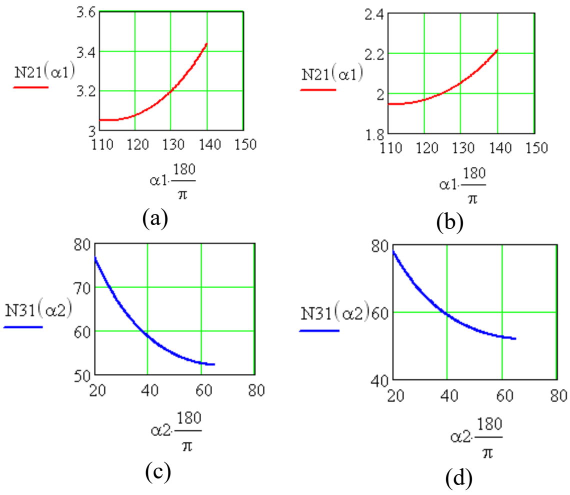


Figure 3: The nature of the change in the reaction of the $N_{21}(\alpha_1)$ and $N_{31}(\alpha_2)$ bonds: a), c) and b), d) – without taking into account and taking into account the manufacturing errors of the inclined surfaces of friction pairs.

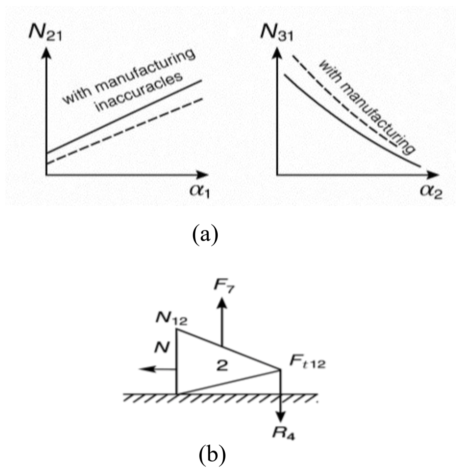


Figure 4: Friction wedge equilibrium.

Comparing the values of $N_{21}(\alpha_1)$ and $N_{31}(\alpha_2)$, it should be noted that the value of $N_{21}(\alpha_1)$ is always less than $N_{31}(\alpha_2)$.This is due to the effect of the longitudinal force of the adjacent F_{1x} wagon on the suspension beam. As a result, there is a more complete contact of the friction pair "friction wedge 3 - friction bar 5" compared to the pair "friction wedge 2 – friction bar 4".

When considering the equilibrium conditions of the friction wedge 2 (see Fig. 4b) it is necessary to take into account the following forces acting on it according to the axiom of equality of action and reaction:

$N_{12} = -N_{21}$ – the normal component of the reaction from the side of the pressure beam;

$F\tau_{12} = -F\tau_{21}$ – the tangential component of the same reaction.

The reaction from the side of the friction bar 4, represented by the normal component N_4 and the tangential component $F_{\tau 4}$ (reaction vector R_4);

The resultant of the reactions of the sets of spring springs 7 is the upward force F_7.

These forces ensure the equilibrium of the wedge 2 in the process of interaction with the support beam and the friction bar. Let's make up the force equilibrium equations for the friction wedge 2:

$$N_{12}\cos(\alpha_1+\frac{\pi}{2})+F_{\tau 12}\cos(\alpha_1+\pi)+N_4\cos(\gamma_0)\cos(\beta_1+\frac{3}{2}\pi)+ \\ +F_{\tau 4}\cos(\gamma_0)\cos(\beta_1)=0, \quad (13)$$

where γ_0 – is the angle accompanying the calculation, as in [5], [6].

