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DEFINING PATTERNS IN THE DYNAMIC LOAD AND STRENGTH OF THE BEARING STRUCTURE OF A COVERED FREIGHT CAR WITH A FILLER IN THE GIRDER BEAM

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This paper reports a study into determining the dynamic load and strength of the bearing structure of a covered freight car under operational modes. A feature of the freight car's bearing structure is that the girder beam has a closed cross-section. To reduce the dynamic load of the frame, the girder beam is filled with a material with viscoelastic properties. Such a solution could contribute to the transformation of the kinetic energy of impact (due to jerk, stretching, compression) into work of viscoelastic friction forces, and, consequently, to reducing the load on the bearing structure.

To substantiate the proposed improvement, the dynamic load on the bearing structure of a covered freight car was mathematically modeled. The calculation was performed for the case of joint impacts at shunting. The study was carried out in a flat coordinate system. It was established that the maximum accelerations acting on the bearing structure of a covered freight car were about 37 m/s^2. The calculated acceleration value is 3.2 % lower than that obtained for the bearing structure of a covered freight car without filler.

The results of calculating the strength of the load-bearing structure of a covered freight car are given. In this case, a finite-element method was applied. The maximum equivalent stresses occur in the zones of interaction between the girder beam and the pivot beams, and amount to 319.5 MPa, which is 8 % lower than permissible. The calculation was also performed regarding other operational modes of loading the freight car's bearing structure.

The model of the dynamic load on the bearing structure of a covered freight car was verified according to the F-criterion.

The research reported here could contribute to designing innovative rolling stock structures, thereby improving the efficiency of their operation

Keywords: transport mechanics, covered freight car, load-bearing structure, dynamic load, innovative freight car

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1. Introduction

Ensuring the competitiveness of the railroad industry necessitates designing and implementing modern rolling stock structures. When constructing such rolling stock, it is important to take into consideration technical solutions that could contribute to the improvement of its technical and economic [1, 2], operational [3, 4], as well as environmental indicators [5, 6].

The most common type of freight car used to transport cargoes that require protection from atmospheric influences is a covered freight car. Under operational conditions, its bearing structure is exposed to significant loads that cause its damage.

The most loaded node of the bearing structure of a covered freight car is the frame. The main longitudinal load, which acts on the frame under operational modes, is perceived by the girder beam. In most covered freight cars, it is composed of two Z-shaped profiles.

The cyclical effect of longitudinal loads on the girder beam may give rise to cracks, deformities, and other damage to it. This circumstance predetermines unscheduled repairs of the freight car, additional costs for its maintenance during operation, or even its decommissioning. In addition, such damage can affect the environmental friendliness and safety of cargo transportation by rail [7, 8]. Therefore, it is a relevant task to improve the load-bearing structure of the covered freight car to ensure its strength under the most adverse operating load modes.

2. Literature review and problem statement

Features in determining the strength of the bearing structure of a covered freight car using experimental research methods are described in work [9]. In this case, the method of electrical tensometry was applied. The calculation results established that the strength of the bearing structure of the covered freight car is ensured.

Work [10] reports the results of bench tests for fatigue resistance of the layout nodes of covered freight cars. The layouts were loaded at a hydraulic-pulse bench. The cited work also highlights the feasibility of the reported results.

It is important to note that the cited works do not propose measures to reduce the loading on the bearing structures of covered freight cars under operational modes.

The bearing structure of the freight car frame was improved in paper [11] to ensure strength under operational modes. The authors described features in calculating the strength of the improved load-bearing structure of the freight car with the help of computer simulation. However, the cited paper does not indicate any prospects for improving the technical and economic indicators of that freight car.

The results of calculating the strength of the bearing structure of a covered freight car are reported in [12]. The calculation was implemented by a finite-element method. The dislocation fields and numerical values of the maximum equivalent stresses in the load-bearing structure of the covered freight car were determined. However, the authors did not propose measures to reduce the load on the freight car's bearing structure during operation.

