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Simulation of spring-friction set of freight car truck, taking into account track profile

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Abstract

The purpose of this article is the mathematical modeling of a spring-friction set of a freight car trolley depending on the profile of the railway track. The article uses the methodology of the results of the generalized model of power interaction of friction pairs "pressure beam-friction wedge" and "friction wedge-friction bar" of the spring-friction set of the cart 18-578 in the mechanical system "track-car-load," in a particular case cart from which such models of the e of the freight car 18-100 can be obtained. The manuscript contains a software study using a computer simulation of a freight wagon trolley, which complements the experimental studies. The mathematical (symbolic) method in the computer algebra system MathCAD (a type of computer-aided design system) is used to solve the equations of equilibrium of the supercharger beam as a real object. In the course of the experimental study, a computational mathematical model of the force action of friction pairs of spring-absorbing trolley apparatuses was developed, and the results of calculations of the reaction of the friction bar to the friction wedge were mathematically obtained. The optimal parameters were obtained analytically, and the indicators of the angles of inclination of the contact surfaces of the trolley of the freight car were justified.

Keywords: Cart of the freight car, Dynamic forces, Force distribution, Rolling stock, Spring friction kit, Track profile, Truck.

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1. Introduction

The JSC "National Company" Kazakhstan Temir Zholy" is committed to developing new strategies for transportation planning, organization, and implementation until 2020, aiming to improve the efficiency of rolling stock and capacity of network sections.

The Company's strategy for this project involves aiming to achieve a 30% increase in the train's average weight. At the moment, in Kazakhstan, a freight train has a unified weight limit of 6000 tons, which is above the unified weight limit for freight trains in the Republic of Kazakhstan.

The introduction of heavy trains involves a complex task linked to the use of more potent locomotives, the increase of axial loads, the reconstruction of track infrastructure and electricity supply, and the improvement of transport process technology.

The routes leading from Ekibastuz to Golden Sopka (Russian Federation) and beyond are most likely to facilitate the implementation of coal delivery in Kazakhstan for heavy traffic.

The mass of freight trains is constrained by the length of the train on the designated section, which is determined by the condition of free installation on station tracks (the maximum number of cars in the composition is 71 conditional cars).

The aforementioned freight train weights will reach 7,100 tons without the need for infrastructure reconstruction, as the freight trains are operated on an axial load of 245 kN.

It is known that the structure of nonlinear inertia forces is very complex. If Eulerian angles are used to describe them, this leads to very cumbersome trigonometric relations between all six coordinates and their derivatives. Studies show that for a rail carriage, the assumption that the roll and pitch angles are small and the yaw angle has a finite value does not introduce a significant error.

When using computer mathematical modeling, each unit of rolling stock, each of its elements and the railway track must be represented as a complex dynamic system consisting of a large number of mechanical elements with different types of connections and constraints between them. That is, the mathematical model should be adequate for the full-scale physical processes occurring in experiments with real track and rolling stock structures [1, 2].

The existing scale of limit values for dynamic indicators and stability from the derailment of freight cars is not sufficient to assess the quality and safety of empty cars. To eliminate this disadvantage, along with traditional dynamic indicators such as the interaction of the chassis of the trolley, the stability reserve of the wheelset from derailment under the condition of rolling the wheel crest onto the rail head, acceleration of the crew elements, and traffic safety, additional criteria are assessed.

In this regard, the main purpose of this work is to assess the safety of the movement of a freight car according to the condition of rolling the wheel crest onto the rail and out of the way with the different technical conditions of the running gear and track [3].

In the freight park of the company, there are 65,521 cars that belong to it. Gondola cars occupy 49.8% of the parking lot, which is the majority. Other car types in the car park are distributed as follows: covered (15.2%), platforms (5.1%), tanks (9.9%), others (20.0%). Freight wagons carry freight loads with an acceptable axial load of 230 kN or less.

Four-axle cars with a capacity of 90 tons (axial loads of about 294 kN) and a car fleet with loads of up to 340 kN on trains weighing 12-20 thousand tons have been manufactured and sold in the United States, Canada, Australia, and other countries since the 1960s. Freight car bodies made entirely of aluminum alloys are commonly used by foreign manufacturers to reduce car container weight to 17-23 tons with a load capacity of 117-120 tons, resulting in a significantly lighter weight for car containers with a load capacity of 27.5 tons to 7 lbs [4, 5].

In contrast, a 1520 mm freight car has a low load capacity (60-70 tons), requires additional expenses for loading, unloading, and mounting of goods, and has a low level of specialization, while the axial load is 230 kN, and the weight of containers is 240 kN.