Taking into account the relation (3), we rewrite the last expressions

$$N_{12}\left(\cos(\alpha_1+\frac{\pi}{2})+f_1\cos(\alpha_1+\pi)\right)+ \\ N_4\cos(\gamma_0)\left(\cos(\beta_1+\frac{3}{2}\pi)+f_3\cos(\beta_1)\right)=0; \quad (14)$$

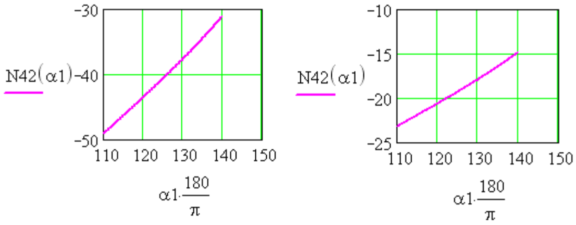
From relation (14), taking into account (9), after intermediate transformations, we determine the normal component of the reaction of the friction wedge 4.

$$N_4=\frac{1}{\begin{array}{l}\sin(\alpha_1)\cos(\alpha_2)+\sin(\alpha_2)f_2\sin(\alpha_1)+ \\ +f_1\cos(\alpha_1)\cos(\alpha_2)+f_1\cos(\alpha_1)f_2\sin(\alpha_2)+ \\ +\sin(\alpha_2)\cos(\alpha_1)(-1)+\sin(\alpha_2)f_1\sin(\alpha_1)+ \\ +f_2\cos(\alpha_1)\cos(\alpha_2)-f_2\cos(\alpha_2)f_1\sin(\alpha_1)\end{array}}\times \\ \times\left[\begin{array}{l}F_{1x}(\cos(\alpha_2)+f_2\sin(\alpha_2))- \\ -(Q_C-F_6)(\sin(\alpha_2)-f_2\cos(\alpha_2))\end{array}\right]\times\frac{(\sin(\alpha_1))+f_1\cos(\alpha_1))}{\cos(\gamma_0)(\sin(\beta_1)+f_3\cos(\beta_1))}. \quad (15)$$

The nature of the change in the reaction of the bonds (friction strip 4) depending on the variation in the angle of inclination (α_1) of the inclined surfaces of the friction pairs "pressure beam - friction strips" at α_2 = const, without taking into account and taking into account the errors in their manufacture in accordance with (15) are shown in in Figure 5a and 5b.

The analysis of the dependences of $N_{42}(\alpha_1)$ shows that with an increase in the angle α_1, the reaction of bonds along the module decreases, and with a decrease, vice versa. When taking into account the manufacturing errors of the inclined surfaces of the friction pairs, the bond reaction is smaller in magnitude than without taking them into account.

Regarding the study, now let's consider the equilibrium condition of the friction wedge 3 (see Fig. 5).



(a) (b)

Figure 5: The nature of the change in the reaction of the $N_{42}(\alpha_1)$: connections: a) and b) – without taking into account and taking into account the manufacturing errors of the inclined surfaces of the friction pairs.

According to the axiom of equality of action and counteraction, the friction wedge 3 is affected by: normal N_{13} = - N_3 and tangent $F_{\tau 13} = - F_{\tau 3}$ components of the reaction of the pressure bar $R_{13} = - R_3$; normal N_5 and tangent $F_{\tau 5}$ components of the reaction of the friction bar 5 R5; as well as the resultant reactions of sets of springs 8 - F_8. Take into account, 2 d : $\gamma_2 \geq \frac{\pi}{2}$ – the angle of inclination of the rear surface *ABED* of the friction wedge 3 in contact with the friction plate 5, rad. ($\gamma_2 = 92^o + 1^o$).

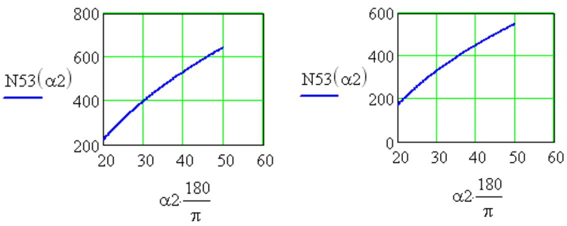
Let us develop the force equilibrium equations for the friction wedge 3 in the same way as for the friction wedge 2:

$$N_{13}\cos(\alpha_2+\frac{3}{2}\pi)+F_{\tau 13}\cos(\alpha_2+\pi)+ \\ +N_5\cos(\gamma_0)\cos(\beta_2+\frac{\pi}{2})+F_{\tau 5}\cos(\gamma_0)\cos(\beta_2)=0. \quad (16)$$

