Coefficient of stability against lift by longitudinal forces of freight cars in trains

A O Shvets\textsuperscript{1,3,4}, O V Shatunov\textsuperscript{2,3}, S S Dovhaniuk\textsuperscript{2,3}, L A Muradian\textsuperscript{2,3}, A L Pularyia\textsuperscript{2,3} and V V Kalashnik\textsuperscript{2,3}

\textsuperscript{1}Department «Theoretical and Structural Mechanics»
\textsuperscript{2}Department «Cars and Car Facilities»
\textsuperscript{3}Dnipro National University of Railway Transport named after Academician V. Lazaryan, Lazaryan St., 2, Dnipro, Ukraine, 49010

\textsuperscript{4}angela\_shvets@ua.fm

Abstract. The paper is devoted to a theoretical study of the car movement stability when exposed to longitudinal forces of a quasistatic nature. The authors obtained the expressions for calculating the resistance coefficient of the car lift by the longitudinal compressive force acting on the car as part of a freight train. The influence of some factor’s combination on the longitudinal forces, at which the car movement stability is still maintained, was analyzed using the analytical dependencies to assess the longitudinal loading of cars in trains. The study was carried out by analytical method for assessing the freight car stability when moving at different speeds along curved track sections. The calculations were made in a curve of small radius taking into account the inertia forces from the unbalanced acceleration. In a theoretical study, the influence of quasistatic longitudinal compressive forces depending on changes in speed and force value, as well as the influence of friction in the wheel flange and rail contact and the eccentricity of fastening the automatic coupler shank on the stability were considered. When applying the results obtained, the stability of freight rolling stock can be increased, which in turn will remove some existing restrictions on the permissible speeds and increase the technical speed of train movement. Using the described methodology for determining the lift resistance coefficient will make it possible to justify the cause of derailment, as well as to develop and put into practice technical measures to prevent the lift of carriages, thrusts and shears of the track.

1. Introduction
Ensuring the movement stability of cars in long freight trains with increased carrying capacity remains an urgent problem for a long period of time. The values of the permissible longitudinal forces are included in the regulatory and technical documents, according to which all newly designed and constructed cars must satisfy the conditions of lift resistance and rolling off the track \cite{1-3}. These documents also serve as guidance when organizing the movement of long freight trains with increased carrying capacity. However, during operation, there is a need to adjust regulatory documents, especially when increasing movement speed and cancelling too stiff restrictions on the conditions of trains driving \cite{4-6}.

The longitudinal quasistatic compressive force in a train is the main operational parameter. Therefore, one of the priority tasks of improving the technology of trains driving is to reduce this power factor in the process of operational work. In addition, the transfer of longitudinal forces of a
quasistatic nature to the elements and units of cars, as well as to the railway track structure, has not yet been sufficiently studied, especially in theoretical terms [7-9].

2. Methodology
The main criterion for assessing the dynamic qualities of the rolling stock is the criterion of the carriage derailment resistance, which is the maximum permissible ratio of lateral horizontal and vertical forces acting on the climbing wheel [10-13]. The choice of design schemes for cars and trains is determined by the set task and the criterion for assessing the movement stability of the car. The position of car is set by the rotation angle of the central axle and/or the value of the lateral swaying of the body relative the track axle [4]. In addition, the design scheme should take into account the possibility of placing a separate car on a rail track, depending on the compressive or tensile forces in a curve or a straight track section [12, 13]. The design scheme should also reflect the characteristics of the transmission of longitudinal forces to the elements of bogies in the vertical and horizontal (transverse) directions. Each case of forces application to the car in a vertical plane should be considered together with various cases of car placing in a horizontal plane and separately for tensile and compressive longitudinal forces in straight and curved track sections [4, 13].

In regenerative braking mode, the cars occupy the position of maximum skew within the gap in the rail track, and the rotation angles of the experimental and neighboring cars are of the same sign (Figure 1, Table 1). When driving along a curve in traction mode, cars occupy a predominantly chordal position. With an increase in the quasistatic compression of the train, the skews increase, causing an increase in the lateral and vertical effects of the wheels of car undercarriage on the track structure. At the same time, the wheels of neighboring bogies act on opposite rail tracks, that is, they tend to thrust the track or shift it in different directions [4, 7].

