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Influence of the instability form on the traffic safety indicator of freight rolling stock

Angela Shvets *[1]

[1] *Ukrainian State University of Science and Technologies, Department of Engineering and Design Specialized Department «Microprocessor-Based Control Systems and Safety in the Railway Transport» (EDSD MBCSS), Ukraine, angela_shvets@ua.fm*

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Abstract

Knowledge of the laws of train movement under various control modes is necessary when programming the equations of train movement when it is necessary to determine the exact position of the train on the railway track and the stability of the wagons at the time of interest. In this regard, one of the main areas of research is the analysis of ensuring the safety of cargo transportation, as well as the stability of wheels from derailment influenced by the longitudinal forces and the form of loss of stability of freight wagons in the train. When considering the stability of a freight car as a rod system, the problem of instability of the I kind (Eulerian instability) was solved, and the efforts and displacements in the compressed-bent rods were determined using the deformation (displacement) method. As a result of theoretical studies, the values of the factor of stability against lift by longitudinal forces were obtained, taking into account the forms of instability. The relevance of this study relates to the need to control the longitudinal forces arising during the train movement, taking into account the increase in speeds, masses, and lengths of trains (especially freight trains) and the locomotive power increase.

1. Introduction

The rail industry is responsible for the mass transportation of vast quantities of goods, and its needs have changed significantly in recent years. Ensuring the safe and timely transportation of goods is one of the main tasks of railway transport. The role of traffic safety questions increases significantly with an increase in train speeds. In this regard, one of the main areas of research is the selection of advanced concepts for analyzing the improvement of operational performance during international cargo transportation. In modern research, much attention is paid to the questions of increasing speeds and loads, ensuring transportation safety, as well as sustainability [1-4].

Elevating the maximum speeds makes it necessary to increase the braking efficiency of the rolling stock. The main limitation of the magnitude of the braking force of the rolling stock is the force of adhesion of the wheels to the rails and the stability of the wheel from the derailment [5-7]. Numerous studies make it possible to obtain the absolute values of the longitudinal dynamic forces during braking and also demonstrate that the forces depend on the weight and length of the train, brake parameters, train speed, braking mode, characteristics, and condition of the draft gear, the size of the gaps in the shock absorbing elements and traction devices, and their distribution along the length of the train at the time of the start of braking [8-10].

Knowledge of the laws of movement of a slowed-down train is necessary when programming the equations of train movement when it is necessary to determine the exact position of the train on the railway track at the point of interest. In the presence of correctly compiled train motion equations, it is not particularly difficult to accurately calculate the length of the braking distances and evaluate the effectiveness of various braking systems [11-14].

The main operational parameter is the longitudinal quasi-static compressive force in the train [15-18]. The priority in improving train control technology is to reduce this force factor in the process of operational work, as well as to determine the critical compressive forces for various forms of wagon instability in the train (Figure 1) [19-22].

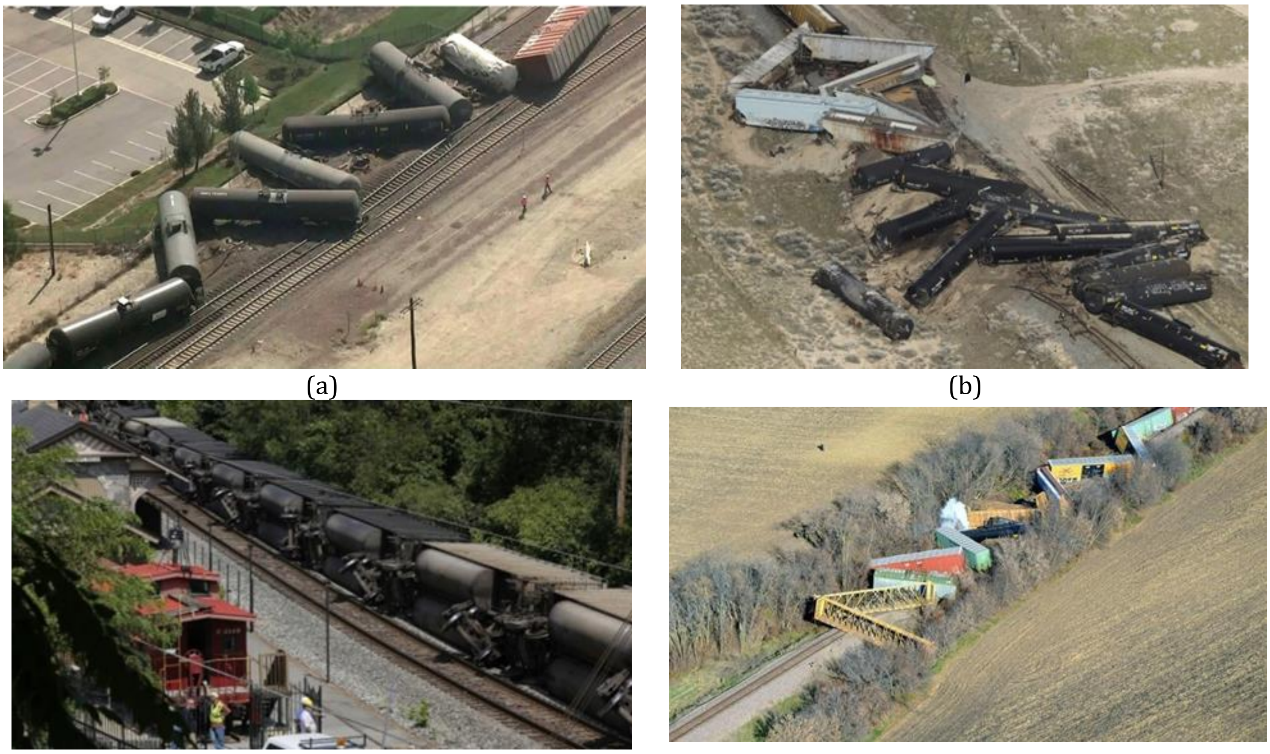


Figure 1. Train derailment in: a – San Bernardino 21.08.2018; b – Eureka, south of Salt Lake City 02.04.2019; c – Maryland 21.08.2012; d – Ellendale, Minn. 11.11.2016

