

Проведено визначення показників міцності несучої конструкції вагона-хопера для перевезення окатишів та гарячого агломерату. Розрахунок здійснений за методом скінчених елементів, який реалізовано в програмному забезпеченні CosmosWorks. Визначено резерви міцності несучих елементів кузова. З метою зменшення матеріалоемності кузова вагона запропоновано використання у якості несучих елементів труб круглого перерізу. Для визначення прискорень, які діють на оптимізовану несучу конструкцію вагона при маневровому стівударянні, проведено математичне моделювання. Встановлено, що прискорення, які діють на несучу конструкцію вагона, складають  $42,4 \text{ м/с}^2$  (4,3 g). Отримана величина прискорення врахована при розрахунках на міцність несучої конструкції вагона-хопера. Максимальні еквівалентні напруження при цьому склали близько 270 МПа і зосереджені в зоні взаємодії хребтової балки зі шворневою, але не перевищують допустимі для марки сталі металоконструкції.

Проведено моделювання вертикальної динаміки оптимізованої несучої конструкції вагона-хопера для перевезення окатишів та гарячого агломерату. При проведенні розрахунків враховувалися параметри ресорного підвищення візка моделі 18-100. Результати розрахунків дозволяють зробити висновок, що прискорення кузова вагона-хопера і візків знаходяться межах допустимих. При цьому у відповідності до вимог нормативної документації оцінку ходу вагона можна охарактеризувати як «відмінну».

Запропоновані технічні рішення обґрунтовують використання круглих труб у якості несучих елементів кузова вагона-хопера для перевезення окатишів та гарячого агломерату. При цьому стає можливим знизити тару вагона-хопера майже на 5% у порівнянні з вагоном-прототипом. Також впровадження круглих труб у несучу конструкцію вагона-хопера забезпечить зменшення витрат на виготовлення в умовах вагонобудівних підприємств.

Проведені дослідження сприятимуть створенню сучасних конструкцій вагонів-хоперів, а також підвищенню ефективності експлуатації залізничного транспорту

**Ключові слова:** вагон-хопер, спеціалізований вагон, несуча конструкція, міцність кузова, динамічна навантаженість, оптимізація кузова

# SUBSTANTIATING THE OPTIMIZATION OF THE LOAD-BEARING STRUCTURE OF A HOPPER CAR FOR TRANSPORTING PELLETS AND HOT AGGLOMERATE

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## 1. Introduction

The successful functioning of the transportation industry predetermined the need to introduce modern vehicles into operation. Since the largest share of the transporting process belongs to transportation by rail, the construction of modern designs of railroad cars must meet special conditions. Specifically, this applies to their bearing structures.

One of the most common types of freight cars operated along tracks at industrial enterprises is the hopper cars for transporting pellets and hot agglomerate with a temperature of up to 700 °C. In addition, hopper cars are used to transport

bulk materials that do not require protection against atmospheric precipitation (Fig. 1). Unloading such cars is possible on both sides of the track through unloading hatches [1].

A special feature of these freight cars is that the sheathing of side walls is not welded to the racks but is hung on the frame.

The effect of operational loads on the bearing structure of a hopper car in combination with the temperature load of the cargo contributes to damaging car body elements. This results in the need for appropriate forms of maintenance and repair and leads, accordingly, to additional costs. Besides, damage to a car body bearing structure may threaten the safety of a freight car travel within a train. The peculiarities

of cargoes transported by a hopper car also predetermine the need for more attention being paid to ensuring the safety of movement in terms of environmental aspects.

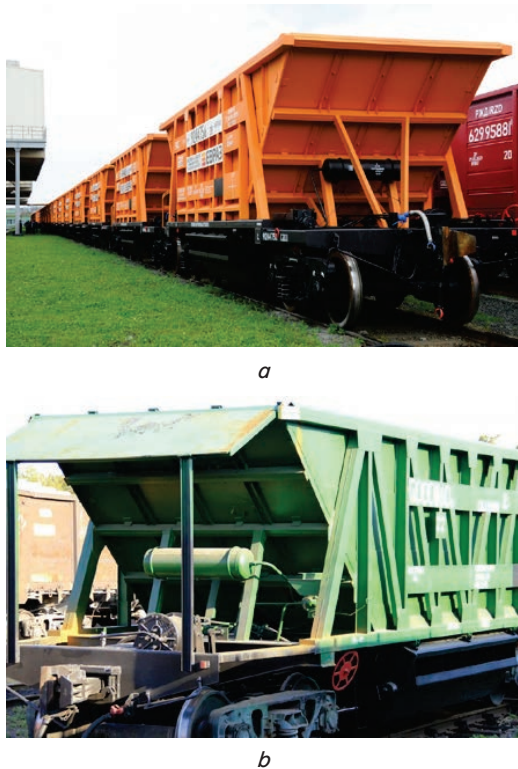


Fig. 1. Hopper cars for transporting pellets and hot agglomerate: *a* – model 20-5197; *b* – model 20-9749

An important area for resolving the issues related to promising hopper cars is the optimization of car bodies. This could make it possible to decrease materials consumption at their manufacturing while maintaining carrying capacity at the level not inferior to the designs of relevant prototype cars. Such a solution would help reduce the cost of freight car manufacturing, as well as their operation, and could improve the efficiency of the transportation process.

## 2. Literature review and problem statement

The optimization of the structure of a supporting device for a tank car used to transport liquid cargoes is described in [2]. The graphic-analytical method was employed by the authors. The calculation of strength was based on a finite element method and implemented in the «Lira» software environment. However, the authors did not pay attention to optimizing the entire bearing structure of a freight car, rather a component of it.

The features of improving the design of a hopper car for transporting grain were highlighted in [3]. A possibility to optimize car body elements was estimated based on the analysis of the most characteristic nodes and structural attributes of specialized bunker freight cars for transporting bulk cargoes. That is the authors built on the experience from operating separate car body components with their subsequent integration in the new structure. At the same time, they failed to substantiate using a given method when optimizing the car body of a freight car for pellets.