For the cart of the freight car model 18-100, rational values of inclination angles of the contact surfaces of friction pairs "above-pressure beam - friction wedge" are defined, at which friction forces in pairs "friction wedge - friction strip," which serve to reduce force action on the side of the above-pressure beam, have the highest values.

Structurally, the spring-friction set of the cart of the freight car (Figure 1), as a physical object from the point of view of theoretical mechanics and as a wedge mechanism from the point of view of the theory of mechanisms and machines, is made so that its friction wedges are contacted only by three solid elements - the above-pressure beam 1, friction bar 4 (or 5) and double springs 7 (or 8) [5].

The aim of the study is to develop a mathematical model of the spring-friction unit of the trolley of freight cars in accordance with the profile of the track. According to the theoretical analysis carried out and the goal set, the following tasks are set in the article [1]:

1. Development of a computer and mathematical model of the force action of friction pairs of spring-absorbing apparatuses in a freight car trolley;

2. To make a mathematical model of how the forces of friction pairs work and to check the balance of the spring beam and friction wedges as real-world object;

3. Analytically in the MathCAD environment to determine the reaction of bonds depending on the geometric parameters of the friction pairs.

4. Determine the reaction of the friction bar to the friction wedge.

5. To obtain graphical dependences of the bond reaction on the variable geometric parameters of the friction pairs "spring beam-friction wedge."

2. Methods

The railway track is not strictly straight in plan and has various irregularities and curvatures. These irregularities and curvatures may be intentional, that is, objectively useful (curves and arrows), or accidental (harmful), allowed during the laying and repair of the track, or formed during its operation due to residual deformations and wear. The track and running gear of the rolling stock are charged with ensuring a reliable direction of movement of wheel pairs in straight and useful curvatures of the track and, if possible, not transmitting harmful curvatures to the sprung parts of it. This problem is solved due to the appropriate choice of the shape of the profiles of the rolling surfaces of wheelsets and rails.

The dynamic qualities of railway crews are often associated with a margin of stability. The coefficient of stability margin against rolling it onto the rail head is an integral indicator of traffic safety, since it is determined taking into account vertical and frame forces, friction forces in contact of the wheel crest with the rail, and the geometric parameters of the wheelset. The stability margin related to the wagon-track system is mainly defined as the stability margin from derailment. It was revealed that one of the reasons for the derailment of wagon wheels in curves is the occurrence of significant lateral forces between the wheel and the rail from the action of the moment of resistance to the cart turning under the body. The descent is the first and most serious indicator of the consequences of the incorrect interaction of the car and the track [6].

Theoretical and experimental studies have established that an empty car is more sensitive (in terms of derailment) to the wear of the running gear, to the irregularities of the track gauge, etc., than a loaded one. The probability of derailment of empty wagons, all other things being equal, is much higher than that of loaded ones.

The article discusses the model and develops the procedure for calculating the reaction of the connections of the spring-friction set of the freight car trolley, which takes into account the possible edge contact of friction bodies with separate faces (Figure 1). Therefore, when creating the calculation model of the friction wedge as the subject of the study, it would be necessary to replace the surface of its contact on the side of the pressure beam and the friction level as external connections with only two coupling reactions in the form of replacing the pressure beam and replacing the friction level [6, 7].