Ensuring the safety of the movement of heavy trains is possible only if there is a well-controlled brake that does not cause large longitudinal forces in the composition under any braking modes. Therefore, it is necessary to more accurately investigate the dependence of the wheel stability coefficient on derailment on various factors and develop measures to increase the braking efficiency of the rolling stock. The purpose of the paper is a theoretical study of the influence of longitudinal forces of a quasi-static nature and the instability form of freight wagons in a train on the stability of a wheel from the derailment.

2. Material and Method

It is known that when carrying out traction calculations and solving problems related to the optimization of energy costs for traction, the train is considered as a one-dimensional mechanical system of solid bodies connected by elastic-viscous bonds [23-30]. When studying a train as a hinged-link system, the following forms of stability loss are possible (Figure 2) [31-34].

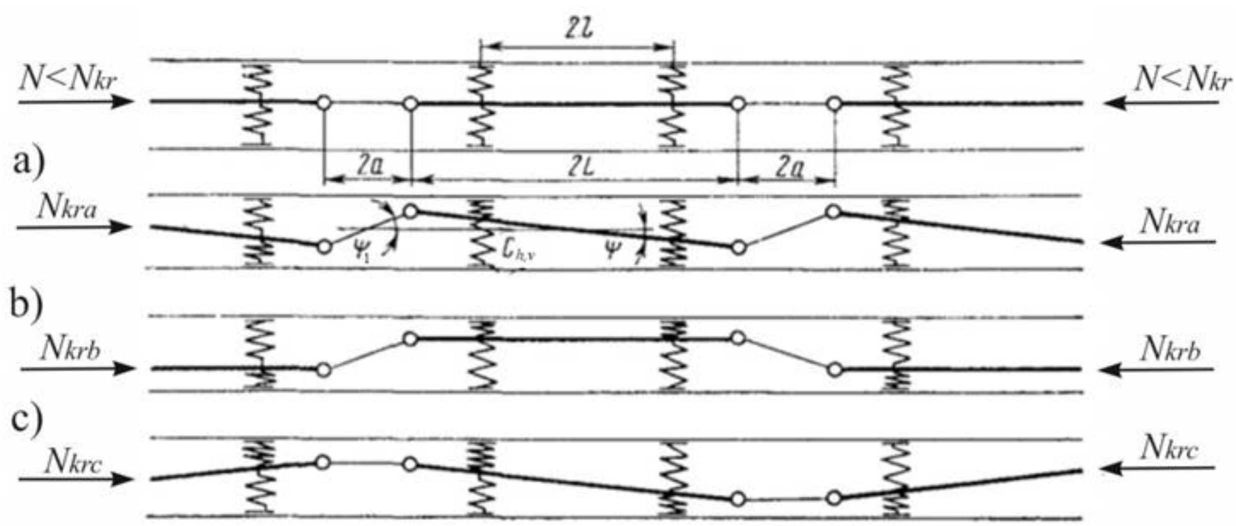


Figure 2. The layout of wagons in the train in case of instability during the transfer of longitudinal force in the horizontal plane

Each case of applying forces to the wagon in the horizontal plane should be considered together with the options for installing the wagon in the vertical plane (Figure 3) and separately for tensile and compressive longitudinal forces on straight and curved sections of the track.

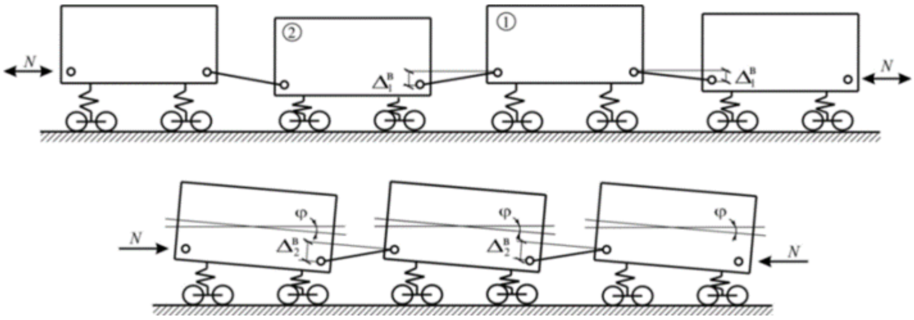


Figure 3. The layout of wagons in a train during the wagon stability loss in a vertical plane

In total, more than 100 options for installing a wagon on a track should be considered. Based on the analysis of a large number of options, a conclusion was made about the most unfavorable schemes for installing wagons on the track [31-34].

2.1. Algorithm for determining the critical compressive force

An increase in the weight and length of trains leads to the need to consider the body of a freight wagon, taking into account the tare weight of the wagon and weight of the cargo, as an elastic massless beam carrying a uniformly distributed load (Figure 4) [34, 35].