Paper [4] reports the structural-parametric optimization of a wagon for transporting cars. The evolutionary scheme of the optimization of the pivot, plate, and plate-rod elements is based on the algorithm that includes the selection of variants for a bearing system employing the criterion of the objective function value. The random changes in the parameters were accounted for, as well as the exchange of parameters between the pairs of options for a bearing system. However, the car body is composed of volumetric elements, which is why reducing them to the rod, plate, and plate-and-rod ones produces a certain error during calculation.

The issue of optimizing the bodies of open-type freight cars with a bearing floor was addressed in [5]. The author constructed an algorithm for the compatible structural and parametric optimization of the side wall and frame of a semi-wagon with a bearing floor under the axial load of 25 t/axle. However, in the cited research the action of principal estimated loads was considered to be quasi-static under normative documents. That is, the author did not determine actual dynamic loads that act on the optimized design of a freight car.

A study [6] reports patterns in the topological optimization of a car body. That involved computer simulation by using the method of finite elements. The research results confirmed the effectiveness of using the proposed methodology for freight car bodies. In this case, the calculation was conducted using an example of a passenger car body. A given procedure did not consider the bearing structure of a freight car for pellets although it is the most loaded type of a freight car in operation.

The structural-optimization concept of designing a car body from aluminum panels was described in [7]. A feature of the manufactured panels is that they are of the «sandwich» type. The characteristic function to search for the optimal combination was determined based on maximum stresses and offsets. However, the proposed freight car design is more expensive in terms of production compared with the entirely-welded one. This hinders the widespread implementation of the proposed solutions at industrial scale.

Paper [8] considers the unified concept of impact strength in order to optimize a rolling stock body. The car body was represented in the form of a rigid frame structure. The optimization problem was to minimize a car body weight taking into consideration constraints for stresses. The authors did not consider the task to reduce materials consumption in the fabrication of a car body for rolling stock.

Studies [9, 10] address the issue of introducing innovative tools in the design of structures for machine-building. The authors apply modern tools of mathematical modeling and computer simulation.

It is important to note that the implementation of such technologies for the load-bearing structures of railroad cars was not considered in the cited works.

Our analysis of literary sources allows us to conclude that the present stage of development of the railroad industry is characterized by the insufficient attention being paid to issues related to the optimization of the specialized rolling stock. This necessitates the substantiation and introduction of new technical solutions for the bearing structures of specialized rolling stock during their design, specifically, freight cars for pellets.

## 3. The aim and objectives of the study

This study aims to substantiate the optimization of the bearing structure of a hopper car for transporting pellets and

hot agglomerate. This would make it possible to reduce the cost of manufacturing a hopper car while meeting the conditions of strength under operating loads.

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

- to determine strength values for a typical bearing structure of the hopper car;
- to determine the strength reserves for basic load-bearing elements of a hopper car body and to build a spatial model of the optimized bearing structure;
- to explore the longitudinal loading of the bearing structure of a hopper car and define its strength indicators;
- to investigate the vertical dynamics of the optimized hopper car.

**4. Determining strength indicators for a typical bearing structure of the hopper car**

Studying the possibility to optimize the bearing structure of a hopper car involved estimation of durability. To this end, we built its spatial model in the SolidWorks software (version 2015) [11].

The chosen prototype of a freight car was the car of model 20-9749, made at DP «Ukrspetswagon» (Ukraine).

To optimize the bearing structure of a hopper car body, the following tasks were solved, which marked the corresponding stages of our study:

- 1) determining the estimated strength reserves for the bearing system of a car model chosen for this study based on the analysis of an integrated theoretical-estimation study of its operation in terms of the perception of operational loads. For this purpose, we built a computer model of the bearing structure of the hopper car model 20-9749 (Fig. 2).

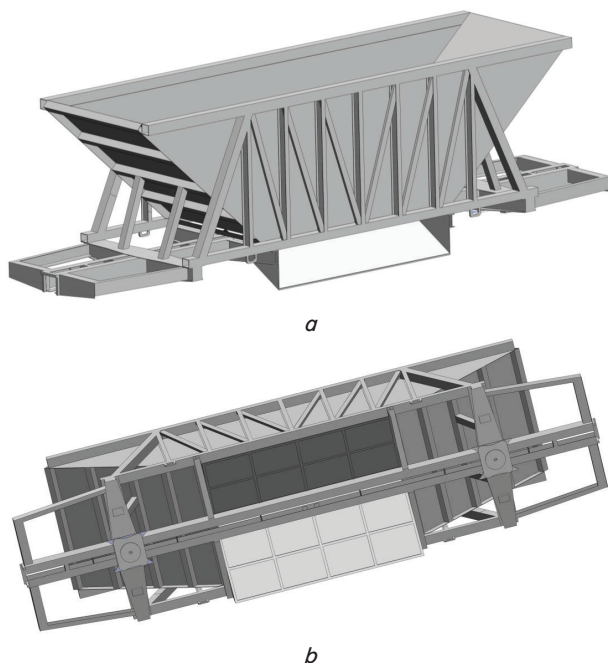


Fig. 2. Spatial geometric model of the bearing structure of the hopper car model 20-9749: a – side view; b – view from below

The basic specifications of the freight car are given in Table 1. The next stage implied simulating the operational

working events and determining the stressed-deformed states by a finite element method;

2) determining the permissible strength characteristics for structural elements of the bearing system of a freight car, chosen as the base, which was performed according to the modern methodology proposed and described in our earlier papers;

3) determining the optimal cross-sections of round pipes proposed for the implementation, taking into consideration the structural and strength constraints;

4) selecting the available types of round pipes from assortment based on the specified optimal parameters;

5) designing a new structure of the hopper car from the selected round pipes;

6) testing the new design of the freight car theoretically and by estimation;

7) analysis of research results.

Table 1

Basic specifications of the hopper car, model 20-9749

Parameter title	Dimensionality
Tare, t	
Minimal	23.5
maximal	25
Load capacity, t	69
Carbody volume, m <sup>3</sup>	45
Maximal estimated static load from a wheelset on rails, kN	235
Base, mm	7,780
Maximal width, mm	3,154
Length based on the axles of couplings, mm	12,000
Motion speed, km/h	not exceeding 40
Dimension in line with GOST 9238-2013	1-VM
Year of production start	2005
Rated service life, years	15

We calculated the bearing structure of a hopper car for strength based on the method of finite elements, implemented in the software environment COSMOSWorks [12–14]. The basic characteristics of the finite-element model of the bearing structure of a hopper car are given in Table 2.