The main reasons of the additional group influence of the bogie wheels on the track caused by quasistatic stretching and compression of the train are as follows: eccentricities of fastening the automatic coupler shank in the vertical $\Delta$ and horizontal plane $e$; vertical skews of freight cars during the train compression. The automatic coupler shank of SA-3 (SA-3M) model transfers the compressive force to the base plate at a point which moves in a horizontal plane across the carriage during the skew-symmetrical skews of cars (according to the herringbone pattern). Reasonably practicable value for cars is $e = 20$ mm. Eccentricity $e$ is possible as a result of deviation from the design position of the automatic coupler shank during the carriage manufacture at the factory, and the value $\Delta$ is caused by the car loading, the vertical stiffness of the central spring suspension and the wheels wear. When an empty car is placed on a train with fully loaded cars, the value can reach 80 mm [7, 10].

![Figure 1. Scheme of forces acting on the car at skew-symmetrical skew.](image-url)
### Table 1. Parameters used in determining the resistance coefficient of car lift by longitudinal forces.

| Designation | Parameter                                                                                      | Dimension     |
|-------------|------------------------------------------------------------------------------------------------|---------------|
| $N$         | longitudinal quasistatic force in an automatic coupler under the action of compressive forces on the car | kN            |
| $G_0$       | car body weight                                                                               | kN            |
| $H_f$       | frame force acting on the wheel set                                                            | kN            |
| $G_{bog}$   | bogie weight                                                                                  | kN            |
| $C_h$       | horizontal stiffness of spring suspension of one bogie                                        | kN/m          |
| $C_v$       | vertical stiffness of spring suspension of one bogie                                          | kN/m          |
| $\varphi_1$, $\varphi_2$ | angles in the vertical plane due to the difference in levels of the automatic coupler axles in the connection of two cars | rad          |
| $\Delta_1$, $\Delta_2$ | difference in levels of the automatic coupler axles in front and behind the car | m            |
| $e$         | eccentricity of the automatic coupler shank as a result of deviation from the design position during the manufacture | m            |
| $2b$        | the distance between the middle of the wheel set axle necks                                   | m            |
| $2\delta_0$ | total lateral acceleration of the car body frame relative to the track axle in the guiding section along the center pin | m            |
| $2L_c$      | length over coupler pulling faces                                                             | m            |
| $2\ell$     | wheel base                                                                                    | m            |
| $2L$        | distance between coupler followers                                                            | m            |
| $R$         | curve radius                                                                                   | m            |
| $a$         | automatic coupler body length from pulling face to the shank end                               | m            |
| $2S$        | distance between wheel rolling circles                                                         | m            |
| $h_a$       | automatic coupler axle height above the rail heads level                                       | m            |
| $h_{hs}$    | height above the rail heads plane to the upper plane of the central spring group               | m            |
| $r$         | radius of a medium-worn wheel                                                                  | m            |
| $h_{c}$     | the height of the car gravity center above the rail heads level                                 | m            |
| $h_{car}$   | height above the rail heads plane to the gravity center of the side surface of the car body   | m            |
| $a_{un}$    | unbalanced acceleration                                                                        | m/s²         |
| $g$         | acceleration of gravity                                                                        | m/s²         |
| $\mu$       | wheel-rail friction coefficient                                                                | m/s²         |
| $\mu_z$     | wheel-rail friction coefficient on non-climbing wheel                                          | m/s²         |
| $\beta$     | inclination angle formed by the conical surface of the wheel flange to the horizontal axle    |               |
Currently, in Ukraine there are significant differences in determining the stability of freight cars under the action of quasistatic compressive longitudinal forces in the regulatory documentation DSTU 7598:2014 “Freight cars. General requirements for the calculations and design of new and modernized 1520 mm gauge cars (unpowered cars)” and the procedure for calculating wheel unloading during forensic inquiries [14]. The document DSTU 7598:2014 regulates calculations when cars are placed according to the maximum skew-symmetric skew scheme (“herringbone” position). At the same time, during the technical examinations on the rolling stock derailments, the necessary calculations are performed for the chordal arrangement of cars on a railway track [14]. The methodology presented in this study will make it possible to carry out necessary calculations for unloading the wheels when the cars are skewed according to the scheme of the maximum skew-symmetric skew (“herringbone” position).