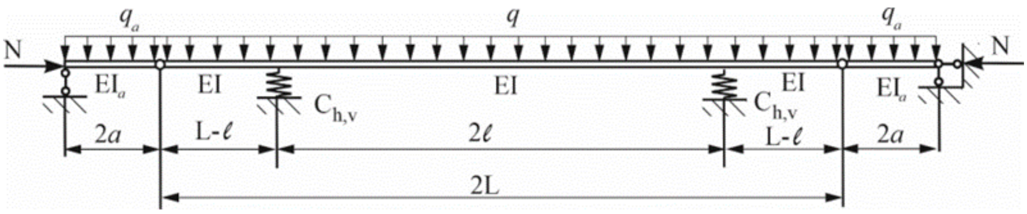


Figure 4. Scheme of a freight wagon, taking into account its weight and loading mode

Here q_a – own weight of two coupler assemblies, respectively related to two lengths of coupler bodies; q – the empty weight of the wagon body together with the suspended equipment and two bolsters in an empty state, referred to its length. When taking into account the loading, the cargo weight is added to the body weight and is considered to be evenly distributed along the entire length; 2ℓ – wheelbase; $2L$ – the distance between coupler followers; $2a$ –automatic coupler body double length from a pulling face to the shank end; $C_{h,v}$ – the horizontal (vertical) stiffness of the spring suspension of one bogie. Let us assume the following: $\ell_1=\ell_5=2a$, $\ell_2=\ell_4=L-\ell$, $\ell_3=2\ell$.

The nominal bending stiffness of a gondola wagon body is approximately equal to three times the stiffness of the center sill (in the corresponding directions) [34]. In works [36, 37], the most unfavorable sections of an automatic coupler from the point of view of strength were established. Rod system in the displacement method has a degree of static indeterminacy equal to 6. The table of reactions of compressed-bent rods from single displacements and loads is given in the work [38]. The basic system of the displacement method, taking into account the symmetry of the rod system, is shown in Figure 5.

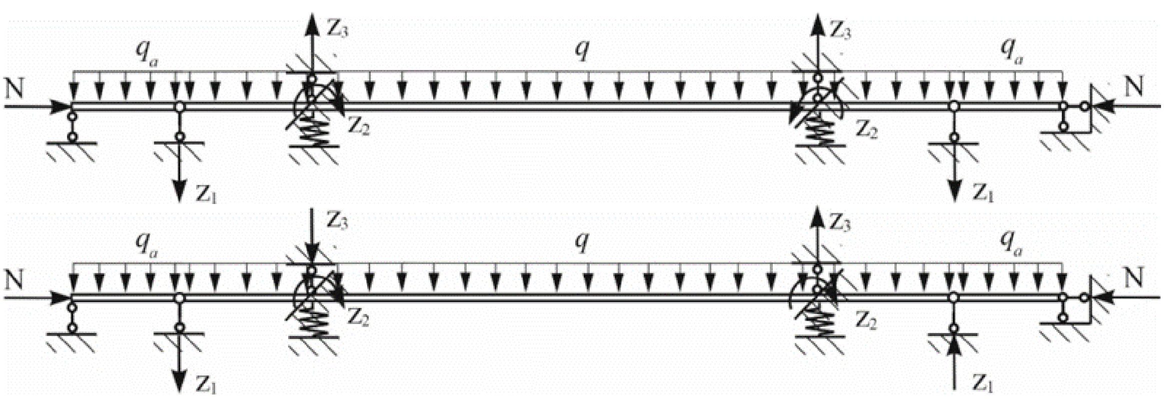


Figure 5. The basic system of the displacement method, taking into account the symmetry of the rod system

When searching for the minimum critical forces of a symmetrical and symmetrically loaded system, it suffices to find two smaller critical parameters for a direct-symmetrical and skew-symmetrical grouping of unknowns. Expressions for the functions of the displacement method for compressed-bent rods are taken in accordance with [38]. The critical parameter v_i or the length reduction factor, which depends on the instability form, is determined by expanding the determinant composed of expressions for the coefficients at unknown [34, 35].

A theoretical study [35] made it possible to obtain dependences for determining the critical parameter for some instability forms, taking into account the rigidity, the weight of the elements of the hinge-rod system, and the gap in the rail track. The following designations for the functions of the displacement method for compressed-bent rods are accepted:

$$v = \ell \cdot \sqrt{\frac{N}{EI}},\ v_1 = \frac{\ell_1}{\ell_3\sqrt{k}} v_3,\ v_2 = \frac{\ell_2}{\ell_3} v_3,\ N_{kr} = \frac{v_3^2 EI}{\ell_3^2}. \tag{1}$$

The critical parameter for a skew-symmetric deformed state (II-nd form) with a directly symmetrical grouping of unknowns, taking into account the loading of the wagon (empty or loaded), is determined by expanding the determinant, consisting of expressions for the coefficients at unknowns according to the dependencies [34, 35]:

$$r_{11} = 2 \cdot \left[-\frac{v_1^2 EI_a}{\ell_1^3} + \frac{3EI}{\ell_2^3} \eta_1(v_2) \right], \tag{2}$$

$$r_{12} = r_{21} = 2 \cdot \frac{3EI}{\ell_2^2} \varphi_1(v_2), \tag{3}$$

$$r_{13} = r_{31} = -2 \cdot \frac{3EI}{\ell_2^3} \eta_1(v_2), \tag{4}$$

$$r_{22} = 2 \cdot \left[\frac{6EI}{\ell_3} \varphi_4(v_3) + \frac{3EI}{\ell_2} \varphi_1(v_2) \right], \tag{5}$$

$$r_{23} = r_{32} = 2 \cdot \left[-\frac{12EI}{\ell_3^2} \varphi_4(v_3) + \frac{3EI}{\ell_2^2} \varphi_1(v_2) \right], \tag{6}$$

$$r_{33} = 2 \cdot \left[C_{h,v} + \frac{3EI}{\ell_2^3} \eta_1(v_2) + \frac{24\eta_2(v_3)}{\ell_3^3} \right]. \tag{7}$$