Table 2

Basic characteristics of the finite-element model of the bearing structure of a hopper car

Parameter title	Value
Number of Jacobian points	4
Number of nodes	126,221
Number of elements	376,670
Maximal size of an element, mm	60
Minimal size of an element, mm	12
Minimal number of elements in a circle	12
Ratio of element's size magnification	1.8
Maximal ratio of sides	1,298,6
Percentage of elements with a ratio of sides less than three	7.43
Percentage of elements with a ratio of sides exceeding ten	32.5



The estimation scheme of the bearing structure of a hopper car under the most unfavorable operational mode – when it is struck at shunting – is shown in Fig. 3.

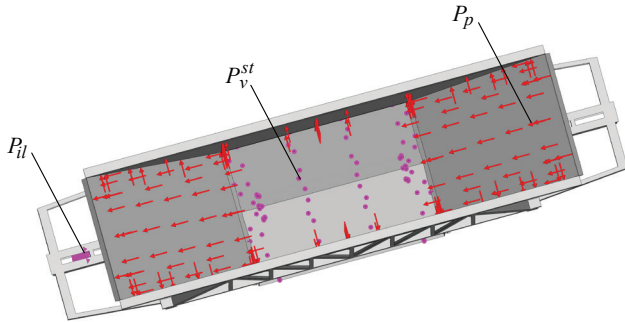


Fig. 3. Estimation scheme of the bearing design of a hopper car when it is struck at shunting

It was taken into consideration that the bearing structure of a car is exposed to the vertical static load  $P_v^{st}$ , predetermined by a cargo weight. In addition, the car body structure is exposed to the spreading efforts from a bulk cargo  $P_p$  whose numerical values were derived from formula (1). The vertical surface of the rear stop is operated under an impact load  $P_{il}$ , whose numerical value, according to normative documents, is 3.5 MN [15–17].

The active pressure of the bulk cargo spread is determined from formula [1]:

$$P_a = \gamma \cdot g \cdot H \cdot \text{tg}^2 \left( \frac{\pi}{4} - \frac{\varphi}{2} \right), \tag{1}$$

where  $\gamma$  is the density of a bulk cargo,  $t/m^3$ ;  $H$  is the height of a side wall, m;  $\varphi$  is the angle of the natural cargo inclination, rad.;  $g$  is the free fall acceleration,  $m/s^2$ .

We fixed the model in the regions when a car body rests on the running parts. The structure’s material is steel, grade 09G2S. The calculation results are shown in Fig. 4, 5.

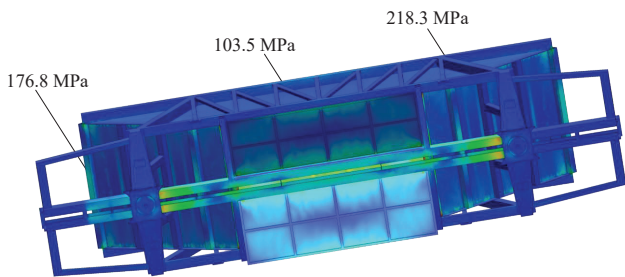


Fig. 4. The stressed state of the bearing structure of a hopper car

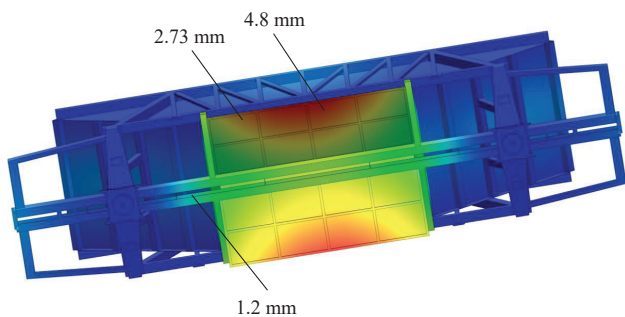


Fig. 5. Displacement at the nodes of the bearing structure of a hopper car

The maximum equivalent stresses in the bearing structure of a hopper car are about 220 MPa; they are concentrated in the region where the girder and pivot beams interact. Maximum displacements occur at unloading bins and make up about 4.5 mm. The maximum deformations are  $3.6 \cdot 10^{-5}$ .

In addition, the calculation was carried out for other operational load diagrams. The calculation results are given in Table 3.

Table 3

Results of calculating the bearing structure of a hopper car for strength

Strength indicator	Loading mode				
	I			III	
	impact	compression	pull-stretching	impact-compression	pull-stretching
Stress, MPa	220	186.6	175.4	196.8	169.5
Displacements at nodes, mm	4.5	3.8	3.7	4.1	3.6
Deformations	$3.6 \cdot 10^{-5}$	$3.4 \cdot 10^{-5}$	$3.3 \cdot 10^{-5}$	$3.5 \cdot 10^{-5}$	$3.2 \cdot 10^{-5}$

Our calculations have made it possible to conclude that the bearing elements of a car body have a significant margin of safety [15–17]. In order to reduce the consumption of materials for the bearing structure of a hopper car, it is necessary to perform its optimization while maintaining the rational strength reserves.

### 5. Determining the strength reserves of the basic bearing elements of a hopper car body and building a spatial model of the optimized structure

To optimize the bearing structure of a hopper car, we used one of the most promising and new methods – a strength reserve optimization.

We determined the permissible strength characteristics for the elements of the bearing structure of the designed hopper car by the following algorithm: first, we determined the permissible values for the resistance momenta of cross-section ( $[W_x]$ ,  $[W_y]$ ) of the implemented profile with the use of the defined strength reserves (determined as the ratio of the derived maximum operational characteristics of strength to their acceptable values). Then, by using the author-developed software and computing package, we determined the optimum characteristics for the components of the car’s elements, followed by deriving the optimum values for the pipe cross-sections, and then, based on the available assortment, we selected the types of pipes.