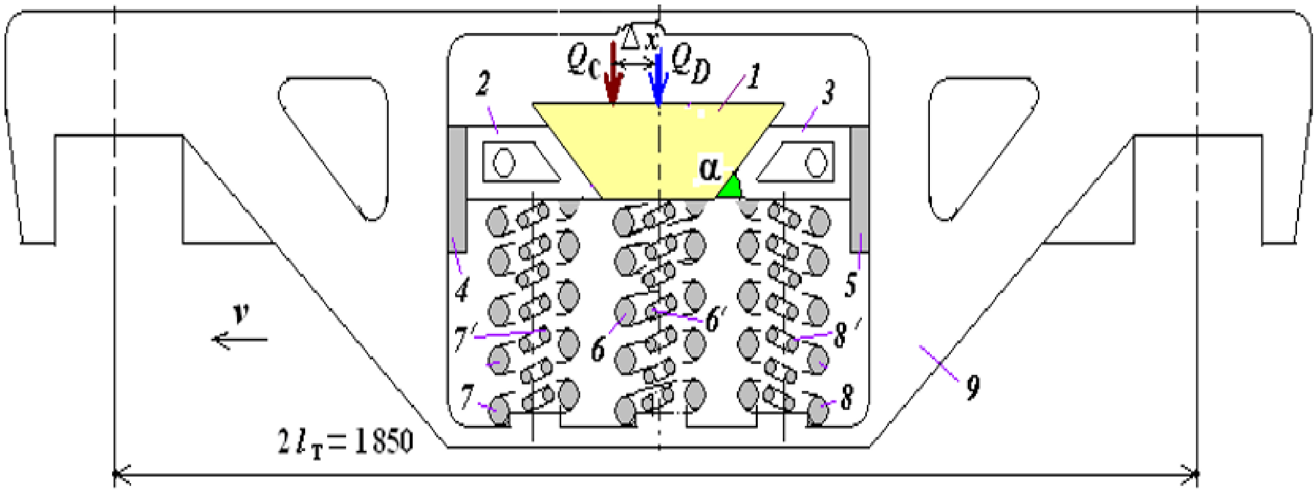

Figure 1.
Spring-friction set of truck loaded with pressure force of car with weight: 1 - back part of above-pressure beam; 2 and 3 - friction wedges; 4 and 5 - friction bars; 6 - sets of springs under the above-pressure beam; 7 and 8 - sets of subcline springs; 9 - side frame.

In the work, the reaction $\overline{R}_1$ has been replaced by four components $\overline{K}_1$ and $\mu\overline{K}_1$; $\overline{K}_2$ and $\mu\overline{K}_2$ (where μ - is the coefficient of friction) instead of two and $\overline{R}_2$ also four components $\overline{B}_1$ and $\mu\overline{B}_1$; $\overline{B}_2$ and $\mu\overline{B}_2$ instead of two, which contradicts the classical principles of theoretical mechanics.

Let's especially note, that at contact (contact) of two solid things, there is inadmissible replacement of one reaction $\overline{R}_1$ of communication with two normal components of this reaction $\overline{K}_1$ and $\overline{K}_2$ (similarly, $\overline{R}_2$ - normal components $\overline{B}_1$ and $\overline{B}_2$). The same observation applies to friction force, since it is a tangent component of bond reactions $\overline{R}_1$ (or $\overline{R}_2$), so it is unacceptable to represent it by two friction forces in the form $\mu\overline{K}_1$ and $\mu\overline{K}_2$ ($\mu\overline{B}_1$ or $\mu\overline{B}_2$). Besides, the presented design model does not show the angle of inclination of the contact surfaces of solid bodies α_1, β_1, γ_1 and coordinate system [8, 9].

On the basis of this, it is possible to note the incorrect representation of the reaction projection of the links $\overline{R}_1$ and $\overline{R}_2$ on the axis of coordinates when the above-pressure beam moves down (loading) and up (unloading) and the moment of forces relative to the point of application of the reaction, which cannot be checked (due to the absence of angles of inclination of the contact surfaces of hard things and coordinate system in the calculation model).

At the same time, we will note that in a projection of reaction of communications $\overline{R}_1$ and $\overline{R}_2$ to axes of coordinates, functions of loading from a spring beam $\overline{Q}_C$, elastic force from the double springs $\overline{F}_6$ and three corners (for example, α_1,

β_1, γ_1) representing three inclined contacted surfaces of a frictional wedge. Here [α_1] 1 is the angle of inclination of the friction wedge with horizontal (blunt angle), which provides the wedge contact with the above-pressure beam; β_1 -is the angle of inclination of the friction wedge with horizontal, which provides the wedge contact with the friction strip vertically; and γ - the angle of inclination of the friction wedge relative to the transverse axis of the opening of the side frame, which provides for contact of the wedge with the friction strip along the horizontal [10].

In China, a concept has been developed that explains derailment by an energy criterion, that is, by a certain limit level of energy that is released when a wheelset hits an uneven track [11]. The numerical feature of the oscillation process, especially the spreads of the forces acting on the wheelset, is used to make a specific assessment of the energy criterion. This is how the energy transferred from the path's unevenness to the wheelset is found.

Based on studies on the model, it is proposed to take a time interval of 0.035 seconds after going beyond the boundaries of the normalized value of the coefficient of stability margin from rolling into the rail head as a condition of descent.

As the main safety criterion, it is accepted as a safety condition that both wheels remain in contact with the rails at any given time. The loss of contact between one of the wheels is considered a potential possibility of derailment. Boundary traffic conditions are calculated for different traffic conditions, which makes it possible to assess traffic safety according to this condition.