The second stability equation for a skew-symmetric grouping is based on the difference between the coefficients for the unknowns:

$$r_{11} = -2\left[\frac{q_a \ell_1}{2} + \frac{q\ell_2}{2} \cdot \left[1 - \frac{1}{4\varphi_2(v_2)} \right] \right], \tag{8}$$

$$r_{12} = r_{21} = -2\left[\frac{q\ell_2^2}{8} \cdot \frac{1}{\varphi_2(v_2)} + \frac{q\ell_3^2}{12} \cdot \frac{1}{\varphi_4(v_3)} \right], \tag{9}$$

$$r_{13} = r_{31} = -2\left[\frac{q\ell_3}{2} + \frac{q\ell_2}{2} \cdot \left[1 + \frac{1}{4\varphi_2(v_2)} \right] \right], \tag{10}$$

$$r_{22} = -2\left[\frac{q\ell_3^2}{12} \cdot \frac{1}{\varphi_4(v_3)} - \frac{q\ell_2^2}{8} \cdot \frac{1}{\varphi_2(v_2)} \right], \tag{11}$$

$$r_{23} = r_{32} = -2\left[\frac{q\ell_3^2}{12} \cdot \frac{1}{\varphi_4(v_3)} - \frac{q\ell_2^2}{8} \cdot \frac{1}{\varphi_2(v_2)} \right], \tag{12}$$

$$r_{33} = 2\left[\frac{q\ell_2}{2} \cdot \left[1 + \frac{1}{4\varphi_2(v_2)} \right] + \frac{q\ell_3}{2} \right]. \tag{13}$$

For a symmetrical deformed state (the I-st form of loss of stability), taking into account the longitudinal force and the loading mode of the wagon, using the reciprocity condition in determining the coefficients for unknowns in the canonical equations, received [34, 35]:

$$r_{11} = 2\cdot\left[-\frac{q_a\ell_1}{2} - \frac{v_1^2 EI_a}{\ell_1^3} - \frac{q\ell_2}{2}\cdot\left[1 - \frac{1}{4\varphi_2(v_2)}\right] + \frac{3EI}{\ell_2^3}\eta_1(v_2)\right], \quad (14)$$

$$r_{21} = r_{12} = 2\cdot\left[\frac{3EI}{\ell_2^2}\varphi_1(v_2) + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2(v_2)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4(v_3)}\right], \quad (15)$$

$$r_{13} = r_{31} = 2\cdot\left[-\frac{q\ell_3}{2} - \frac{q\ell_2}{2}\cdot\left[1 + \frac{1}{4\varphi_2(v_2)}\right] - \frac{3EI}{\ell_2^3}\eta_1(v_2)\right], \quad (16)$$

$$r_{22} = 2\cdot\left[\frac{v_3 EI}{\ell_3 \tan\left(v_3/2\right)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4(v_3)} + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2(v_2)} + \frac{3EI}{\ell_2}\varphi_1(v_2)\right], \quad (17)$$

$$r_{23} = r_{32} = 2\cdot\left[\frac{3EI}{\ell_2^2}\varphi_1(v_2) + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2(v_2)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4(v_3)}\right], \quad (18)$$

$$r_{33} = 2\cdot\left[C_{h,v} + \frac{q\ell_2}{2}\cdot\left[1 + \frac{1}{4\varphi_2(v_2)}\right] + \frac{3EI}{\ell_2^3}\eta_1(v_2) + \frac{q\ell_3}{2}\right]. \quad (19)$$

Wagons always have a certain skew relative to each other due to the tortuous movement of crews, the presence of gaps in the track, spring, bushing and rocker units, and other reasons [31-34].

In the studies [34, 35], it was found that for the layout of wagons in the train in Figure 2(a), for a skew-symmetric deformed state with a directly symmetric grouping of unknowns, it is necessary to use the expression for the coefficient r_{11}:

$$r_{11}^a = 2\cdot\left[-\frac{2v_1^2 EI_a}{\ell_1^3} + \frac{3EI}{\ell_2^3}\eta_1(v_2)\right] = \frac{6EI}{\ell_2^2}\times\left[\frac{\eta_1(v_2)}{\ell_2} - \frac{2v_1^2\ell_2^2 k}{3\ell_1^3}\right]. \quad (20)$$

For skew-symmetric grouping of coefficients for unknowns, expression (9) is used for the coefficient r_{11}. For the scheme in Figure 2(c):

$$r_{11}^c = 2\cdot\frac{3EI}{\ell_2^3}\eta_1(v_2) = \frac{6EI}{\ell_2^2}\times\frac{\eta_1(v_2)}{\ell_2}. \quad (21)$$

The critical parameter v_3 for a skew-symmetric deformed state is determined by expanding the determinant, composed of expressions for the coefficients at unknowns from the dependences (20) and (3)-(7), as well as (21) and (3)-(7).