Thus, the objective function of optimization is to reduce the consumption of materials for the bearing structure of a car body:

$$M_{gw} \rightarrow \min, \tag{1}$$

where  $M_{gw}$  is the gross weight of a car, t.

In the course of our research, we selected the optimum cross-sections of pipes taking into consideration the following constraints:

1) the dimension of a car, that is the optimized structure, was designed considering the existing dimensions of the prototype freight car;

2) the estimated stresses in the optimized structure must be less than the permissible ones:

$$\sigma_{eq} < [\sigma], \tag{2}$$

where  $\sigma_{eq}$  are the equivalent stresses in the structure, MPa;  $[\sigma]$  are the permissible stresses, MPa.

When choosing the optimal parameters for pipes for the girder and main longitudinal beams of the frame, we took into consideration that the cantilever and middle parts of

the beams should have the same wall thickness to enable the process of structure manufacturing.

The results from the above studies are listed in Table 4.

Taking into consideration the selected intersections of pipes we built a model of the improved design of a hopper car (Fig. 6–8).

To substantiate the solutions that we accepted, the following stage of our study implies determining the longitudinal loading on the bearing structure of a hopper car and defining the strength indicators.

Table 4

Determining the optimum parameters for intersections of elements in the bearing structure of a hopper car

Frame element	Mass 1 m, kg	Length, m	n	$\sigma_{eq}$ , MPa	$I_x$ , cm <sup>4</sup>	$I_y$ , cm <sup>4</sup>	$W_x$ , cm <sup>3</sup>	$W_y$ , cm <sup>3</sup>	[ $W_x$ ], cm <sup>3</sup>	[ $W_y$ ], cm <sup>3</sup>	Pipe optimal parameters			Mass of 1 m of round pipe, kg
											$W$ , cm <sup>3</sup>	D, mm	S, mm	
Girder beam	66.5	10.73	1.15	218.3	27696	808	1231	101	1070.4	87.8	1072.26	530.0	5.0	64.74
Transverse middle beam	7.53	1189	1.53	163.7	98.3	30.6	19.66	9.71	12.85	6.34	14.56	83.0	3.0	5.92
Upper strapping	32.34	9190	2.03	123.5	1589.15	1589.15	198.6	198.6	97.83	97.83	98.39	159.0	5.5	20.82
Lower strapping	29.27	8330	1.51	165.3	586.79	1230.39	73.35	246.1	48.58	162.98	175.83	219	5.0	26.39
Rack	13.7	2035	2.0	125.7	572	41.9	81.7	11.5	40.85	5.75	43.3	140.0	3.0	10.14
Brace	8.59	2094	2.17	115.6	175.0	22.6	34.9	7.37	16.1	3.4	25.28	108.0	3.0	7.77
Pipe brace	32.34	2197.4	2.2	114.4	1589.15	1589.15	198.6	198.6	90.3	90.3	90.3	159.0	5.0	18.99
Belt	17.1	2819	1.76	142.5	469.71	469.71	78.285	78.285	44.48	44.48	44.7	114.0	5.0	13.44
Extreme channel	8.59	3365	1.24	202.7	175.0	22.6	34.9	7.37	28.15	5.94	28.29	114.0	3.0	8.21
Pipe brace	17.1	1225	1.17	213.9	469.71	469.71	78.285	78.285	66.9	66.9	69.11	140.0	5.0	16.65

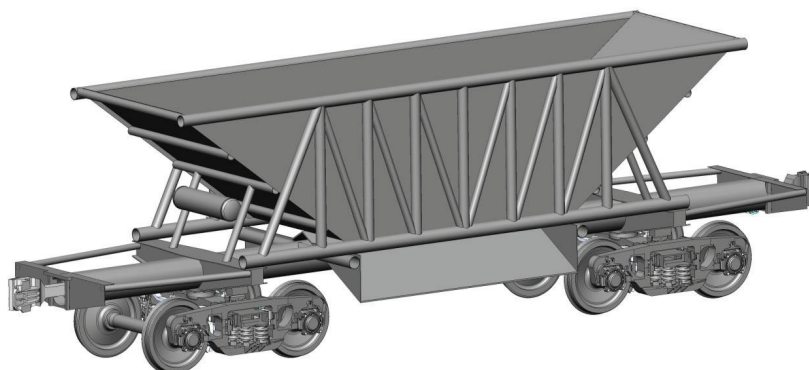


Fig. 6. The hopper car made from round pipes

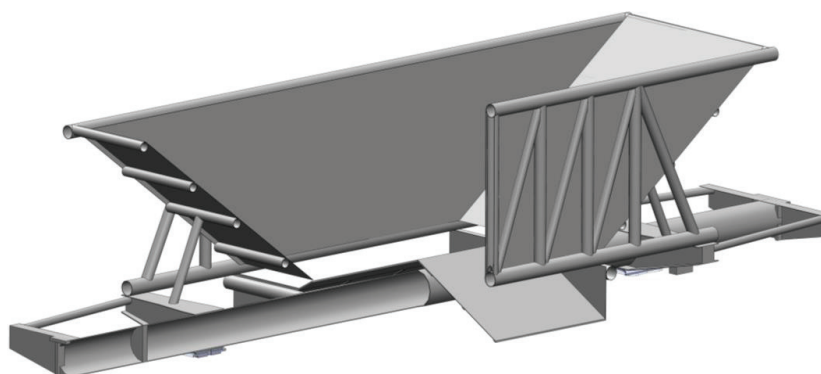


Fig. 7. Bearing structure of the hopper car made from round pipes

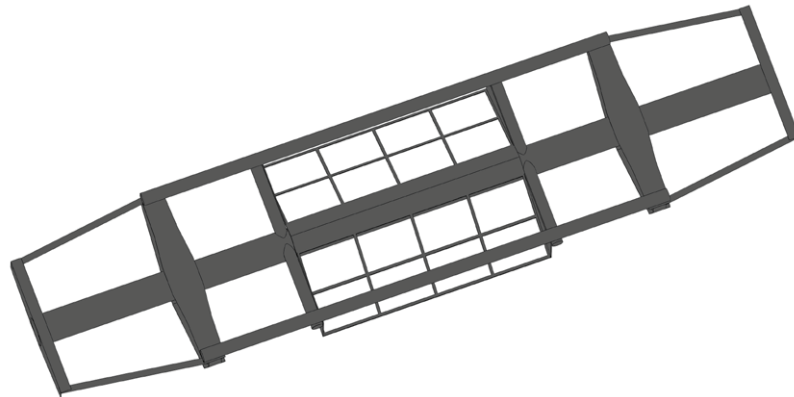


Fig. 8. A frame of the hopper car made from round pipes

**6. Studying the longitudinal loading on the bearing structure of a hopper car and determining its strength indicators**

