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Concept of Freight Wagons Made of Round Pipes

Monograph

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Abstract

The purpose of the book is to highlight the results and features of the research carried out by the authors on the dynamics and strength of the main types of freight wagons, as well as tank containers made of round pipes under the main operating conditions of loading. Theoretical provisions, methodological foundations and practical solutions for the implementation of round pipes as bearing elements of bodies of the main types of wagons and tank containers are presented.

The monograph is intended for scientific and technical specialists, whose activities are related to the design and research of the mechanics of structures of railway wagons, including scientists, designers, researchers, doctoral students and graduate students.

Also, the results presented in the monograph may be of interest to specialists, whose activities are related to the design of tank containers.

The monograph can be used as a teaching aid for undergraduates and bachelors of relevant specialties.

Keywords

Transport mechanics; railway transport; optimization of wagons; dynamics and strength of wagons; bearing structures of wagons.

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Introduction

The fruitful functioning of the transport industry necessitates the implementation of modern vehicles into operation. Since the main segment of the transportation process is assigned to rail transport, special conditions should be imposed on the creation of modern wagon designs. In particular, this applies to their bearing structures.

To reduce the material consumption of the bearing structures of wagons, as well as tank containers, while ensuring the conditions for strength and operational reliability, it has been proposed and scientifically justified to use round pipes as elements of their bearing systems.

The aim of research is to highlight the features and results of the research carried out to determine the feasibility of using round pipes as bearing elements of railway wagons and tank containers. The paper presents examples of the practical implementation of round pipes in the bearing systems of the main types of wagons: flat wagon, gondola wagon, covered wagon, hopper wagon, as well as 1CC tank container. The possibility of using round pipes in the manufacture of articulated wagons has been substantiated. The concept of an automatic coupler fitting is proposed, which can be implemented on wagons, the center beams of which have a closed section.

To solve the main objective of research — to substantiate the expediency of using round pipes as elements of bearing systems of railway wagons, the following scientific and practical objectives are set:

1. Justify the implementation of round pipes as bearing elements of railway wagons and tank containers. Conduct comprehensive strength calculations of bearing elements of railway wagons and tank containers made of round pipes.
2. Create a concept of an automatic coupler fitting to reduce the dynamic loading of the bearing structures of wagons made of round pipes under operating conditions.
3. Carry out mathematical modeling of the dynamic loading of the bearing structures of wagons made of round pipes equipped with a fitting concept.
4. Carry out a computer simulation of the dynamic loading of the bearing structures of round-pipe wagons equipped with a fitting concept.
5. Check the adequacy of the developed models.
6. Create computer models of the main types of articulated wagons from round pipes.

7. Carry out mathematical modeling of dynamic loading of bearing structures of articulated wagons made of round pipes.

8. Conduct mathematical modeling of the dynamic loading of the bearing structures of articulated wagons made of round pipes equipped with a fitting concept.

The object of research is the process of loading the bearing structures of wagons from round pipes under operating conditions of loading.

The subject of research is the bearing structures of wagons made of round pipes.

The monograph is intended for scientific and technical specialists, whose activities are related to the design and research of the mechanics of structures of railway wagons, including scientists, designers, researchers, doctoral students and graduate students.

The monograph can be used as a teaching aid for undergraduates and bachelors of relevant specialties.

Chapter 1
Features of the creation of bearing structures of wagons made of round pipe

1.1 Description of the general procedure for optimizing the bearing structures of wagons

Constant competition between railway transport and other modes of transport both on the domestic market for cargo transportation and within international transport corridors necessitates the design and implementation of innovative wagon designs (wagons with significantly improved technical, economic and operational performance). In this case, special attention should be paid to the creation of the optimal parameters of their structures.

Optimal structural design comes from the inception of general design. Optimal design of wagon structures, as a section of optimal design, is formed in the 60s of the XX century.

The works of V. Lukin, V. Lozbinev, F. Lozbinev, A. Savchuk, A. Bitiutskyi, S. Sorokina, V. Tsarapkin, Ya. Kulbovskyi, N. Podlitov and etc. are devoted to the development of the theory of optimal design of wagons.

To reduce the material consumption of the bearing structures of wagons while ensuring the conditions of strength and operational reliability, it is proposed to use round pipes as elements of their bearing systems (Fig. 1.1).

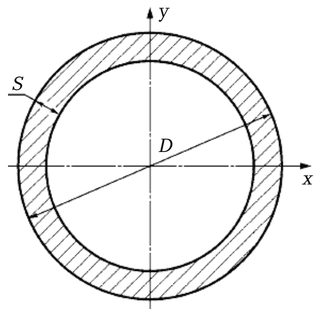


Fig. 1.1 Section of a round pipe

One of the most promising and new methods is used to optimize the bearing structures of the wagons — optimization by strength reserves.