For the layout of wagons in a symmetrical deformed state (Figure 2(b)), when transferring the longitudinal force, the coefficient r_{11} is determined from the dependence [34, 35]:

$$r_{11}^b = 2\cdot\left[-\frac{q_a\ell_1}{2} - \frac{2v_1^2 EI_a}{\ell_1^3} - \frac{q\ell_2}{2}\cdot\left[1 - \frac{1}{4\varphi_2(v_2)}\right] + \frac{3EI}{\ell_2^3}\eta_1(v_2)\right]. \quad (22)$$

The critical parameter v_3 (I-th form) is determined by expanding the determinant, composed of expressions for the coefficients at unknowns according to the dependencies (22) and (14)-(19). The presented dependencies can be used to determine the form of loss of stability of freight wagons both in the vertical and the horizontal plane.

In the study [34, 35], the rail track was represented by one geometric line in the horizontal and vertical planes. However, a railway track is formed by two parallel lines of rails. In addition, an important feature of the running parts of the wagons is the structural possibility of vertical and transverse horizontal displacement of the body relative to the track axis. This is necessary to maintain the cohesive state of the rolling stock when moving along circular and S-shaped curves, as well as hump yards.

Under these conditions, the loss of stability of the hinge-rod system "automatic coupler - wagon body" is necessary. Therefore, it is necessary to establish when a necessary condition becomes dangerous during operational work. In this regard, it is required to develop refined methods to assess the safety of the movement of rolling stock, allowing to determine the moment of "obvious derailment", that is, to assess both the fulfilment of the necessary and sufficient conditions for derailment [31-34]. This leads to the need to study the provision of stability margin from extrusion of the wagon by longitudinal forces and rolling of wheel flanges onto the rail head, taking into account the gap in the rail track 2δ, the angle of rotation ψ, and displacements Δ of the cross section along the stop plates of automatic couplers [34, 35].

When studying the form of loss of stability of freight wagons in the horizontal plane, it is necessary to take into account that the rail track is formed by two geometric lines passing by the inner faces of the rail heads. The angle ψ and displacement of the section along the stop plates of the automatic couplers Δ in the horizontal plane during the compression of the wagon when installed on the track with a deviation of the center plates across the track (skew-symmetric deformed state) by the value δ are determined:

$$\psi = \frac{\delta}{\ell} = \frac{2\delta}{\ell_3},\ \Delta = \psi L = \frac{\delta L}{\ell} = \frac{\delta}{\ell_3}\left(2\ell_2 + \ell_3\right). \tag{23}$$

All missing values of the reactions of supports from displacements (δ, Δ), angle of rotation, and loads were obtained by solving the differential equation for bending a compressed-bent rod.

For a skew-symmetric deformed state (form II) with a directly symmetrical grouping of unknowns, taking into account the loading of the wagon (empty or loaded) and the displacement of the body by the gap δ in the railway track, the critical parameter is determined by expanding the determinant composed of expressions for the coefficients at unknowns according to dependencies (2)-(7):

$$D = \begin{bmatrix} \Delta\cdot r_{11} & \Delta\cdot r_{12} & \Delta\cdot r_{13} \\ \Delta\cdot r_{21} & \psi\cdot r_{22} & \delta\cdot r_{23} \\ \Delta\cdot r_{31} & \delta\cdot r_{32} & \delta\cdot r_{33} \end{bmatrix} = 0. \tag{24}$$

The second stability equation for a skew-symmetric grouping (the difference in the coefficients for unknowns) depends on the movement of the bogies within the gap in the rail track. The determinant consists of expressions for the coefficients at unknowns according to the dependencies (8)-(13).

Using the reciprocity condition when calculating the coefficients of the unknowns in the canonical equations, the following is obtained:

$$r_{11} = 2\cdot\left[-\frac{q_a\ell_1}{2} - \frac{\delta v_1^2 EI_a}{\ell_1^3} - \frac{q\ell_2}{2}\cdot\left[1 - \frac{1}{4\varphi_2\left(v_2\right)}\right] + \frac{3\delta EI}{\ell_2^3}\eta_1\left(v_2\right)\right], \tag{25}$$

$$r_{21} = r_{12} = 2\cdot\left[\frac{3\delta EI}{\ell_2^2}\varphi_1\left(v_2\right) + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2\left(v_2\right)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4\left(v_3\right)}\right], \tag{26}$$

$$r_{13} = r_{31} = 2\cdot\left[-\frac{q\ell_3}{2} - \frac{q\ell_2}{2}\cdot\left[1 + \frac{1}{4\varphi_2\left(v_2\right)}\right] - \frac{3\delta EI}{\ell_2^3}\eta_1\left(v_2\right)\right], \tag{27}$$

$$r_{22} = 2\cdot\left[\frac{v_3 EI}{\ell_3 \tan\left(v_3/2\right)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4\left(v_3\right)} + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2\left(v_2\right)} + \frac{3EI}{\ell_2}\varphi_1\left(v_2\right)\right], \tag{28}$$

$$r_{23} = r_{32} = 2\cdot\left[\frac{3\delta EI}{\ell_2^2}\varphi_1\left(v_2\right) + \frac{q\ell_2^2}{8}\cdot\frac{1}{\varphi_2\left(v_2\right)} - \frac{q\ell_3^2}{12}\cdot\frac{1}{\varphi_4\left(v_3\right)}\right], \tag{29}$$

$$r_{33} = 2\cdot\left[C_h\delta + \frac{q\ell_2}{2}\cdot\left[1 + \frac{1}{4\varphi_2\left(v_2\right)}\right] + \frac{3\delta EI}{\ell_2^3}\eta_1\left(v_2\right) + \frac{q\ell_3}{2}\right]. \tag{30}$$

For the layout of wagons in Figure 2 (a) for a skew-symmetric deformed state (II-nd form of loss of stability), taking into account the total (transverse) gap in the railway track, when opening the determinant (24), it is necessary to use expression (20) for the coefficient r_{11}. For the scheme in Figure 2 (c), dependence (21) is taken. For a skew-symmetric grouping of coefficients at unknowns, the critical parameter is determined by expanding the determinant composed of the expressions (8)-(13). In this case, the critical parameter depends solely on the weight of the elements of the hinged-rod system.