**6.1. Studying the longitudinal loading on the bearing structure of a hopper car**

The mathematical model given in [18] was applied to determine the dynamic loads acting on a car body when it is struck at shunting. A given model is constructed to determine accelerations as a component of the dynamic loading, which act on a platform car hosting containers-tanks when it is struck at shunting. Therefore, the model was refined to determine the accelerations as a component of the dynamic loading operating on a freight car under the action of a longitudinal impact force.

$$M'_C \cdot \ddot{x}_C + M' \cdot \varphi_C = S_m, \tag{1}$$

$$I_C \cdot \ddot{\varphi}_C + M' \cdot \ddot{x} - g \cdot \varphi_C \cdot M' = F_\varphi, \tag{2}$$

$$M_C \cdot z_C = C_1 + C_2 - F_{FR} (\text{sign} \dot{\Delta}_1 - \text{sign} \dot{\Delta}_2), \tag{3}$$

where

$$M'_C = M_C + 2 \cdot m_u + \frac{n \cdot I_W}{r^2}; \quad M' = M_C \cdot h;$$

$$F_\varphi = l \cdot F_{FR} (\text{sign} \dot{\Delta}_1 - \text{sign} \dot{\Delta}_2) + l(C_1 - C_2);$$

$$C_1 = k_1 \cdot \Delta_1; \quad C_2 = k_2 \cdot \Delta_2;$$

$$\Delta_1 = z_C - l \cdot \varphi_C; \quad \Delta_2 = z_C + l \cdot \varphi_C.$$

$M_C$  is the mass of the bearing structure of a freight car;  $I_C$  is the moment of inertia of a freight car relative to the longitudinal axis;  $S_m$  is the magnitude of the longitudinal force of impact against an auto-coupling;  $m_u$  is the weight of the undercarriage;  $I_W$  is the moment of inertia of a wheelset;  $r$  is the radius of an average worn wheel;  $n$  is the number of undercarriage axles;  $l$  is the half of a freight car base;  $F_{FR}$  is the absolute value of a dry friction force in the spring set;  $k_1, k_2$  is the spring rigidity of the undercarriage spring suspensions;  $x_C, \varphi_C, z_C$  are the coordinates corresponding to the longitudinal, angular relative to the transverse axis, and the vertical freight car movement, respectively.

The differential equations were solved in the software package Mathcad [19–22].

The initial displacement and speed are taken to equal zero. The input parameters of the model are the technical characteristics of a car body, spring suspension, as well as the value of the longitudinal impact against an auto-coupling.

During calculations, we took into consideration the parameters of the spring suspension of the model 18-100 undercarriage.

The longitudinal impact force, which acts on the vertical surface of the rear stop of the auto-coupling, was adopted to equal 3.5 MN [15–17].

The calculation results have made it possible to conclude that the accelerations, which act on the bearing structure of a freight car when it is struck at shunting, amount to 42.4 m/s<sup>2</sup> (4.3 g).

**6.2. Determining the indicators of strength for the bearing structure of a hopper car**

To determine the strength of the bearing structure of a hopper car made from round pipes we performed calculations based on the method of finite elements.

The basic characteristics of the finite element model are given in Table 5.

Table 5

Basic characteristics of the finite element model of the bearing structure of a hopper car made from round pipes

Parameter title	Value
Number of Jacobian points	4
Number of nodes	808,646
Number of elements	2,480,364
Maximal size of an element, mm	20
Minimal size of an element, mm	4
Minimal number of elements in a circle	8
Ratio of element's size magnification	1.6
Maximal ratio of sides	183.35
Percentage of elements with a ratio of sides less than three	37.8
Percentage of elements with a ratio of sides exceeding ten	1.51

The estimation scheme of the bearing structure of a hopper car is shown in Fig. 9. The designations of loads accepted in the estimation scheme are identical to those considered when calculating a typical bearing structure of a hopper car.

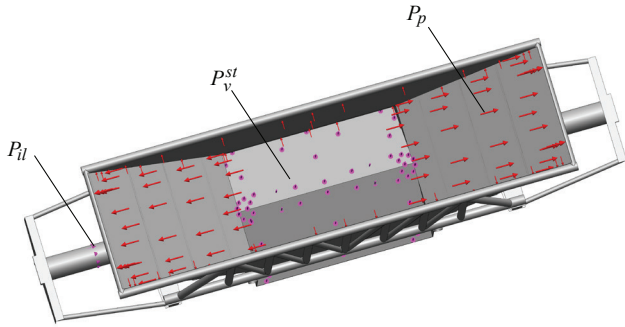


Fig. 9. Estimation scheme of the bearing structure of a hopper car when it is struck at shunting

The results from calculating the strength of the bearing structure of a hopper car when it is struck at shunting are shown in Fig. 10.

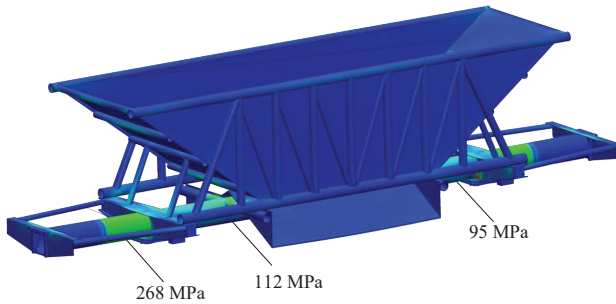


Fig. 10. The stressed state of the bearing structure of a hopper car

The distribution of stresses lengthwise the girder beam of a hopper car is shown in Fig. 11.