Features in the optimization of load-bearing structures of freight cars are described in [13]. The optimization aimed to reduce the material consumption for the freight car's bearing structure by using aluminum panels in the body. The use of the proposed panels also helps reduce the dynamic load on a freight car due to the presence of elastic connections in them. However, the issue of reducing the dynamic load on the freight car's frame was disregarded in the cited work.

The authors of [14, 15] make improvements to the load-bearing structures of freight cars to reduce their dynamic load under operational modes. The set goal is achieved by using a material with energy-absorbing properties for the load-bearing structures of freight cars containing round cross-section pipes. It is important to say that such load-bearing structures of freight cars are characterized by complexity from a manufacturing point of view.

The substantiation of choosing a new profile for the girder beam of the freight car is given in work [16]. A series of design solutions for the frames of freight cars of various types are presented. The results of calculating the strength of the load-bearing structures of freight cars are reported, taking into consideration the proposed solutions. However, the cited work does not propose measures to reduce the load on the freight car's bearing structure under operational modes.

Paper [17] reports the results of optimization of the freight car frame according to the criterion of minimum material consumption. The proposed design of the freight car frame was estimated for strength by a finite-element method. The authors applied European standards for the load-bearing structure of the freight car in operation. At the same time, the improved frame structure does not reduce the dynamic load on the freight car.

The strength of the freight car frame at a longitudinal impact was determined in [18]. The possible options for strengthening the freight car frame, as well as elements that could contribute to the absorption of impact energy, were considered. It is important to say that the study was carried out regarding the frame of a passenger car, which experiences a lower load in operation than the freight one. The authors did not pay attention to the improvement of a freight car frame.

The above review of literary sources [9–18] allows us to conclude that the issue related to reducing the load on the bearing structure of a covered freight car under operational load modes requires further research and development.

3. The aim and objectives of the study

The aim of this work is to identify patterns regarding the dynamic loading and strength of the bearing structure of a covered freight car with filler in the girder beam. This could make it possible to improve the strength of the bearing structure of a covered freight car under operational load modes, reduce maintenance costs, as well as contribute to working out the designs for innovative freight car structures.

To accomplish the aim, the following tasks have been set:

– to propose measures to reduce the load on the bearing structure of a covered freight car under operational modes;

– to carry out mathematical modeling of the dynamic load on the bearing structure of a covered freight car with filler in the girder beam;

– to calculate the strength of the load-bearing structure of a covered freight car;

– to verify the model of the dynamic load on the covered freight car.

4. The study materials and methods

To determine the optimal parameters for the profile to make the girder beam, the optimization method by strength reserve was used. Minimization of the material consumption for the frame was chosen as an optimization criterion.

In order to determine the dynamic loads acting on the improved frame design, the mathematical model given in [19] was used. However, within the framework of our study, the model was adapted to determine the dynamic load on the bearing structure of a covered freight car. The model also takes into consideration the friction force that occurs between the heels and sub-heels, due to the effect of the longitudinal force on the rear stop of the autocoupling device [20–22]. Differential equations were solved using the Mathcad software suite (Boston, USA) [23–26] by applying the Runge-Kutta method. Starting conditions were accepted as zero [27–29].

When calculating the strength of the load-bearing structure of a covered freight car, a finite element method was used, implemented in the SolidWorks Simulation software package (CosmosWorks), (France) [30–32]. In this case, the finite-element model of the bearing structure of a covered freight car was formed from spatial isoparametric tetrahedra [33–35], whose optimal number was determined by the graphic-analytical method [36–38]. The number of elements in the grid was 608,959, nodes – 208,409. The maximum size of the grid element was 100 mm, minimum – 20 mm, the maximum aspect ratio of the elements was 1,224, the percentage of elements with an aspect ratio less than three was 10.1; more than ten – 50. The minimum number of elements in a circle was 22, the ratio of increasing the size of the element was 1.9. The material of the structure was steel of grade 09G2C [20, 39].

To verify the built model of the dynamic load on the bearing structure of a covered freight car, an F-criterion [40–42] was used. As a variation parameter, the force of an impact at an autocoupling was taken into consideration. The initial parameter is acceleration in the bearing structure of a covered freight car.