The basis of the methodology for determining the lift resistance coefficient by longitudinal forces is the research presented in the works [9, 15, 16]. The carriage movement in the curve is considered at different angles on both sides of the car: the tilt angles of the bodies to the track plane $\varphi_1$ and $\varphi_2$; automatic coupler rotation angles relative to the track axle $\psi_1$ and $\psi_2$; the car body rotation within the gap in the track by an angle $\psi$ is taken into account. The car (Figure 1) is subject to compressive longitudinal forces, and the carriage is placed on the rail track according to the maximum skew-symmetrical skew scheme. The longitudinal forces $N_1$ and $N_2$, acting in automatic couplings in front and behind the car, are assumed to be the same.

The forces acting on the car are projected on the track plane, as well as on the planes perpendicular to it – longitudinal and transverse relative to the track axle. The designations given in Figure 1, as well as the parameters used to determine the resistance coefficient of car lift by longitudinal forces, are summarized in Table 1. The efforts depicted in Figure 1 are determined according to the methodology given in the works [9, 16].

Angles in the vertical plane due to the difference in levels of the automatic coupler axles in the connection of two cars:

$$\varphi_1 = \frac{\Delta_1}{2a}, \quad \varphi_2 = \frac{\Delta_2}{2a}. \quad (1)$$

Consider the definition of angles $\psi$, $\psi_1$, and $\psi_2$ in the horizontal plane during compression of the car in the case of installation in a rut with the deviation of the body center plate across the path (in different directions from the longitudinal axis of the car) by the value $\delta_0$ (Figure 2).

![Figure 2](image-url)  

**Figure 2.** The scheme of the car to determine the components of the longitudinal forces in the horizontal plane.

In Figure 2, the dashed line shows the interaction forces and the position of the central axis with the chordal arrangement of the car under study under the action of compressive forces.
When the car axis is skewed (Figure 1) due to a lateral deflection from the track axis of the center plate section of the car frame due to the wheel set acceleration in the rail track, bearings are skewed along the axle necks, axle boxes – relative to the bogie frame ($BB' = A'A'' = \delta_0$), the car axis will additionally deflect due to spring deformation under the action of transverse components of longitudinal force ($B'B'' = A'A$). Let us find the rotation angles of the car body in a curve in the horizontal plane:

$$\psi = \frac{\delta}{\ell},$$  \hspace{1cm} (2)

$$\psi_1 = \frac{\delta}{\ell} \left( 1 + \frac{L}{a} + \frac{L_c}{R} \right),$$ \hspace{1cm} (3)

$$\psi_2 = \frac{\delta}{\ell} \left( 1 + \frac{L}{a} - \frac{L_c}{R} \right).$$ \hspace{1cm} (4)

Since the angles $\psi$, $\psi_1$ and $\psi_2$ depend on the argument $\delta$, which in turn depends on $H_i$, the calculation by formulas (2)-(4) is possible by the method of successive approximations. With sufficient accuracy for calculation, it is possible to limit ourselves to the first approximation and formulas (2), (3), (4), taking into account the elastic transverse deformation of the spring sets of the bogies, will take the following form:

$$\psi = \frac{\delta_0 + e}{\ell} \left[ 1 + \frac{N \cdot L}{2 \ell^2 \cdot C_h} \left( 2 + \frac{L}{a} \right) \right],$$ \hspace{1cm} (5)

$$\psi_1 = \frac{\delta_0 + e}{\ell} \left[ 1 + \frac{N \cdot L}{2 \ell^2 \cdot C_h} \left( 2 + \frac{L}{a} \right) \right] \left( 1 + \frac{L}{a} \right) + \frac{L_c}{R},$$ \hspace{1cm} (6)

$$\psi_2 = \frac{\delta_0 + e}{\ell} \left[ 1 + \frac{N \cdot L}{2 \ell^2 \cdot C_h} \left( 2 + \frac{L}{a} \right) \right] \left( 1 + \frac{L}{a} \right) - \frac{L_c}{R}.\hspace{1cm} (7)$$

From expressions (5), (6), (7) it is possible to obtain dependencies for determining the angles in the case of a “clear” skew of the car in a straight track section. If we neglect the elastic transverse deformation of the spring sets of the bogies and the eccentricity of the automatic coupler shank location, these formulas take the form obtained in the works [4, 15] and DSTU 7598:2014.