For the layout of wagons in a symmetrical deformed state (Figure 2(b)) when transferring the longitudinal force, taking into account the total (transverse) gap in the railway track δ, the coefficient r_{11} is determined from the dependence:

$$r_{11}^{b}=2\cdot\left[-\frac{q_a\ell_1}{2}-\frac{2\delta v_1^2 EI_a}{\ell_1^3}-\frac{q\ell_2}{2}\cdot\left[1-\frac{1}{4\varphi_2(v_2)}\right]+\frac{3\delta EI}{\ell_2^3}\eta_1(v_2)\right]. \quad (31)$$

The critical parameter v_3 for the symmetric form of loss of stability (I-th form) is determined by expanding the determinant, composed of expressions for the coefficients at unknowns according to the dependences (31) and (25)-(30).

2.2. Determination of the traffic safety indicator of the wagon under the action of a compressive force

As noted earlier, the spatial oscillations of the train are considered as a chain of rigid bodies connected to each other, representing the movement of wagon bodies in the train. The forces acting on the body of each individual wagon from the side of the automatic couplers (Figure 6) depend on the movement of the wagons and the features of the draft gears that the neighboring wagons are equipped with. The dependence of the longitudinal force on the total deformations of draft gears and other elements of the automatic coupling of wagons is assumed to be known [12, 39, 40].

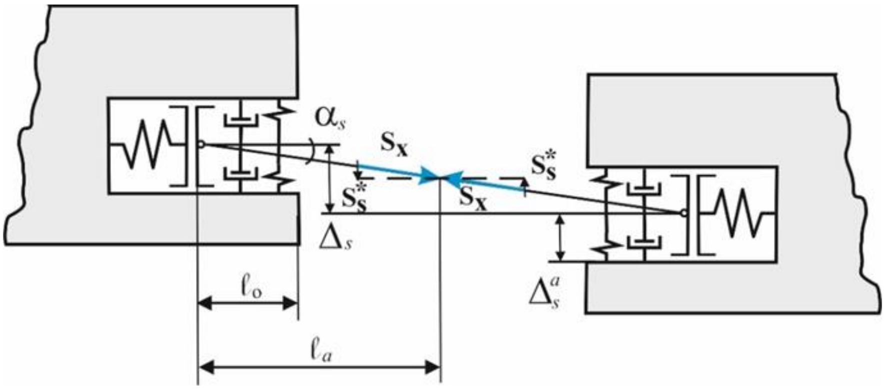


Figure 6. Schematic view of the automatic couplers of wagons

The permissible limits are set for the values of the parameters to assess the safety of the movement of wagons. If they are exceeded, there is a possibility of an emergency situation. The stability indicator of wagons against derailment, as is known, is estimated by the ratio of horizontal transverse (lateral) forces to vertical forces acting at the point of contact of the wheel flange with the rail head [31-34]. The calculation scheme should take into account the possibility of installing a separate wagon on a track, depending on the forces of compression or tension on a curved or straight section of the track, and also reflect the features of the transfer of longitudinal forces to the elements of the bogies in the vertical and horizontal (transverse) directions. The method for determining the stability indicator of the wheelset of a wagon under the action of compressive longitudinal forces was given for straight [34] and curved sections of the railway track [18, 41] in the presence of a difference in the heights of the axles of two adjacent wagons.

The expression for determining the traffic safety indicator on a straight section of the track is as follows [34]:

$$C_{\mathrm{lr}}=\frac{tg\beta-\mu}{1+\mu\cdot tg\beta}\cdot\frac{P_{\mathrm{wag}}^{\mathrm{st}}+\frac{N^2}{C_{\mathrm{h}}}\cdot\frac{\psi_a^2\cdot h_{\mathrm{hs}}}{\delta_0\cdot S}+N\cdot 2\psi_a\cdot\frac{h_{\mathrm{hs}}}{S}}{\mu\cdot P_{\mathrm{wag}}^{\mathrm{st}}+\frac{N^2}{C_{\mathrm{h}}}\cdot\frac{\psi_a^2}{\delta_0}\cdot\left(2-\mu\cdot\frac{h_{\mathrm{hs}}}{S}\right)+N\cdot 2\psi_a\cdot\left(2-\mu\cdot\frac{h_{\mathrm{hs}}}{S}\right)} \quad (32)$$