A brief analysis of the safety criteria shows that, depending on the capabilities and complexity of the model, it can be based on:

a) a power attribute, that is, the ratio of lateral and vertical reactions on the wheel of a certain value of the time of the duration of the output beyond the normalized value;

b) kinematic, that is, a certain amount of transverse or vertical displacement of the wheel, taking into account time, etc.- for more complex models that implement the kinematics of motion, not the vibrations.

Considering asymmetric placement of the solid-state load relatively as longitudinal (towards the front trolley) as well as the transverse (towards the outer rail thread) axis of symmetry of the car, it is necessary to obtain analytical formulas for determining the reaction of bonds in friction pairs "pressure beam-friction wedge" and "friction wedge-friction strip" depending on their geometric parameters (angles of inclination of the surfaces in contact), which allow finding their rational values.

A review and analysis of the existing calculation methods for the dynamic qualities of the car showed that currently there are not enough methods for assessing the "explosion hazard" of a freight car, which would include: systematization and ranking of the parameters of the way-car-cargo system affecting traffic safety; a mathematical model of the car-way system, adaptive to the degradation changes of this system; the use of the vanishing criterion for practical tasks of preventing rolling stock derailments; and developing recommendations for the maintenance of wagons and tracks depending on speed [12].

3. Results and Discussion

In order to solve the set applied task, we will use the most important provisions of theoretical mechanics: the principle of freedom from ties, the condition of balance of forces in statics, Coulomb's Law, and the axiom of equality of action and counteraction. Let us show, for example, matrix, symbolic, and numerical solution to the problem in the MathCAD environment [13, 14].

Consider the unsymmetrical arrangement of the unit weights relative to the longitudinal and transverse axes of symmetry of the car, for example, towards supports B and D, by the values and (Figure 2).

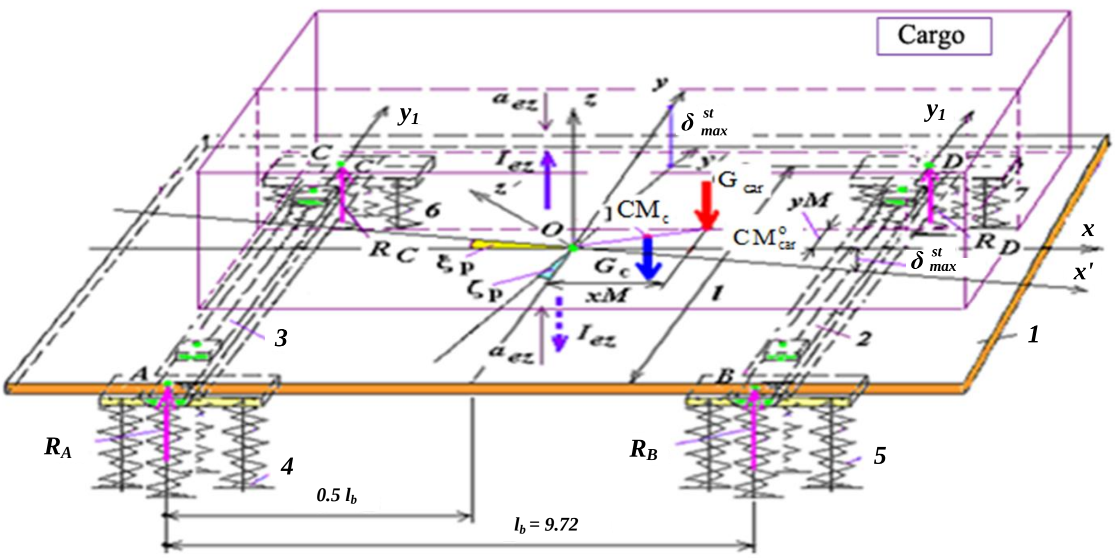

Figure 2.
Arrangement of load with displacement of common center of mass: 1 - car frame, 2 and 3 - above-pressure beams, 4-7 - spring sets.

Location of common center of mass of load CM^{o}_{car} relative to transverse xM and longitudinal yM axis of symmetry of car (m) is taken depending on weight of load and height of common center of mass of car with load. Thus, for example, for the weight of the load.

G = 650 kN (65 ts) and the height of the general center of gravity of the car with the load above the rail head level less than 1500 mm xM = 0.820 and yM = 0.171 m.

At the same time, the frame of the car and above-pressure beams of the front and back cart, because of the shift of the center of mass CM^{s}_{car} of the mechanical system "freight car frame," will be inclined towards this support on corners ξ_p and ζ_p, overloading sets of springs of a support A and B (or C and D) and unloading the same springs of a support C and D (or A and D). In such an inclined position of the frame of the car with load and the above-pressure beams, there will be a car with load as part of the train (Figure 3).