The vertical and horizontal forces $H_i$ acting on the wheel set are determined from the balance of the unsprung part of the bogie. As a result of solving a system of linear equations, we determine the vertical and lateral transverse reaction of the rail to the climbing wheel, as well as the lift resistance coefficient by longitudinal forces.

$$V = \frac{G_0}{8} + \frac{G_{bog}}{4} + \frac{N}{8} \left( (\psi_1 - \psi_2) \cdot \frac{h_m}{S} \right) \left[ \varphi_1 \cdot \left( \frac{L}{\ell} + 1 \right) + \varphi_2 \cdot \left( \frac{L}{\ell} - 1 \right) \right] \cdot \frac{b}{S} + (\psi_1 + \psi_2 + 2\psi) \cdot \frac{L \cdot h_m}{\ell \cdot S},$$ \hspace{1cm} (8)

$$L = \mu \cdot \left( \frac{G_0}{8} + \frac{G_{bog}}{4} \right) + \frac{N}{8} \left( (\psi_1 - \psi_2) \cdot \left( 2 - \mu \cdot \frac{h_m}{S} \right) - \mu \cdot \left( \varphi_1 \cdot \left( \frac{L}{\ell} + 1 \right) + \varphi_2 \cdot \left( \frac{L}{\ell} - 1 \right) \right) \cdot \frac{b}{S} + 
+ (\psi_1 + \psi_2 + 2\psi) \cdot \frac{L}{\ell} \cdot \left( 2 - \mu \cdot \frac{h_m}{S} \right) \right),$$ \hspace{1cm} (9)
If it is necessary to take into account inertia forces in the curve, the vertical (8) and the lateral (9) reactions of the rail on the climbing wheel are added to the inertia forces caused by unbalanced acceleration and wind pressure on the side surface of the car body.

The dependency for determining the resistance coefficient of car lift by longitudinal forces, taking into account the inertia forces and wind pressure on the car body’s side surface in the curve for the climbing wheel set of the front carriage, has the following form:

\[
C_{u} = \frac{tg \beta - \mu}{1 + \mu \cdot tg \beta} \cdot \frac{P_{\text{car}} + N \cdot \left[ \left( \psi_{1} - \psi_{2} \right) \cdot \frac{h_{\text{hs}}}{S} - \frac{\varphi_{e} \cdot b}{S} + \left( \psi_{1} + \psi_{2} + 2\psi \right) \cdot \frac{L \cdot h_{\text{hs}}}{\ell \cdot S} \right]}{\mu \cdot P_{\text{car}} + N \cdot \left[ \left( \psi_{1} - \psi_{2} \right) \cdot \left( 2 - \mu \cdot \frac{h_{\text{hs}}}{S} \right) - \mu \cdot \varphi_{e} \cdot \frac{b}{S} + \left( \psi_{1} + \psi_{2} + 2\psi \right) \cdot \frac{L}{\ell} \cdot \left( 2 - \mu \cdot \frac{h_{\text{hs}}}{S} \right) \right]}. \tag{10}
\]

Where \( P_{\text{car}} \) – is the static pressure of the car, taking into account unloading from longitudinal force, kN.