5. Results of studying the dynamic load and strength of the bearing structure of a covered freight car

5. 1. Measures to reduce the load on the bearing structure of a covered freight car during operation

In order to reduce the load on the girder beam of a covered freight car as the main bearing element of the frame, it is proposed to fabricate it from a special profile that forms a closed cross-section (Fig. 1). Cross-sectional parameters were selected based on the calculation of the typical frame structure and from determining its strength reserves (Table 1). Table 1 lists the following introduced designations: σ_{eq} is the maximum equivalent stresses in the girder beam; W_i – the momentum of cross-section resistance relative to the corresponding axis; $[W_i]$ is the permissible cross-section resistance momentum relative to the corresponding axis.

Table 1

Cross-sectional parameters for a typical girder beam in a covered freight car

Mass 1 m, kg	Length, m	σ_{eq}, MPa	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3
125.67	13.87	330.0	202.13	3,634.99	192.5	3,461.9

It is assumed that the inner cavity of the girder beam is filled with filler with viscoelastic properties. This would contribute to transforming the kinetic energy of an impact (jerk, stretching, compression) into work of viscoelastic friction forces. This solution is proposed at the concept level.

It is important to note that the girder beam can be also represented by other types of profiles that form its closed cross-section. Possible options for fabricating a girder beam are shown in Fig. 2.

The bearing structure of a covered freight car with a closed-profile girder beam is shown in Fig. 3. The covered freight car of model 11-217 was chosen as a prototype.

It was established that using a special profile for fabricating a girder beam ensures a decrease in the frame weight by almost 4 % compared to the standard structure.

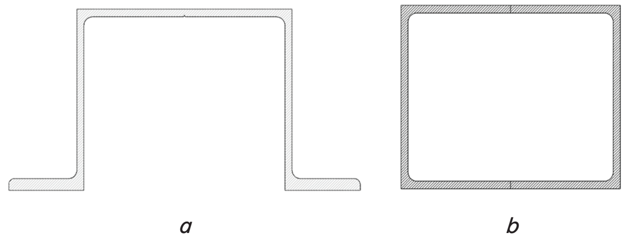

Fig. 1. Cross-section of the girder beam of a covered freight car: *a* – standard structure; *b* – improved design

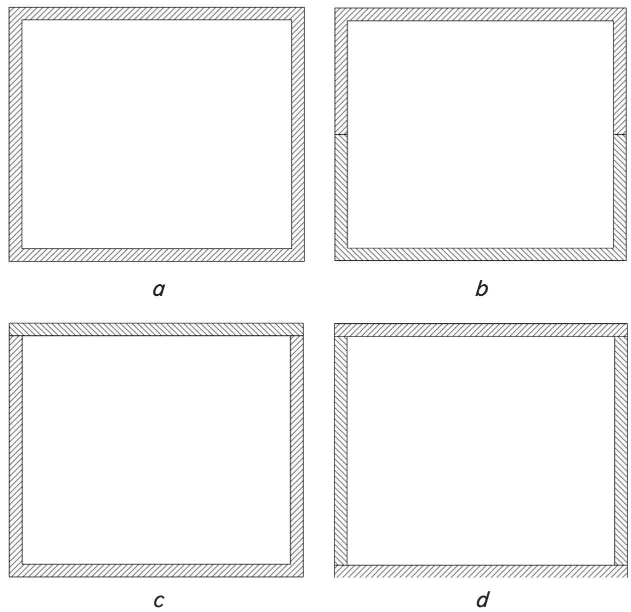

Fig. 2. Fabrication variants of the girder beam in a freight car: *a* – rectangular cross-section pipe; *b* – trough profiles; *c* – U-shaped profile closed by a horizontal sheet; *d* – sheet structure

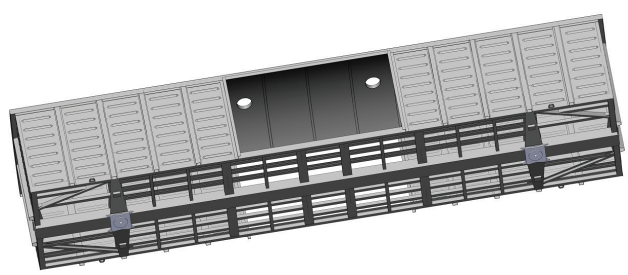

Fig. 3. The load-bearing structure of a covered freight car

5. 2. Mathematical modeling of dynamic load on the bearing structure of a covered freight car with a filler in the girder beam