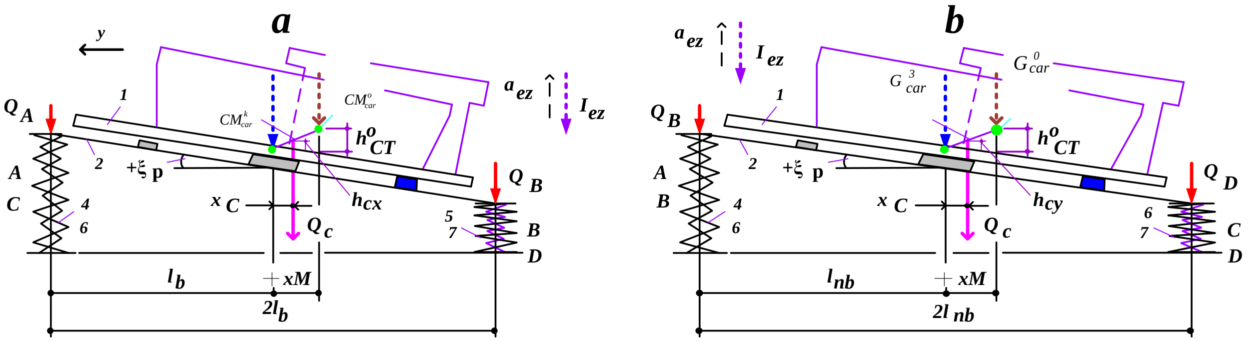

Figure 3.
Redistribution of pressure force of car frame to sets of truck springs. Load displacement: *a* - along the wagon; *b* - car break.

Then the pressure force of the above-pressure beam in the form of $\overline{Q}_D$ (or $\overline{Q}_C$) will act on sets of springs of carts of the freight car. Each set of springs consists of a double row of springs, one of which with a smaller average diameter D_{ia} is placed inside the other with a larger average diameter D_i, $D_{ia} < D_i$ at that (i = 6, 7, 8). Such a structure of the spring sets is applied when the maximum tangential stress of a spring with a larger average diameter D_i on the order of 50% exceeds the allowable stress $[\tau]$. Therefore, a second spring with a smaller average diameter is further introduced to discharge the outer spring D_{ia}.

Consider cases where the rolling stock is moving along a curved section of the track and consider that the frame of the car with the load is only in contact with the surface of the slide with the polymer demfer of the model 18-578 trolley. Let the force $\overline{Q}_c$ act on the slide at some distance *Δx* from its transverse axis (or side frame), and the force $\overline{Q}_D$ (or $\overline{Q}_C$) - on the thrust bearing of the above-pressure beam (Figure 1). Such application of load to spring friction set of model 18-578 trolley is connected with feature of design of elastically-roller slides with polymer damper with limitation of roller from falling out (Figure 4) [15, 16].

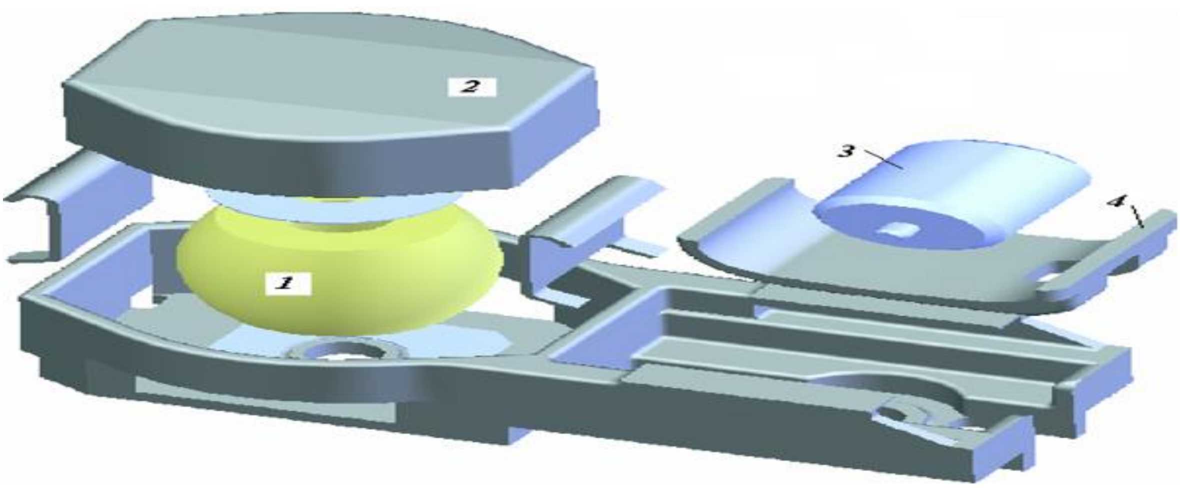

Figure 4.
Elastic-roll slide of model truck 18-578: 1 - polymer damper; 2 - slipper; 3 - roller; 4 - roller limiter.