In dependency (11), it is recalculated for the climbing wheel set of the front carriage;

\[
P_{\text{car}}^{\text{st}} = P_{\text{car}} - N \cdot \varphi_{e} \cdot \frac{b}{S}, \tag{12}
\]

\( P_{\text{car}} = G_{o} + 2G_{\text{bog}} \) – car weight, kN;

\( \varphi_{e} \) – the rotation angle of the car body in a vertical plane, caused by the presence of a difference in the levels of automatic coupler axles in front and behind the car, rad;

\[
\varphi_{e} = \varphi_{1} \cdot \left( \frac{L}{\ell} + 1 \right) + \varphi_{2} \cdot \left( \frac{L}{\ell} - 1 \right). \tag{13}
\]

\( \psi_{a} \) – the angle formed by the longitudinal axle of the automatic coupler body and the axle of the central sill of the car frame in a horizontal plane, rad;

\[
\psi_{a} = \left( \delta_{0} + e \right) \cdot \frac{L}{\ell^{2}} \cdot \left( 2 + \frac{L}{a} \right). \tag{14}
\]

\( \psi_{\text{cur}} \) – car rotation angle, depending on the location in the curve, rad;

\[
\psi_{\text{cur}} = \frac{2L}{R}. \tag{15}
\]

\( h_{\text{hs}} = r + r_{n} \approx r \) – height above the level of the rail heads plane to the upper plane of the central spring set, m. Often this parameter in a number of studies is taken equal to the radius of the average worn wheel.

\( P_{\text{in}} \) – inertia force from unbalanced acceleration, kN;
\[ P_{\text{in}} = \left( G_0 + 2 \cdot G_{\text{bog}} \right) \cdot \frac{a_{\text{in}}}{g} = P_{\text{car}} \cdot \frac{a_{\text{in}}}{g}. \]  

Dependency (15) is calculated and cannot be used to determine stability in the experiments. The calculation is based on a static calculation scheme. However, the car is a complex mechanical system and during movement, the interaction between its individual parts and between the car and the track structure is dynamic [9].

In the work [17], a dependency is given for assessing the movement safety; on the basis of the experiment, it takes into account the wheel set speed influence on the friction coefficient value in the contact between the wheel flange and the rail. The dependency for determining the lift resistance coefficient by longitudinal forces, taking into account the inertia forces in the curve and the component of the wind loading action on the side surface of the car body, will take the following form:

\[
C_{\text{lr}} = \frac{\tan \beta - \mu_0 \cdot (1 - 0.002v)}{1 + \mu_0 \cdot (1 - 0.002v) \cdot \tan \beta}. 
\]

\[
P_{\text{lay}} + \frac{N^2}{C_h} \cdot \frac{\psi^2 \cdot h_{\text{in}}}{(\delta_0 + e) \cdot S} + N \cdot \left[ 2\psi_a \cdot \frac{h_{\text{in}}}{S} + \psi_{\text{car}} \cdot \frac{h_{\text{car}}}{S} \right] + P_{\text{in}} \cdot \frac{h_k}{S} + F_w \cdot \frac{h_{\text{car}}}{S}
\]

\[
\mu_2 \cdot P_{\text{lay}} + \frac{N^2}{C_r} \cdot \frac{\psi^2 \cdot h_{\text{in}}}{(\delta_0 + e) \cdot S} + N \cdot \left[ 2\psi_a \cdot \left( 2 - \mu_2 \cdot \frac{h_{\text{in}}}{S} \right) + \psi_{\text{car}} \cdot \left( 2 - \mu_2 \cdot \frac{h_{\text{car}}}{S} \right) \right] +
\]

\[
+ P_{\text{in}} \cdot \left( 2 - \mu_2 \cdot \frac{h_k}{S} + F_w \cdot \left( 2 - \mu_2 \cdot \frac{h_{\text{car}}}{S} \right) \right)
\]

Where \( v \) – speed of movement in km/h;

\( \mu_0 \) – is the friction coefficient at the adhesion limit at \( v = 0 \). For railway rolling stock, in the calculations it is recommended to take this parameter equal to \( \mu_0 = 0.3 \div 0.33 \).

3. Calculation results

To assess the influence of the longitudinal forces on the lift resistance coefficient, Figures 3, 4 show the results of movement calculations of empty and loaded open car of 12-532 model in a curve with a radius of 250 m and elevation of 150 mm with a transverse gap of the car body frame relative to the track axle in directing section of 50 mm. The calculations were performed without taking into account the friction forces and taking into account the friction in the contact of the wheel flange and the rail at a speed from 10 to 120 km/h.