In order to substantiate the proposed technical solutions, we have mathematically modeled the dynamic load on the bearing structure of a covered freight car during joint impacts at shunting. The estimation scheme of the covered freight car is shown in Fig. 4. The study was carried out in a flat coordinate system, the XZ plane.

The differential equations of the movement of a covered freight car are as follows:

$$M_{gm} \cdot \ddot{x} + (M_c \cdot h) \cdot \ddot{\varphi} = P_x, \tag{1}$$

$$I_c \cdot \ddot{\varphi} + (M_c \cdot h) \cdot \ddot{x} - g \cdot \varphi \cdot (M_c \cdot h) = P_y, \tag{2}$$

$$M_c \cdot \ddot{z} = k_1 \cdot \Delta_1 + k_2 \cdot \Delta_2 - F_{fr}\left(\mathrm{sign}\dot{\Delta}_1 - \mathrm{sign}\dot{\Delta}_2\right), \tag{3}$$

in this case,

$$\Delta_1 = z - l \cdot \varphi; \quad \Delta_2 = z + l \cdot \varphi, \tag{4}$$

$$P_x = P_l - 2P_{fr} - \beta \cdot \dot{x} - c \cdot x, \tag{5}$$

$$P_y = l \cdot F_{fr}\left(\mathrm{sign}\dot{\Delta}_1 - \mathrm{sign}\dot{\Delta}_2\right) + l\left(k_1 \cdot \Delta_1 - k_2 \cdot \Delta_2\right), \tag{6}$$

where M_{gm} is the gross mass of a covered freight car; M_c is the mass of the bearing structure of a covered freight car; I_c is the moment of inertia of a covered freight car; P_l is the longitudinal force on the rear stop of the autocoupling; P_{fr} is the friction forces that arise between the heels of the frame and the sub-heels of bogies; c is the stiffness of the material, which fills the girder beam of a covered freight car; β is the coefficient of viscous resistance of the material, which fills the girder beam of a covered freight car; l is the half-base of a covered freight car; F_{fr} is the value of dry friction force in the spring set (bogie model 18-100); k_1, k_2 is the rigidity of springs in the suspension of bogies in a covered freight car; x, φ, z are the coordinates corresponding, respectively, to the longitudinal, angular around the transverse axis, and the vertical movement of a covered freight car.

The differential equations of displacements of a covered freight car were solved in the following form

$$Q(t,y) = \begin{bmatrix} y_2 \\ y_4 \\ y_6 \\ \dfrac{P_x - (M_c \cdot h) \cdot \dot{y}_4}{M_{gm}} \\ \dfrac{P_y - (M_c \cdot h) \cdot \dot{y}_2 + g \cdot y_3 \cdot (M_c \cdot h)}{I_B} \\ \dfrac{k_1 \cdot \Delta_1 + k_2 \cdot \Delta_2 - F_{fr}\left(\mathrm{sign}\dot{\Delta}_1 - \mathrm{sign}\dot{\Delta}_2\right)}{M_c} \end{bmatrix}, \tag{7}$$

$$Z = rkfixed(Y0, tn, tk, n, F),$$

where $Y0$ is a vector that contains the initial conditions [43–46], tn, tk are the values that determine the initial and final integration variable, n' is the fixed number of steps, Q is a symbol vector.

In this case, $y_1 = q_1$, $y_3 = q_3$, $y_5 = q_5$, $y_2 = \dot{y}_1$, $y_4 = \dot{y}_3$, $y_6 = \dot{y}_5$.