At the same time, the spring-friction set of the trolley is located on the side of the external rail thread. Let us bear in mind that elastic forces of sets of subcline springs also exert pressure on the pressure beam and friction wedges, through which and on the side frames of car bogies [17, 18]. Besides, reactions (elastic forces) F_6 of sets of springs under the above-pressure bars (5 pcs) *6* will resist downward movement of the above-pressure beam *1*, and reactions (elastic forces) F_7 and F_8 sets of subcline springs *7* and *8* will resist downward movement of friction wedges *2* and *3* (Figure 1).

Frictional levels of *4* and *5* (Figure 1) have to have surely not parallelism in the vertical plane (corners of inclinations concerning a horizontal $\beta_1 = 89^\circ - 1^\circ$ and $\beta_2 = 91^\circ + 1^\circ$), at the same time the distance between frictional levels has to be below 4 - 10 mm more, than above. Not parallelism of frictional levels across - no more than *3* mm (corners of inclinations of surfaces of contacts of frictional wedges *2* and *3* with frictional levels of *4* and *5* rather cross axes of an aperture of a side frame of the car are respectively equal $\gamma_1 = 88^\circ - 1^\circ$ and $\gamma_2 = 92^\circ + 1^\circ$). According to the principle of release from ties, the design models of the pressure beam *1* and friction wedges *2* and*3* of the cargo trolley 18-578 will be presented as shown in Figure 5.

Considering balance of anabove-pressurebeam *1* (Figure 5*a*) we are convinced that *1*affect an above-pressure beam: reaction $\overline{R}_{21}$ and $\overline{R}_{31}$ frictional wedges *2* and *3* which are displayed on normal and tangent components of - N_{21}, N_{31} and $F_{\tau 21}$, $F_{\tau 31}$; active forces $\overline{Q}_c$, $\overline{Q}_D$ (or $\overline{Q}_C$) acting on the side of the car frame with the load, and reactive forces in the form of equal reaction of sets of springs *6* $\overline{F}_6$, and in compliance with condition $F_{62} < F_6 < F_{63}$ (springs *62* and *63*in Figure 1 and 2 are not shown). At the same time, we consider that inclined surfaces of the above-pressure beam are made with errors, i.e., where and are angles of inclination of surfaces of the above-pressure beam to the horizon, rad. ($\alpha_1 \approx 134^\circ 30' + 1^\circ$, $\alpha_2 \approx 45^\circ 30' + 1^\circ$).