A similar expression for a curved section of a railway track, taking into account the forces of inertia for the climbing wheelset of the front bogie, has the following form [18, 41]:

$$C_{\mathrm{lr}}=\frac{tg\beta-\mu}{1+\mu\cdot tg\beta}\cdot\frac{P_{\mathrm{wag}}^{\mathrm{st}}+\frac{N^2}{C_{\mathrm{h}}}\cdot\frac{\psi_a^2\cdot h_{\mathrm{hs}}}{\delta_0\cdot S}+N\cdot\left[2\psi_a\cdot\frac{h_{\mathrm{hs}}}{S}+\psi_{\mathrm{cur}}\cdot\frac{h_a}{S}\right]+P_{\mathrm{in}}\cdot\frac{h_{\mathrm{c}}}{S}}{\mu\cdot P_{\mathrm{wag}}^{\mathrm{st}}+\frac{N^2}{C_{\mathrm{h}}}\cdot\frac{\psi_a^2}{\delta_0}\cdot\left(2-\mu\cdot\frac{h_{\mathrm{hs}}}{S}\right)+N\cdot\left[2\psi_a\cdot\left(2-\mu\cdot\frac{h_{\mathrm{hs}}}{S}\right)+\psi_{\mathrm{cur}}\cdot\left(2-\mu\cdot\frac{h_a}{S}\right)\right]+P_{\mathrm{in}}\cdot\left(2-\mu\cdot\frac{h_{\mathrm{c}}}{S}\right)} \quad (33)$$

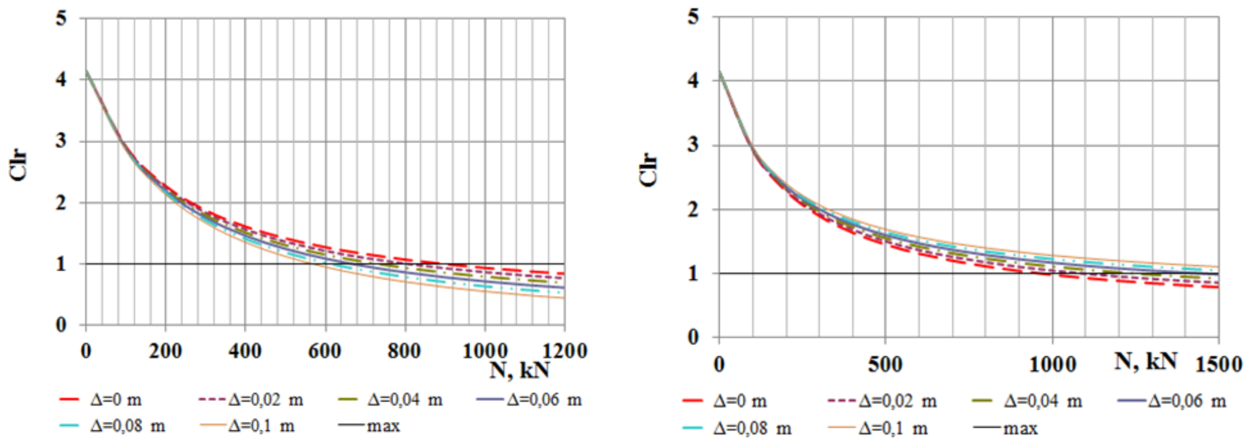
Where $P_{\mathrm{wag}}^{\mathrm{st}}$ – is the static pressure of the wagon, taking into account unloading from the longitudinal force, kN; P_{in} – inertia force from unbalanced acceleration, kN; ψ_a – the angle formed by the longitudinal axle of the automatic coupler body and the axle of the central sill of the wagon frame in a horizontal plane, rad; ψ_{cur} – wagon rotation angle, depending on the location on the curve, rad; $h_{\mathrm{hs}}\approx r$ – height above the level of the rail heads plane to the upper plane of the central spring set, m; $2\delta_0$ – total lateral acceleration of the wagon body frame relative to the track axle in the guiding section along the center pin, m; $2S$ – the distance between rolling wheel circles, m; h_a – automatic coupler axle height above the rail heads level, m; h_c – the height of the wagon gravity center above the rail heads level, m; μ – wheel-rail friction coefficient; β – inclination angle formed by the conical surface of the wheel flange to the horizontal axle.

The dependencies for determining the indicator of traffic safety (32-33) will have differences in the expressions of the static pressure of the wagon $P_{\mathrm{wag}}^{\mathrm{st}}$, taking into account the unloading from the longitudinal force for the front and rear bogies, respectively, in the form of instability [18, 41].

3. Results

Most of the existing methods used to assess the safety of the movement of wagons set permissible limits for the values of the parameters, beyond which there is a possibility of an emergency situation. The stability factor of wagons against derailment, as is known, is estimated by the ratio of horizontal transverse (lateral) forces to vertical forces acting at the point of contact of the wheel flange with the rail head [31-34].