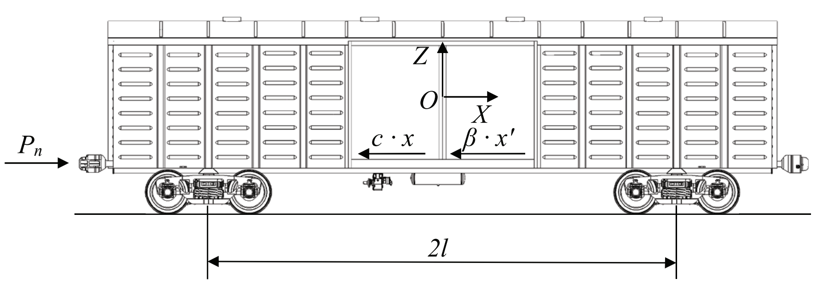

Fig. 4. Estimation scheme of the covered freight car

The generalized accelerations of the bearing structure of a covered freight car were calculated in the $ddq_{j,i}$ array:

$$ddq_{j,1} = \frac{P_x - (M_c \cdot h) \cdot \dot{y}_4}{M_{gm}}, \tag{8}$$

$$ddq_{j,2} = \frac{P_y - (M_c \cdot h) \cdot \dot{y}_2 + g \cdot y_3 \cdot (M_c \cdot h)}{I_c}, \tag{9}$$

$$ddq_{j,3} = \frac{k_1 \cdot \Delta_1 + k_2 \cdot \Delta_2 - F_{fr}\left(\mathrm{sign}\dot{\Delta}_1 - \mathrm{sign}\dot{\Delta}_2\right)}{M_c}. \tag{10}$$

The calculation results are shown in Fig. 5. It was established that the maximum accelerations acting on the bearing structure of a covered freight car were about 37 m/s^2. The calculated acceleration value is 3.2 % lower than that obtained for the bearing structure of a covered freight car without filler. When performing calculations, the coefficient of viscous resistance of the material which fills the girder beam was taken equal to 118 kN·s/m, and the rigidity – 80 kN/m.

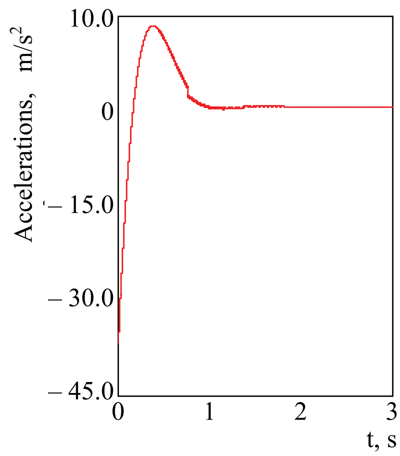

Fig. 5. Accelerations acting on the bearing structure of a covered freight car

The resulting value of acceleration was taken into consideration when estimating the strength of the load-bearing structure of a covered freight car.

5. 3. Calculating the strength of the bearing structure of a covered freight car

To calculate the load-bearing structure of a covered freight car, its finite-element model was constructe (Fig. 6). When building an estimation scheme of the bearing structure of a covered freight car, it was taken into consideration that it is exposed to the vertical load P_v^{st} (Fig. 7). In this case, the full carrying capacity of the freight car loaded with conditional cargo was used. The movement of cargo in the body was not taken into consideration. The rear stop of the autocoupling on one side of the freight car was exposed to a longitudinal load P_l, which was balanced by the forces of inertia of the mass of the freight car on the opposite side. In this case, P_l was adopted equal to 3.5 MN. The presence of the material with viscoelastic properties in the girder beam was modeled by installing the "spring-damper" links between the rear stops of the autocoupling devices using relevant options from the SolidWorks Simulation software package. The model was fixed in the areas where the bearing structure rested on the running parts [47–50]. The results of strength calculation are shown in Fig. 8, 9.

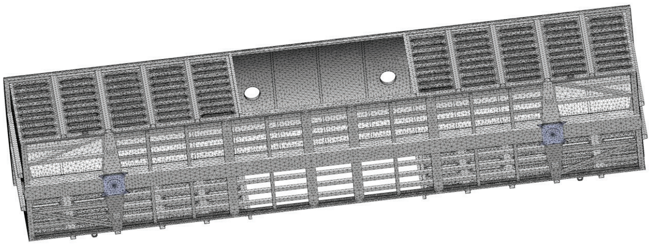

Fig. 6. The finite-element model of the bearing structure of a covered freight car

The maximum equivalent stresses occur in the zones of interaction between the girder beam and the pivot beams and are 319.5 MPa, which is 8 % lower than permissible ones. The maximum movements occur in the middle part of a covered freight car frame and are equal to 8.6 mm.