The research is carried out in the following stages [1–3]:

— 1 determination of the design strength reserves of the bearing system of the wagon model selected for the study based on the analysis of complex theoretical and design studies of its work on the perception of operational loads. For this, computer models of wagons of the main types are developed and their adequacy is checked, simulated operational working situations and stress-strain states are determined by the finite element method;

– 2 determination of the permissible strength characteristics for the structural elements of the bearing systems of the wagons, selected as basic ones, which is carried out according to modern methodology;

– 3 determination of the optimal cross-sections of circular pipes proposed for implementation, taking into account structural and strength constraints;

– 4 selection of existing versions of round pipes from the assortment on the basis of certain optimal parameters;

– 5 development of new designs of railway wagons with selected round pipes;

– 6 comprehensive theoretical and design verification of new wagon designs, which includes calculations: for the first and third design modes, for fatigue strength, and determination of the design service life;

– 7 analysis of research results.

The determination of the permissible strength characteristics for the elements of the bearing structures of the wagons being developed is carried out according to the following algorithm: first, the permissible values of the cross-section resistance moments ($[W_x]$, $[W_y]$) of the introduced profile are determined using certain strength reserves (determined as the ratio of the obtained maximum operational strength characteristics to their permissible values). After that, with the help of the software-computing complex developed by the author, the optimal characteristics of the constituent elements of the wagon are determined, the optimal values of the pipe sections are obtained, after which, using the assortment, the existing pipe designs are determined [4–6].

So the target optimization function is to reduce the material consumption of the bearing body structure:

$$M_{gr} \to \min, \tag{1.1}$$

where M_{gr} – the gross weight of the wagon, t.

When conducting research, the choice of optimal pipe sections is carried out taking into account the following restrictions:

1) wagon size, that is, the optimized design, is designed taking into account the existing size of the prototype wagon;

2) calculated stresses in the optimized design must be less than the permissible ones:

$$\Sigma_{eq} < [\sigma], \tag{1.2}$$

where σ_{eq} – equivalent stresses in the structure, MPa; $[\sigma]$ – allowable stress, MPa.

When choosing the optimal parameters of the pipes of the center and main longitudinal beams of the frame, it is taken into account that the cantilever and middle parts of the beams must have the same wall thickness with the manufacturability of the structure.

1.2 Optimization of the bearing structure of a flat wagon

The results of the analysis of the experience of reliable operation of the existing models of flatcars indicated that the model of the flatcar 13-401 manufactured by JSC Dniprodzerzhynsk WBP can be taken as the base for the study.

In order to study the stress-strain state of the bearing structure of the flat wagon, its spatial model is built (Fig. 1.2) in the SolidWorks software environment.

The results of checking its adequacy, by comparing the calculated values with known experimental data, indicated the possibility of further application. After that, to simulate operational working situations (according to the first and third design modes) and determine the stress states of the bearing elements, the following work is carried out.

The numerical values of the forces acting on the flat wagon in operation are calculated in accordance with [7 – 9] are given in Tables 1.1, 1.2. At the same time, it is taken into account that the full bearing capacity of the platform wagon is used with a conventional load. Vertical forces acting on the bearing structure of a flat wagon are applied in a ratio of 5/16 to the main side beams, 10/16 to the center beams [10, 11].

Strength analysis is carried out using the finite element method in the CosmosWorks software environment [12]. The optimal number of mesh elements of the finite element model is determined using the graphical-analytical method. The number of grid elements is 368732, nodes is 14938. The maximum size of a grid element is 235.62 mm, the minimum is 47.12 mm, the maximum aspect ratio of elements is 332, the percentage of elements with an aspect ratio of less than three is 24.6, more ten is 31.5.

The fixing of the model is carried out at center plates and side beams of the pivot beams of the bearing structure of the flat wagon.

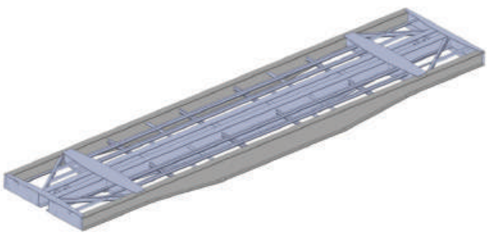


Fig. 1.2 Spatial geometric computer model of a flat wagon model 13-401

Table 1.1 Numerical values of the forces acting on the model 13-401 flat wagon in operation

Force type	I d.m.	III d.m.
Vertical static, kN	800.496	800.496
Vertical dynamic, kN		112.87
Centrifugal, kN		150.81
Framed, kN		194.21
Wind, kN		2.65

Table 1.2 Values of longitudinal forces acting on a flat wagon in operation

Longitudinal force, MN			
Design modes			
I		III	
Quasi-static force	Hit, jerk	Quasi-static force	Hit, jerk
−3.0 +2.5	−3.5 +2.5	−1.0 +1.0	−1.0 +1.0

In the study of the strength of the flat wagon under load conditions, the longitudinal force corresponding to the «shock-compression» mode is applied to the rear stop of the automatic coupler, and on the other side of the flat wagon, the automatic coupler equipment is attached to the same element. When modeling the strength of a flatcar under the conditions of the «stretch-jerk» mode, the longitudinal force is applied to the front stops from one end of the flatcar, and the front stops are fixed on the other.