The distribution of stresses lengthwise the girder beam of a hopper car when it is struck at shunting is shown in Fig. 11.

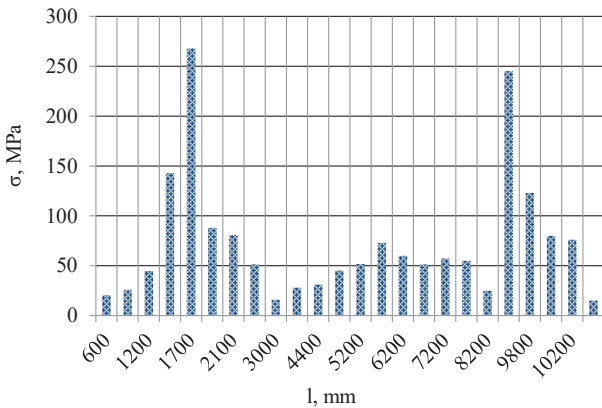


Fig. 11. Distribution of stresses lengthwise the girder beam of hopper car

In this case, the maximum equivalent stresses in the bearing structure of a hopper car are about 270 MPa and are concentrated in the region where the girder and pivot beams interact while not exceeding the permissible ones [15–17].

The maximum displacements in the bearing structure of a hopper car occur at unloading bins and are about 5.2 mm (Fig. 12).

The maximum deformations in the bearing structure of a hopper car are  $5.7 \cdot 10^{-5}$ .

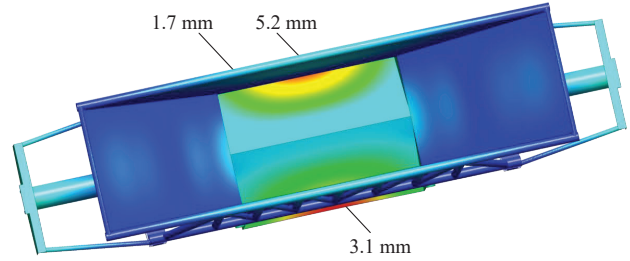


Fig. 12. Displacements at the nodes of the bearing structure of a hopper car

In addition, we calculated other operational load diagrams. The calculation results are given in Table 6.

Table 6

Results from calculating the bearing structure of a hopper car for strength

Strength indicator	Loading mode				
	I			III	
	impact	compression	pull-stretching	impact-compression	pull-stretching
Stress, MPa	270	234.2	215.3	246.8	203.5
Displacements at nodes, mm	5.2	4.4	3.7	4.8	3.2
Deformations	$5.7 \cdot 10^{-5}$	$5.9 \cdot 10^{-5}$	$6.1 \cdot 10^{-5}$	$5.8 \cdot 10^{-5}$	$6.3 \cdot 10^{-5}$

Our calculations have made it possible to conclude that the strength of the bearing structure of a hopper car is ensured.

### 7. Studying the vertical dynamics of the optimized hopper car

In order to determine the vertical accelerations of the optimized bearing structure of a hopper car, we used the mathematical model given in [23]. The calculation was carried out in a flat coordinate system. The equations of motion of the estimated model take the form:

$$M_1 \cdot \frac{d^2}{dt^2} q_1 + C_{1,1} \cdot q_1 + C_{1,3} \cdot q_3 + C_{1,5} \cdot q_5 = -F_{FR} \cdot \left( \text{sign} \left( \frac{d}{dt} \delta_1 \right) + \text{sign} \left( \frac{d}{dt} \delta_2 \right) \right), \quad (4)$$

$$M_2 \cdot \frac{d^2}{dt^2} q_2 + C_{2,2} \cdot q_2 + C_{2,3} \cdot q_3 + C_{2,5} \cdot q_5 = F_{FR} \cdot l \cdot \left( \text{sign} \left( \frac{d}{dt} \delta_1 \right) + \text{sign} \left( \frac{d}{dt} \delta_2 \right) \right), \quad (5)$$

$$M_3 \cdot \frac{d^2}{dt^2} q_3 + C_{3,1} \cdot q_1 + C_{3,2} \cdot q_2 + C_{3,3} \cdot q_3 + B_{3,3} \cdot \frac{d}{dt} q_3 = F_{FR} \cdot \text{sign} \left( \frac{d}{dt} \delta_1 \right) + k_1 (\eta_1 + \eta_2) + \beta_1 \left( \frac{d}{dt} \eta_1 + \frac{d}{dt} \eta_2 \right), \quad (6)$$

$$M_4 \cdot \frac{d^2}{dt^2} q_4 + C_{4,4} \cdot q_4 + B_{4,4} \cdot \frac{d}{dt} q_4 = -k_1 (\eta_1 - \eta_2) - \beta_1 \cdot a \cdot \left( \frac{d}{dt} \eta_1 - \frac{d}{dt} \eta_2 \right), \quad (7)$$



$$M_5 \cdot \frac{d^2}{dt^2} q_5 + C_{5,1} \cdot q_1 + C_{5,2} \cdot q_2 + C_{5,5} \cdot q_5 + B_{5,5} \cdot \frac{d}{dt} q_5 = F_{FR} \cdot \text{sign} \left( \frac{d}{dt} \delta_2 \right) + k_1 (\eta_3 + \eta_4) + \beta_1 \left( \frac{d}{dt} \eta_3 + \frac{d}{dt} \eta_4 \right), \quad (8)$$

$$M_6 \cdot \frac{d^2}{dt^2} q_6 + C_{6,6} \cdot q_6 + B_{6,6} \cdot \frac{d}{dt} q_6 = -k_1 \cdot a \cdot (\eta_3 - \eta_4) - \beta_1 \cdot a \cdot \left( \frac{d}{dt} \eta_3 - \frac{d}{dt} \eta_4 \right), \quad (9)$$

where  $M_i$  is the inertial coefficients for the elements of an oscillatory system;  $C_i$  is the characteristic of elasticity for the elements within a vibratory system;  $B_i$  is the scatter function;  $a$  is the half of an undercarriage base;  $q_i$  is the generalized coordinates that correspond to the translational and angular displacements around the vertical axis, respectively, of a car body, the first and the second undercarriages, as well as a cargo;  $k_i$  is the rigidity of a spring suspension;  $\beta_i$  is the damping coefficient;  $F_{FR}$  is the force of absolute friction in a spring assembly.