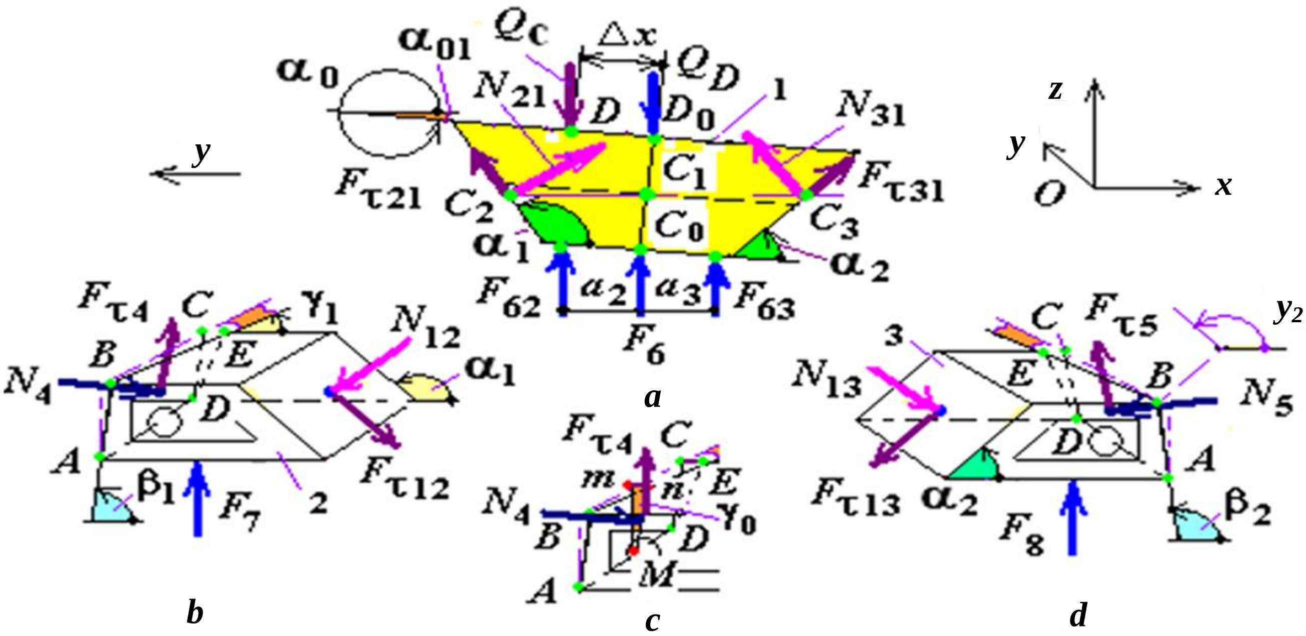

Figure 5.
Design models of above-pressure beam 1 and friction wedges 2 and 3 of freight trolley 18-578.

At an analytical research we will assume that corners of inclinations of surfaces (α_1 and α_2) a above-pressure beam *1*, frictional wedges of *2, 3* and frictional levels *4, 5* (β_1, β_2 and γ_1, γ_2) have various values ($\alpha_1 \neq \alpha_2$, $\beta_1 \neq \beta_2$ and $\gamma_1 \neq \gamma_2$),that correspond or to their production with errors, or consider uneven wear of their surfaces. We also assume that the sliding friction coefficients f between the contacting surfaces of the pressure beam (f_1 and f_2), friction wedges and strips (f_3 and f_4) have different values.

Due to the application of active forces $\overline{Q}_c$, $\overline{Q}_D$, (or $\overline{Q}_C$) acting from the side of the car frame with weight through the elastically-roller slide of constant contact with the polymer damper of the model 18-578 trolley at some distance Δx (according to the drawing 66 mm) from the axis of symmetry of the above-pressure beam, this beam will turn around its longitudinal axis by some angle α_0 (or sharp angle $\alpha_{01} = 2\pi - \alpha_0$). Acute angle is α_{01} found by formula [19]:

$$\alpha_{01} = \operatorname{arctg}\left(\frac{\Delta z_D}{\Delta x + a_0}\right), \qquad (1)$$

Where Δz_D –extreme value of vertical movement of a point of application (in Figure 5 a point D) external loading $\overline{Q}_D$ (or $\overline{Q}_C$) on a above-pressure beam *1*, mm (according to the drawing of 8±2 mm);

a_0 – distance from an axis of symmetry of a above-pressure beam to an axis of a skating rink elastic, mm (according to the drawing of 90 mm).

It is possible, that the possibility of redistribution of forces $\overline{Q}_D$ (or $\overline{Q}_C$) between sub cline springs and friction bars [20] is not excluded. Thus, for example, the double springs *62, 6* and *63*. which are in direct contact with the pressure beam *1* will be reset. As a result, there will be a possibility that the central double springs and the friction wedge *2* are unloaded, i.e. the condition $F_{62} < F_6 < F_{63}$ is quite possible.

In case, if the frame of the car with the load is in full contact with the surfaces of the elastic-roller slides with the polymer damper (I.e. surfaces of slide and roller), which corresponds to the application of external load $\overline{Q}_D$ (or $\overline{Q}_C$) at the point D_0 (Figure 5), then Δz_D = 0 and accordingly α_{01} = 0 and F_{62} = F_6 = F_{63}, i.e. symmetrical application of external load relative to longitudinal axis of symmetry of above-pressure beam, as in bogie of model 18-100 [14, 15].

Elastic forces of sets of springs F_{62}, F_6, F_{63}, F_7 and F_8, and are determined by multiplying the stiffness coefficient c_i (kN/m) by their vertical displacement $\Delta z_i = \delta_i$ (m),, which are found by force characteristics of springs in the form of Ghazavi and Taki [6]:

$$F_i = |c_i \delta_i| \qquad (2)$$

For carts of freight cars of model 18-578 statically the deflection δ_{st} under gross weight reaches 68 mm (0.068 m) (while at the empty car reaches 13 mm), and model 18-100 are in limits of 46 – 50 mm (0,046 – 0,050 m).. Spring deflection must be less than static, i.e. $\delta_i < \delta_{st}$.