Taking into account relations (3) from expression (16), taking into account (10) after transformations, we determine the normal component of the reaction of the friction wedge 4.

$$N_5 = \frac{1}{\begin{array}{l}\sin(\alpha_1)\cos(\alpha_2) + \sin(\alpha_2) f_2 \sin(\alpha_1) + \\ + f_1 \cos(\alpha_1)\cos(\alpha_2) + f_1 \cos(\alpha_1) f_2 \sin(\alpha_2) + \\ + \sin(\alpha_2)\cos(\alpha_1)(-1) + \sin(\alpha_2) f_1 \sin(\alpha_1) + \\ + f_2 \cos(\alpha_1)\cos(\alpha_2) - f_2 \cos(\alpha_2) f_1 \sin(\alpha_1)\end{array}} \times \left[\begin{array}{l}(Q_C - F_6)\left(\sin(\alpha_1) + f_1\cos(\alpha_1)\right) - \\ - F_{1x}\left(\cos(\alpha_1) - f_1 \sin(\alpha_1)\right)\end{array}\right] \times \frac{\left(\sin(\alpha_2) - f_2\cos(\alpha_2)\right)}{\cos(\gamma_0)\left(\sin(\beta_2) - f_4\cos(\beta_2)\right)}. \quad (17)$$

The nature of the change in the reaction of the bonds (friction strip 5) depending on the variation in the angle of inclination (α_2) of the inclined surfaces of the friction pairs "pressure beam - friction strips" at α_1= const without taking into account and taking into account the errors in their manufacture in accordance with formula (17) are shown in Figure 6a and Figure 6b.



(a) (b)

Figure 6: The nature of the change in the reaction of the $N_{53}(\alpha_2)$ bonds: a) and b) – without taking into account and taking into account the manufacturing errors of the inclined surfaces of friction pairs.

The analysis of the dependences of $N_{53}(\alpha 2)$ shows that with an increase in the angle α2, the bond reaction modulus increases, and with a decrease, it decreases accordingly. When taking into account the manufacturing errors of the inclined surfaces of the friction pairs, the bond reaction value is lower compared to the calculation without taking them into account.

An analytical study of the values of $N_{42}(\alpha 1)$ and $N_{53}(\alpha 2)$ allows us to note that $N_{42}(\alpha 1)$ is significantly less than $N_{53}(\alpha 1)$ due to the effect of the longitudinal force of the mixed wagon F_{1x} on the superstructure beam. For this reason, premature wear of the friction pair "friction wedge 3 – friction bar 5" in comparison with the friction pair "friction wedge 2 – friction bar 4" is natural.

5 CONCLUSIONS

This paper has demonstrated the effective application of modern software systems; such as Mathcad, COMSOL, and Universal Mechanism in studying and modeling the frictional behavior of spring friction assemblies in freight car bogies, with a focus on the model 18-9597. Through a combination of analytical and numerical methods, the study derived precise mathematical relationships that define the reaction forces in friction pairs, accounting for geometric parameters, manufacturing tolerances, and the influence of longitudinal forces from adjacent cars, the investigation revealed significant asymmetries in force distribution due to the uneven inclination angles and friction coefficients, which in turn cause unequal wear between friction pairs specifically, greater wear in the friction wedge 3 friction bar 5 pair compared to friction wedge 2 friction bar 4. This observation is critical for improving maintenance strategies and optimizing component design.

By integrating the principles of kinetostatics and Coulomb friction into symbolic calculations and validating the model through simulation, the study provides a robust framework for predicting performance degradation and enhancing the operational reliability of rail suspension systems. These results pay to the strategic implementation of digital diagnostic tools in railway engineering, aimed at extending service life and ensuring the safety and efficiency of freight transportation.

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