It was assumed that the longitudinal forces acting on the cars are static or quasi-static external forces that slowly change in time. During the calculations, the difference in height between the longitudinal axles of automatic coupler of a freight train was taken into account. It is permitted no more than \( \Delta_1 = 80 \text{ mm} \) in front of a group of empty cars standing behind the loaded cars in a train. Behind the car under study, the difference in the levels of the automatic coupler axles is taken to be \( \Delta_2 = 0 \text{ mm} \), in addition, the presence an eccentricity of fastening the automatic coupler shank \( e = 20 \text{ mm} \) was taken into account. During calculation of the influence of wind loading on the side surface of the body was taken into account. This force is applied to the geometric center of the side surface, directed outward of the curve and taken equal to 10 kN.

The resistance coefficient of an empty open car lift in Figure 3 is given as follows: (a), (c) excluding the friction forces in the contact of the wheel flange and the rail and (b) taking into account these factors; (a), (b) with and (c), (d) without the eccentricity of fastening the automatic coupler shank. Analysis of the obtained values of the resistance coefficient of an empty open car lift (Figure 3) shows that dependency (17) gives stiffer results as compared to dependency (11).
Figure 3. Resistance coefficient of an empty open car lift.

Thus, accounting the different friction in the contact between the wheel and the rail with an eccentricity of fastening the automatic coupler shank according to (17) shows that there is no lift resistance even at 400 kN, and (11) with a compressive force of 480 kN. The eccentricity of fastening the automatic coupler shank has a significant influence on the value of longitudinal force that can lift an empty freight car. The absence of eccentricity leads to an increase in the longitudinal force by 100 kN when calculating for both dependencies. Excluding wind power, the longitudinal force decreases by an average of 30-50 kN. If the wind pressure is directed inside the curve, then the force of the wind load will contribute to the additional unloading of the climbing wheel, which will significantly affect the conditions of car lift.
The resistance coefficient of a loaded open car lift in Figure 3 is given in the same way: excluding the friction forces in the contact of the wheel flange and the rail and (b) taking into account these factors; (a), (b) with and (c), (d) without the eccentricity of fastening the automatic coupler shank. In the loaded mode (Figure 4), the open car under study maintains the established resistance coefficient for freight cars under the action of a quasistatic compressive force of 1 MN permissible by regulatory documentation.

4. Conclusions
As a result of theoretical studies, the dependencies of the resistance coefficient of car lift by longitudinal forces are obtained taking into account the eccentricity of fastening the automatic coupler shank and the friction value in the contact of the wheel and rail. The influence of longitudinal compressive forces on the stability of freight rolling stock when moving in a curve of small radius with speeds up to a design value of 120 km/h is investigated.

Based on the theoretical studies of the stability of the freight carriage using the example of open car of the model 12-532, it is possible to draw the following conclusions:

- the presence of eccentricity of fastening the automatic coupler shank for empty cars significantly affects the value of the car lift resistance coefficient;
- the influence of freight cars on the track is possible by reducing the total transverse gap of the car body frame $\delta$ by reducing the eccentricity of fastening the automatic coupler shank $\epsilon$ and increasing the horizontal stiffness of the spring suspension $C_h$, as well as by reducing the gaps in axle boxes, center plate arrangements and other units of rolling stock, affecting lateral movements (skews) of the body;
- in the loaded state, the open car under study maintains the set lift resistance coefficient for freight cars under the action of quasistatic compressive force of 1 MN permissible by the regulatory documentation.

The investigated dependence for determining the lift resistance coefficient by longitudinal forces was compiled for a freight rolling stock on the 18-100 bogie models and similar ones. For the running gears with structural differences from the 18-100 three-piece bogie, dependencies (11) and (17) can have significant differences.