The calculation was performed for the main load modes of a covered freight car in operation. The calculation results are shown in Fig. 10.

Our calculations have shown that the strength of the bearing structure of a covered freight car under the main operating modes is ensured.

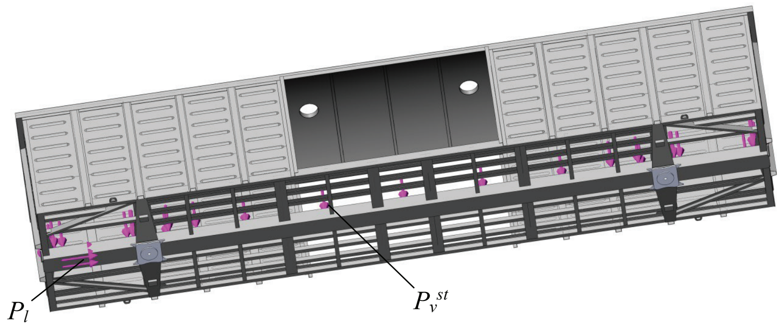

Fig. 7. Estimation scheme of the bearing structure of a covered freight car

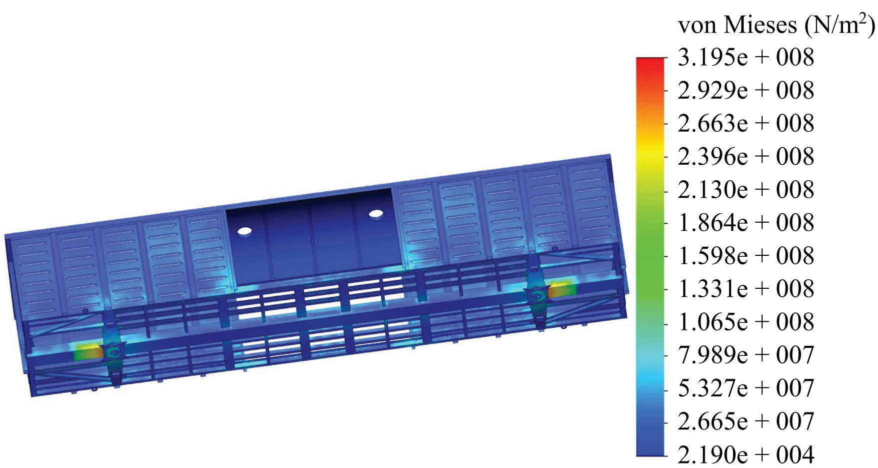

Fig. 8. The stressed state of the bearing structure of a covered freight car

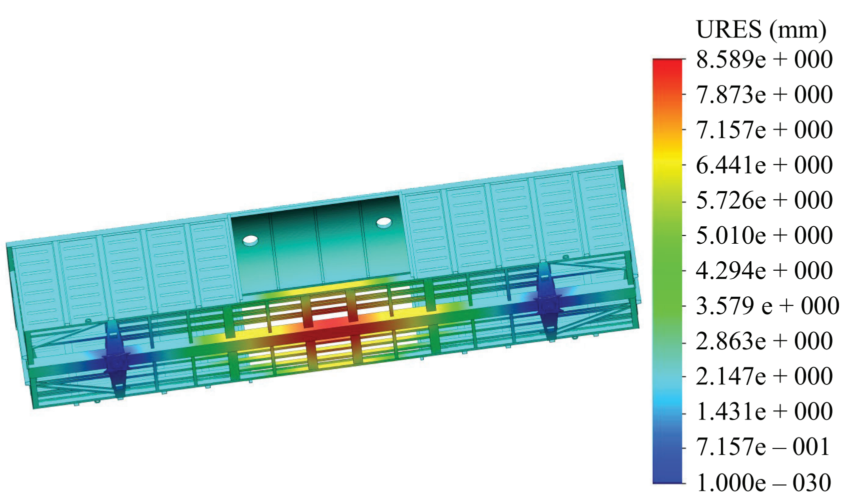

Fig. 9. Displacements in the nodes of the bearing structure of a covered freight car