It was taken into consideration that the estimation scheme of a car includes three solids: a car body and two undercarriages. The interaction between a car body and the undercarriages is enabled by elastic-friction elements [23].

In this case, equations (4) and (5) characterize the displacement of a car body at oscillations due to jumps and galloping. Equations (6) and (7) refer to the undercarriage traveling first in the forward direction, (8) and (9) – second.

The mathematical model was solved in the Mathcad software.

The initial displacement and speeds are taken to equal zero. The input parameters to the model are the technical characteristics of a car body, a spring suspension, as well as the disturbing actions.

During calculations, we took into consideration the parameters of a spring suspension of the model 18-100 undercarriage.

The calculation results are shown in Fig. 13, 14.

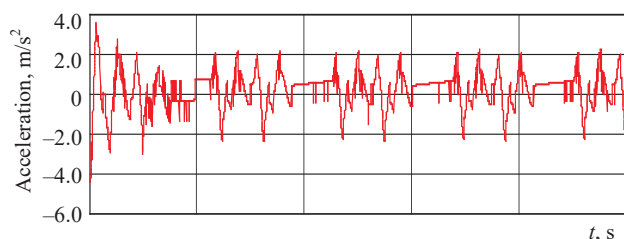


Fig. 13. The character of change in the accelerations of a hopper car body in the center of masses over time

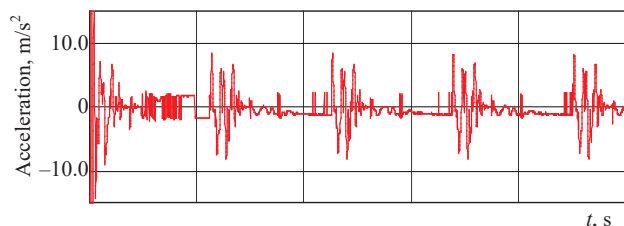


Fig. 14. The character of change in the accelerations of a hopper car's undercarriages over time

Our calculations make it possible to conclude that the accelerations of a hopper car body and the undercarriages

are within the allowable limits. In this case, in accordance with the requirements of normative documents, the freight car ride quality can be characterized as «excellent» [15, 16].

### 8. Discussion of results of optimizing the bearing structure of a hopper car for transporting pellets and hot agglomerate

To reduce the consumption of materials for the bearing structure of a hopper car that transports pellets and hot agglomerate, we performed calculations for strength. The reserves of strength have been determined and it has been proposed to use round pipes as the load-bearing elements of the structure. It is important to note that the proposed car body design is rather non-trivial given that our analysis of modern structures of freight cars has revealed that no such solutions had been adopted in their design. However, it becomes possible to reduce the freight car tare by almost 5 % compared to a prototype freight car.

The proposed solutions have been justified by our theoretical calculations for strength using the method of finite elements. It has been established that the maximum equivalent stresses are within the admissible ones and are about 270 MPa. In this case, they are concentrated in the region where the girder and pivot beams interact. In the console parts of the car body, the stresses accept the minimum value.

However, the calculation for strength did not consider a possible deviation of the auto-coupling body in the horizontal plane. This could affect the angle of applying the load to the stops of the auto-coupling, which are in the girder beam. Further studies need to account for this limitation.

The designed bearing structure of a hopper car is meant for the vertical dynamics. A known mathematical model was used for this purpose. The most common types of car body oscillations during operation were taken into consideration: bouncing and galloping. The maximum acceleration, which operates on a car body, was about 0.4 g. Consequently, the assessment of a freight car ride quality while maintaining the safety of motion is excellent.

The obtained results allow us to conclude that it is appropriate to use round pipes as the bearing elements of a hopper car body. This could be an alternative solution when constructing modern freight car structures.

It is important to note that the estimation method used in building the bearing structure of a freight car, strength reserve optimization, is much simpler compared to available ones. Our calculation of the car body strength has made it possible to draw a conclusion about its reliability.

However, further studies must involve experimental determining of the strength indicators of the optimized bearing structure of a hopper car. That may be done through a physical experiment using the similarity method.

Our research could contribute to constructing the fundamentally new modern structures of rolling stock using non-standard car body components while maintaining appropriate strength parameters.

### 9. Conclusions

1. We have calculated the strength of a typical bearing structure of the hopper car using the method of finite elements. The research was undertaken in the COSMOSWorks



software environment. The most unfavorable operating modes at a freight car body loading have been taken into consideration. When constructing an estimation car body model, we have taken into account those design elements that rigidly interact with each other.

The maximum equivalent stresses in this case occur in the region where the girder and pivot beams interact and are about 220 MPa. The maximum displacements emerge at unloading bins and make up about 4.5 mm. The maximum deformations are  $3.6 \cdot 10^{-5}$ .

It has been established that the bearing elements of a freight car body have a significant margin of safety. In order to reduce the consumption of materials for the bearing structure of a hopper car, it is necessary to perform its optimization while maintaining the rational strength reserves.

2. The reserves of strength for the basic bearing elements of a hopper car body have been determined and a spatial model of the optimized bearing structure has been built. In this case, it has been proposed to use pipes with a circular cross-section as the bearing elements of the car body. That has made it possible to reduce the hopper car tare by almost 5 % compared to a prototype freight car.

3. We have determined the dynamic loads that act on a car body when it is struck at shunting as the case of the largest loading on the structure during operation. It has been established that under the action of a longitudinal loading of

3.5 MN on the rear stop of an auto-coupling the accelerations that act on the bearing structure of a car body are about 4.3 g. The derived magnitude of acceleration has been taken into consideration when calculating the strength of the bearing structure of a hopper car. The maximum equivalent stresses are about 270 MPa and are concentrated in the region where the girder and pivot beams interact while not exceeding the permissible levels. The maximum displacements in the bearing structure of a hopper car occur at unloading bins and are about 5.2 mm, the maximum ones are  $5.7 \cdot 10^{-5}$ .