Compared to the existing methods for determining the stability of freight rolling stock under the action of compressive longitudinal forces, the proposed method allows assessing the influence of longitudinal force and movement speed of a freight car on the value of the stability coefficient. It also allows taking into account the eccentricity of fastening the automatic coupler shank, the pressure of wind loading on the side surface of the body, as well as calculating the unloading or additional loading of each wheel separately during the technical examination.
Using the described methodology for determining the lift resistance coefficient will allow justifying the cause of derailment, as well as to develop and put into practice the technical measures to prevent car lifts, thrusts and track shears. When applying the results obtained, the stability of freight rolling stock can be increased, which in turn will remove some existing restrictions on permissible speeds and increase the technical speed of train movement.

References
[1] Pshinko O, Ursulyak L, Kostrytsia S, Fedorov Ye and Shvets A 2019 The influence of the «train-track» system parameters on the maximum longitudinal forces' level Transport Problems 4 14 pp 161–172
[2] Fomin O V 2015 Increase of the freight wagons ideality degree and prognostication of their evolution stages Scientific Bulletin of National Mining University 2 pp 68–76
[3] Kelykh M and Fomin O 2014 Perspective directions of planning carrying systems of gondolas Scientific and technical journal «Metallurgical and Mining Industry» 6 pp 64–67
[4] Cherkashin Yu M and Kostin G V 1992 Determination of permissible longitudinal forces in a train under the condition of ensuring the stability of the movement of cars Study of strength, stability, impact on the track and maintenance of wagons in trains of increased mass and length. Collection of works VNIIZhT (Moscow: Publisher Transport) pp 4–31
[5] Shimanovsky A O, Sakharau P A and Kovalenko A V 2018 Modeling of train longitudinal dynamics in MSC.ADAMS software Aktualnye voprosy mashinovedeniya 7 pp 75–78
[6] Shimanovsky A O and Sakharau P A 2019 Effect of gap clearances in automatic coupling devices on longitudinal forces in intercar connections of homogeneous train. Mekhanika mashin, mekhanizmov i materialov 2 47 pp 42–50
[7] Lysuky V S 2002 The causes and mechanisms of the vanishing wheel from the rail. The problem of wear of wheels and rails (Moscow: Publisher Transport) p 215
[8] Zheleznov K, Akulov A, Zabolotnyi O, Ursulyak L, Chabanuk Ye, Shvets A, Kuznetsov V and Radkevych A 2019 The revised method for calculating of the optimal train control mode Archives of Transport 3 51 pp 21–34
[9] Shvets A A, Zheleznov K I, Akulov A S, Zabolotnyi A N and Chabanyuk Ye V 2016 Determination the permissible forces in assessing the lift resistant factor of freight cars in trains Science and Transport Progress 1 61 pp 189–192
[10] Fomin O, Shvets A, Hauser V and Prokopenko P 2019 Transversal displacement of freight wagons bogies AIP Conference Proceedings 2198 020002
[11] Shvets A A, Zheleznov K I, Akulov A S, Zabolotnyi A N and Chabaniuk Ye V 2015 Some aspects of the definition of empty cars stability from squeezing their longitudinal forces in the freight train Science and Transport Progress 4 58 pp 175–189
[12] Shvets A A, Zheleznov K I, Akulov A S, Zabolotnyi A N and Chabaniuk Ye V 2015 Determination of the issue concerning the lift resistance factor of lightweight car Science and Transport Progress 6 60 pp 134–148
[13] Danovich V D and Malysheva A A 1998 Mathematical model of spatial oscillations of the coupling of five cars moving along a rectilinear section of the track Transport. Stress loading and durability of a rolling stock (Dnepropetrovsk: Dniprop University of Railway Transport named after Academician V. Lazaryan) pp 62–69
[14] Sokol E N 2002 Derailment and collision of rolling stock (Kiev: Transport of Ukraine) p 364
[15] Vershynsky S V 1970 Dynamics, durability and the stability to heavy wagons and high-speed trains Collection of works VNIIZhT (Moscow: Publisher Transport) 425 p 208
[16] Shvets A O 2020 Stability of freight cars under the action of compressive longitudinal forces Science and Transport Progress (Preprint gr-qc/10.15802/stp2020/199485)
[17] Andrievsky S M and Krylov V A 1969 Wheel derailment Research in the dynamics and strength of locomotives (Moscow: Publisher Transport) 393 pp 20–41