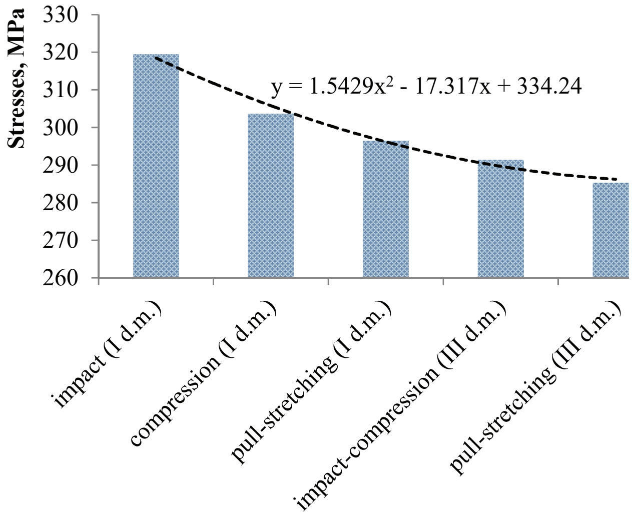

Fig. 10. Results of calculating the strength of the load-bearing structure of a covered freight car

5. 4. Verifying the model of the dynamic load on a covered freight car

Based on the estimation scheme shown in Fig. 7, we determined accelerations in the load-bearing structure of a covered freight car. The obtained accelerations were taken into consideration for verifying the model of the dynamic load on the load-bearing structure of a covered freight car (1) to (3). The calculation results are summarized in Table 2.

Table 2

Numerical values of accelerations acting on the bearing structure of a covered freight car

The force of hitting an autocoupling device, MN	Acceleration value, m/s²	
	Mathematical model	Computer model
2.8	29.4	31.8
2.9	30.1	33.6
3.0	32.5	34.7
3.1	33.7	35.3
3.2	34.3	36.2
3.3	35.0	37.4
3.4	36.1	38.6
3.5	37.0	39.1

It was established that there is a linear relationship between the force of hitting the autocoupling device of a covered freight car and the value of accelerations that arise at the same time (Fig. 11).

The discrepancy between the results of mathematical modeling and computer simulation of the dynamic load on the bearing structure of a covered freight car is shown in Fig. 12.

The maximum percentage of discrepancy was 10.4 %; it occurs with a force of hitting the autocoupling device of 2.9 MN.

Our calculations have demonstrated that with the reproducibility variance S_r=6.24 and the adequacy variance S_{ad}=7.33, the actual value of the F-criterion is F_a=1.17, which is less than the tabular value of the criterion, F_t=3.58, at the level of significance α=0.05. The calculations performed have shown that our hypothesis about the adequacy of the model built is not disputed.

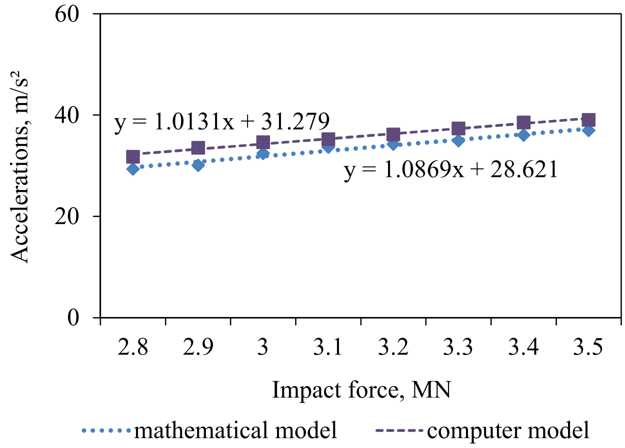

Fig. 11. Dependence of accelerations acting on the bearing structure of a covered freight car on the force of hitting an autocoupling device