Stiffness of external and internal springs is found according to data of Federal State unitary enterprise "Scientific and Production Corporation "Uralvagonzavod," kN/mm:

$$c_{01} = \frac{1}{3.267} = 0.306,$$

$$c_{02} = \frac{1}{7.088} = 0.141.$$

Then the stiffness of the double springs is, kN/mm:

$$c_1 = c_{01} + c_{02} = 0.447$$

At that, elastic forces of double springs F_6, F_{62}, F_{63}, F_7 and F_8, and, which are directly in contact with pressure beam 1 and friction wedges *2* and *3* (refer to Figure 5, *a*), are found by formulas:

$$F_6 = c_1(\Delta z - \Delta z_{D0}),$$
$$F_{62} = 2c_1(\Delta z - \Delta z_D),$$
$$F_{63} = 2c_1 \Delta z,$$
$$F_7 = c_1(\Delta z - \Delta z_D),$$
$$F_6 = c_1 \Delta z, \qquad (3)$$

Based on the performed analytical studies of the spring-absorbing devices of the bogie model 18-578 of the freight car, the following can be noted [18].

1. The developed calculated and constructed mathematical models of the force action of friction pairs "spring beam-friction wedge" and "friction wedge-friction bar" of spring-absorbing devices of the bogie model 18-578 of a freight car are generalized, in a particular case from which such models of bogies 18-100 can be obtained.

2. The constructed generalized mathematical models of the force action of friction pairs "over-spring beam-friction wedge" and "friction wedge - friction bar" of the trolley model 18-578 (in the special case and model 18-100) allowed us to derive the equilibrium equations of the over-spring beam and friction wedges as physical objects.

3. The derived linear algebraic system of linear equilibrium equations of the spring beam was solved using an analytical (symbolic) method in MathCAD, made it possible to obtain analytical formulas for determining the reaction of

bonds depending on the geometric parameters (angles of inclination of the contacting surfaces) of friction pairs, allowing to find their rational values.

4. The results of the research can be used in the design of the contacting surfaces of the friction pairs "pressure beam-friction wedges" and "friction wedges-friction bars".

4. Conclusions

Summarizing the results of the analytical studies performed, note the following:

1. The developed design and constructed mathematical model of the force effect of friction pairs "above-pressure beam-friction wedge" and "friction wedge-friction bar" of the spring-absorbing vehicles of the bogie model 18-578 of the freight car are generalized, in particular case from which such models of truck 18-100 can be obtained.

2. The generalized mathematical model of force action of friction pairs "above-pressure beam-friction wedge" and "friction wedge-friction bar" of the bogie model 18-578 (in particular case and model 18-100) allowed deriving equations of equilibrium of above-pressure beam and friction wedges as physical objects.

3. The derived linear algebraic system of linear equations of equilibrium of the above-pressure beam is solved using an analytical (symbolic) method in the medium MathCAD, making it possible to obtain analytical formulas for determining the reaction of bonds depending on geometric parameters (angles of inclination of surfaces in contact) of friction pairs, which allow to find their rational values.

4. The results of calculations of the reaction of the friction bar *5* to the friction wedge *3* showed that in the friction pair "friction wedge *3* - friction bar *5*" external load $\overline{Q}_c$ ($F_{\tau 5}$ = 170.6 kN) applied to the pressure beam *1* is practically reduced by 1.3 times ($F_{\tau 5}$ = 133.1 kN), however 1.28 times more than friction force ($F_{\tau 4}$ = 103.8 kN), which occurs on the contact surface of friction wedge *2* with strip *4*. Accordingly, the friction force between the friction pairs "friction wedge *3* -friction bar *5*" is greater, i.e. the friction bar *5* is heavily worn relatively to the friction bar *4*. This process of friction pair wear "friction wedge *3* - friction bar *5*" definitely takes place in the practice of model 18-578 carts.

5. Graphical dependencies of reaction of connections from varied geometric parameters (angles of inclination of surfaces in contact -– α_1 and α_2) of friction pairs "above-pressure beam-friction wedge" are built, on the basis of which their rational values can be determined, at which friction forces in pairs "friction wedge-friction strip" have the greatest values. For example, for the model 18-100 bogie, rational values of the inclination angles of the surfaces to be contacted are obtained -– $\alpha_1 = 126^o$ and $\alpha_2 = 54^o$.

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