4. The vertical dynamics of the optimized hopper car has been investigated. The calculation was carried out in a flat coordinate system. In this case, the accelerations of the bearing structure of a hopper car were 0.4 g, the acceleration of undercarriages – about 1 g. At the same time, in accordance with the requirements of normative documents, the freight car ride quality can be described as «excellent».

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### References

1. Lukin, V. V., Shadur, L. A., Koturanov, V. I., Hohlov, A. A., Anisimov, P. S. (2000). *Konstruirovaniye i raschet vagonov*. Moscow, 731.
2. Vatulia, G., Falendysh, A., Orel, Y., Pavliuchenkov, M. (2017). Structural Improvements in a Tank Wagon with Modern Software Packages. *Procedia Engineering*, 187, 301–307. doi: <https://doi.org/10.1016/j.proeng.2017.04.379>
3. Kebal, Y., Shatov, V., Tokotyev, A., Murashova, N. (2017). Improving the design of hopper wagons for transporting grain. *Zbirnyk naukovykh prats DETUT. Seriya: «Transportni systemy i tekhnolohiyi»*, 30, 113–122.
4. Serpik, I. N., Sudarev, V. G., Tyutyunnikov, A. I., Levkovich, F. N. (2008). Evolyutsionnoye modelirovaniye v proektirovaniye nesushchih sistem vagonov. *Vestnik Vserossiyskogo Nauchno-Issledovatel'skogo Instituta Zheleznodorozhnogo Transporta*, 5, 21–25.
5. Beyn, D. G. (2011). Analiz napryazhennogo sostoyaniya nesushchego nastila pola chetyrehosnogo poluvagona s gluhim kuzovom. *Vestnik Bryanskogo gosudarstvennogo tekhnicheskogo universiteta*, 1 (29), 47–51.
6. Kuczek, T., Szachniewicz, B. (2015). Topology optimisation of railcar composite structure. *International Journal of Heavy Vehicle Systems*, 22 (4), 375. doi: <https://doi.org/10.1504/ijhvs.2015.073206>
7. Lee, H.-A., Jung, S.-B., Jang, H.-H., Shin, D.-H., Lee, J. U., Kim, K. W., Park, G.-J. (2015). Structural-optimization-based design process for the body of a railway vehicle made from extruded aluminum panels. *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, 230 (4), 1283–1296. doi: <https://doi.org/10.1177/0954409715593971>
8. Mrzygłód, M., Kuczek, T. (2013). Uniform crashworthiness optimization of car body for high-speed trains. *Structural and Multidisciplinary Optimization*, 49 (2), 327–336. doi: <https://doi.org/10.1007/s00158-013-0972-z>
9. Kučera, P., Pištěk, V. (2017). Testing of the mechatronic robotic system of the differential lock control on a truck. *International Journal of Advanced Robotic Systems*, 14 (5), 172988141773689. doi: <https://doi.org/10.1177/1729881417736897>
10. Pistek, V., Klimes, L., Mauder, T., Kucera, P. (2017). Optimal design of structure in rheological models: an automotive application to dampers with high viscosity silicone fluids. *Journal of Vibroengineering*, 19 (6), 4459–4470. doi: <https://doi.org/10.21595/jve.2017.18348>
11. Alyamovskiy, A. A. (2007). *SolidWorks/COSMOSWorks 2006–2007. Inzhenernyy analiz metodom konechnykh elementov*. Moscow, 784.
12. Alyamovskiy, A. A. (2010). *COSMOSWorks. Osnovy rascheta konstruksiy na prochnost' v srede SolidWorks*. Moscow, 785.
13. Lovskaya, A. (2015). Computer simulation of wagon body bearing structure dynamics during transportation by train ferry. *Eastern-European Journal of Enterprise Technologies*, 3 (7 (75)), 9–14. doi: <https://doi.org/10.15587/1729-4061.2015.43749>
14. Fomin, O., Gerlici, J., Lovska, A., Kravchenko, K., Prokopenko, P., Fomina, A., Hauser, V. (2019). Durability Determination of the Bearing Structure of an Open Freight Wagon Body Made of Round Pipes during its Transportation on the Railway Ferry. *Communications - Scientific Letters of the University of Zilina*, 21 (1), 28–34.

15. DSTU 7598:2014. Vahony vantazhni. Zahalni vymohy do rozrakhunkiv ta proektuvannia novykh i modernizovanykh vahoniv koliyi 1520 mm (nesamokhidnykh) (2015). Kyiv, 162.
16. GOST 33211-2014. Vagony gruzovye. Trebovaniya k prochnosti i dinamicheskim kachestvam (2016). Moscow, 54.
17. EN 12663-2. Railway applications - structural requirements of railway vehicle bodies – Part 2: Freight wagons (2010). BSI, 54. doi: <https://doi.org/10.3403/30152552u>
18. Bogomaz, G. I., Mehov, D. D., Pilipchenko, O. P., Chernomashentseva, Yu. G. (1992). Nagruzhennost' konteynerov-tsistern, raspolozhennyh na zheleznodorozhnoy platforme, pri udarah v avtostsepku. Dynamika ta keruvannia rukhom mekhanichnykh system, 87–95.
19. Kelrykh, M., Fomin, O. (2014). Perspective directions of planning carrying systems of gondolas. Metallurgical and Mining Industry, 6, 64–67.
20. Fomin, O. (2014). Modern requirements to carrying systems of railway general-purpose gondola cars. Metallurgical and Mining Industry, 5, 40–44.
21. Kondratiev, A. V., Gaidachuk, V. E., Kharchenko, M. E. (2019). Relationships Between the Ultimate Strengths of Polymer Composites in Static Bending, Compression, and Tension. Mechanics of Composite Materials, 55 (2), 259–266. doi: <https://doi.org/10.1007/s11029-019-09808-x>
22. Lovska, A. (2018). Simulation of Loads on the Carrying Structure of an Articulated Flat Car in Combined Transportation. International Journal of Engineering & Technology, 7 (4.3), 140–146. doi: <https://doi.org/10.14419/ijet.v7i4.3.19724>
23. Domin, Yu. V., Cherniak, H. Yu. (2003). Osnovy dynamiky vahoniv. Kyiv: KUETT, 269.