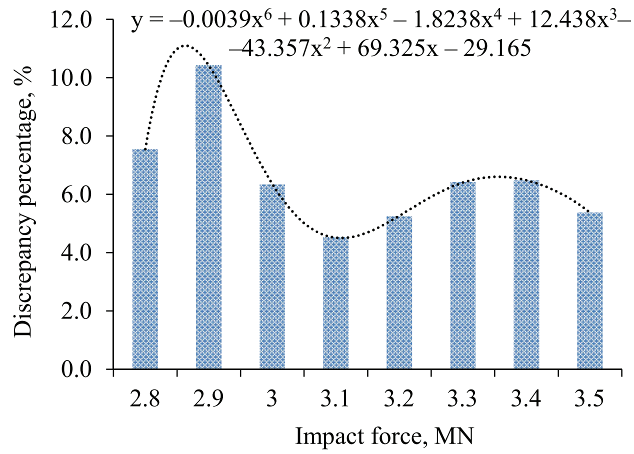

Fig. 12. Discrepancy between the results of mathematical modeling and computer simulation

6. Discussion of results of studying the dynamic load and strength of the bearing structure of a covered freight car

To reduce the load on the girder beam of a covered freight car as the main bearing element of the frame, it is proposed to fabricate it from a closed profile (Fig. 1), filled with filler with viscoelastic properties.

In order to study the dynamic load on the bearing structure of a covered freight car, we performed mathematical modeling. The calculation was carried out for the case of joint impacts at freight car shunting. The results of our calculation showed that the maximum accelerations acting on the bearing structure of a covered freight car were 3.2 % lower than those obtained for the bearing structure without filler (Fig. 5). The limitation of the mathematical model is that it does not take into consideration the deviation of the body of the autocoupling device relative to the longitudinal axis of the freight car. In addition, the impact against the autocoupling device is assumed to be absolutely tough.

We have calculated the maximum equivalent stresses that arise in the bearing structure of a covered freight car. It was established that they are 8 % lower than the permissible ones and amount to 319.5 MPa. It is important to say that the calculations did not take into consideration the friction force between the body heels and the bogie sub-heels.

The built model of the dynamic load on the freight car was verified according to the F-criterion.

As part of further research, it is planned to determine the vertical load on the bearing structure of a covered freight car. In addition, it is necessary to experimentally determine the load on the proposed structure of a covered freight car body. This can be done, for example, by the likeness method.

The study reported here could contribute to advancing the design of innovative rolling stock structures and improve their operational efficiency.

7. Conclusions

1. We have proposed measures to reduce the load on the bearing structure of a covered freight car under operational modes. The peculiarity of the bearing structure of a covered freight car is that the girder beam is made of a closed profile filled with filler with viscoelastic properties. This could contribute to the transformation of the kinetic energy of impact (jerk, stretching, compression) into the work of viscoelastic friction forces.

2. The dynamic load on the bearing structure of a covered freight car with filler in the girder beam was mathematically modeled. The maximum accelerations acting on the load-bearing structure of a covered freight car were about 37 m/s^2. The calculated acceleration value is 3.2 % lower than that obtained for the bearing structure of a covered freight car without filler. When performing calculations, the coefficient of viscous resistance of the material that fills the girder beam was taken equal to 118 kN·s/m, and the rigidity – 80 kN/m.

3. We have calculated the strength of the load-bearing structure of a covered freight car with filler in the girder beam. The calculation was implemented by a finite-element method. The maximum equivalent stresses occur in the zones of interaction between the girder beam and the pivot beams and are 319.5 MPa, which is 8 % lower than permissible ones. The maximum movements occur in the middle part of a covered freight car frame and are equal to 8.6 mm.

4. The model of the dynamic load on a covered freight car with filler in the girder beam was verified. The F-criterion was used as an estimation criterion. The results of our calculations showed that the actual value of the criterion is less than the tabular one at the significance level α=0.05. Therefore, the adequacy hypothesis is not disputed.

Acknowledgments

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