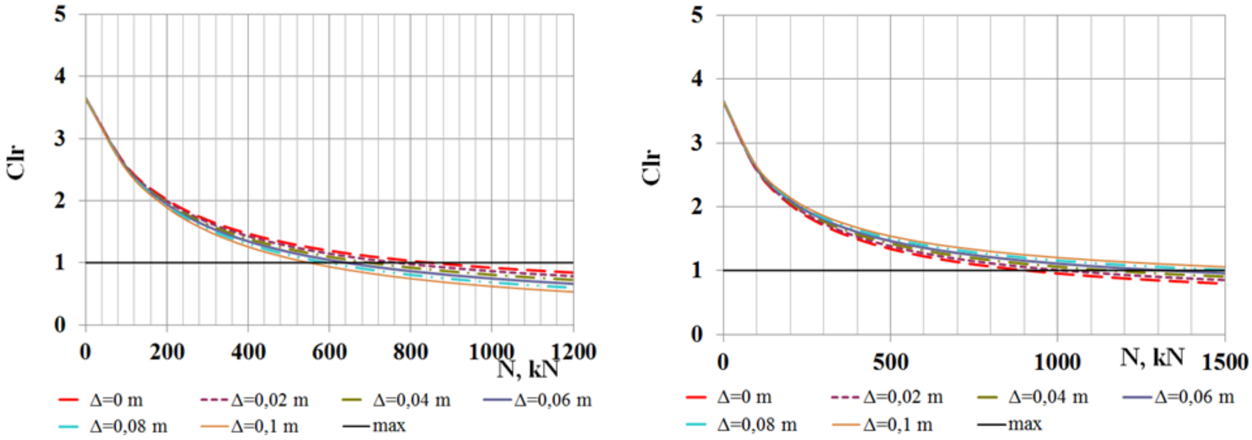
Let us calculate the stability of the wheelset of a wagon under the action of compressive longitudinal forces according to the dependences for straight [34] and curved sections of the railway track [18, 41] in the presence of a difference in the heights of the axles of two adjacent wagons. It is envisaged that the loss of stability of wagons occurs according to the I-st (loaded front bogie, see Figure 7(a) and Figure 8(a) and according to the II-nd form (unloaded front bogie, see Figure 7(b) and Figure 8(b). The calculations took into account the difference in height between the longitudinal axles of automatic couplers in a freight train from 0 to 0.1 m with a step of 0.02 m (Figure 7). Behind the studied wagon, the difference in the levels of the axes of the automatic couplers is taken equal to Δ_2=0.04 m since this value corresponds to the allowable design difference between neighboring wagons.



(a) (b)

Figure 7. Empty wagon traffic safety indicator on a straight section of the track: *a* – unloaded front bogie; *b* – loaded front bogie.

To assess the influence of the form of wagon instability under the action of compressive longitudinal forces on the traffic safety indicator, Figure 8 shows the results of calculations of the movement of an empty wagon along a curve with a radius of R=300 m with an elevation of 150 mm and at a speed of movement equal to 70 km/h.



(a) (b)

Figure 8. Empty wagon traffic safety indicator on a curved section of the track: *a* – unloaded front bogie; *b* – loaded front bogie

Theoretical studies with a difference in the levels of the automatic coupler axles equal to Δ_2=0 m behind the investigated wagon gave dependences similar to those in Figure 7 and Figure 8. The given results confirm that the loss of stability of wagons in the I-th and II-nd form occurs with a significant difference in the magnitude of the longitudinal compressive force. The traffic safety indicator of an empty wagon depends to a much greater extent on the form of instability than on the curvature of the section and the difference in the levels of the axes of the automatic couplers of neighboring wagons. At the same time, it should be noted that an increase in the difference in the levels of the axles of the automatic couplers of neighboring wagons has a positive effect on the traffic safety indicator in the I-th form of stability loss in the case of additional loading of bogies.

4. Discussion

When performing an analytical simulation, the data on the magnitudes of longitudinal forces refer to a single wagon. That is, these forces must be applied to the wagon as part of the train. It is known that if the locomotive in the head of the train applies a traction force or brakes. Then, in some sections of the train, a longitudinal force may arise, under which the stability conditions for the wagon located in this section will not be met. However, such an increase in force does not occur constantly and depends on many factors (the state of the composition, the section number, the rise time of the thrust force, the braking stage, etc.).

The analysis of train crash patterns (Figure 1 and Figure 9) demonstrates that the loss of stability of wagons occurs in various forms of instability and indicates a significant influence of the intensity of the increase in the magnitude of the longitudinal force in the train section from time [42, 43].

(a) (b)

Figure 9. Train derailment in: *a* – Scottish 06.05.2022; *b* – Newcastle, NSW 29.07.2020

Emergency situations in Figure 9 also indicate that loss of stability may occur in 1-2 rolling stock units or a larger group, depending on the proximity of the values of the longitudinal compressive force at the boundaries of this group [42, 43].

Many railways locomotive depots use regime train driving cards to improve the skills of locomotive drivers and teach them rational train driving techniques. The regime cards recommend the most expedient locomotive control methods on specific sections of train traffic in order to: excerpt specific sections of train traffic travel times, travel speed, overcome difficult sections of the profile and track plan, places to check braking efficiency, the possibility of saving electric energy or diesel fuel while maintaining traffic safety. The use of the above methodology for determining critical longitudinal forces when compiling regime cards will make it possible to recommend rational train operation not only at the lowest energy costs but also to implement measures to improve traffic safety.

In order to carry out a continuous analysis of the magnitude of the resulting longitudinal forces under train conditions and to prevent large compressive forces, it is possible to recommend the installation on locomotives of a special pre-calibrated automatic coupler with glued sensors that allow measuring the resulting longitudinal forces. The signals from the specified sensor output to the locomotive control panel will allow the driver to control the magnitude of the resulting longitudinal forces to prevent large values of compressive forces.

5. Conclusion

The use of the method of determining the critical parameter for the I-th and II-nd forms of instability under the action of quasi-static longitudinal forces will allow us to justify the cause of the derailment, as well as to develop and put into practice the technical measures to prevent the lift of the carriages, widening and shear of the track. Using the methodology in compiling the process flow diagrams for driving the trains will make it possible to recommend rational train driving not only at the lowest energy costs but to implement technical measures to improve the stability of freight rolling stock, which in turn will allow removing some existing restrictions on permissible speeds and increasing the train speed.

In order to carry out continuous analysis in train conditions of the value of the resulting longitudinal compressive forces and to prevent large compressive forces, it is necessary to equip locomotives with a system for monitoring and recording longitudinal forces arising on the automatic coupler of the wagons.

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Conflicts of interest

The authors declare no conflicts of interest.

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