The calculation results are shown in Table 1.3.

Table 1.3 Results of strength analysis of the bearing structure of a flat wagon

Strength index	Load mode				
	hit	compres-sion	jerk-stretch	hit-com-pression	jerk-stretch
Tension, MPa	306	286.3	242.8	224	230.8
Displacement in knots, mm	7.6	7.4	7.5	78.2	6.8
Deformations	$2.4 \cdot 10^{-3}$	$2.4 \cdot 10^{-3}$	$2.4 \cdot 10^{-3}$	$3.3 \cdot 10^{-3}$	$5.7 \cdot 10^{-3}$

The obtained results allow to conclude that the maximum equivalent stresses in the bearing structure of a flatcar arise at the AND design mode under impact conditions. At the same time, in the constituent elements of the frame, the maximum equivalent stresses are much less than the permissible ones and have a significant margin of safety. Therefore, in order to reduce the material consumption of the bearing structure of the flat wagon, it is necessary to optimize it with the provision of rational safety margins by using their surplus.

The results of determining the optimal parameters of the cross-sections of the elements of the bearing structure made of round pipes of the model 13-401 flat wagon are shown in Table 1.4.

Table 1.4 — Determination of the optimal parameters of the sections of the bearing structure elements of the flat wagon model 13-401

Frame element	Mass 1 m, kg	Length, m	n	σ_{eq}, MPa	I_x, cm^4	I_y, cm^4	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3	Optimal pipe parameters			Mass of 1 m round pipe, kg
											W, cm^3	D, mm	S, mm	
Center beam:														
console part	108*	13.4	1.13	306.09	1853.82	24516.94	195.14	1343.4	173.13	1191.86	1204	325	17	129.13
middle part			5.99	57.6	1862.44	77164.35	196.05	1286.07	32.74	214.88	506.07	219	17	84.69
Main longitudinal beam:														
console part	108*	13.4	8.58	40.21	1853.82	24516.94	195.14	1343.4	22.74	156.57	173.39	177.8	8	33.5
middle part			6.77	50.95	1862.44	77164.35	196.05	1286.07	28.96	189.95	208.11	193.7	8	36.64
Intermediate longitudinal beam	9.46	9.7	14.4	24	199.45	18.79	75.53	3.76	5.3	1.8	6.53	57	3	4.0
Intermediate cross beam	9.46	1142	4.1	84	199.45	18.79	75.53	3.76	18.13	0.92	20.04	76	5.5	9.56
Thrust	11.5	2091.5	5.32	64.87	420.26	50.21	110.6	8.37	20.8	1.57	21.73	89	4	8.38
Console corner (short)	5.72	1246	12.6	27.4	11.18	42.94	3.55	13.63	0.28	1.08	6.53	57	3	4.0
Console corner (long)	5.72	1918	28.3	12.2	11.18	42.94	3.55	13.63	0.12	0.48	6.53	57	3	4.0

* *The table shows the weight of 1 linear meter of I-beam No. 60.*

When choosing the optimal parameters for the pipes of the center and main longitudinal beams of the frame, it is taken into account that the cantilever and middle parts of the beams must have the same wall thickness with the manufacturability of the structure.

To check the structural performance, manufacturability, strength according to the first and third design modes, fatigue strength, and to determine the design service life of a flat wagon with a bearing system of round pipes, the authors developed its computer model in the SolidWorks software environment.

Taking into account the data given in Table 1.4, a spatial model of a flat wagon with optimal parameters of structural elements is built (Fig. 1.3).

The interaction of the spine beam with the pivot is carried out through a special adapter (Fig. 1.4), which consists of a support 1 and base plates 2.

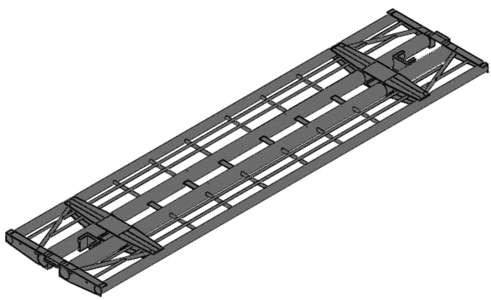


Fig. 1.3 Bearing structure of the platform wagon model 13-401 made of round pipes

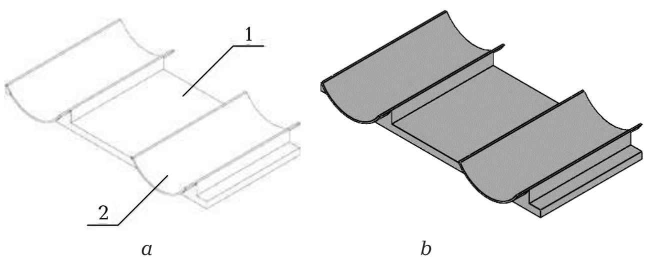


Fig. 1.4 Pivot beam adapter: *a* — design features; *b* — spatial model

The thickness of the support is selected based on the thickness of the I-beam of the center beam of the prototype platform wagon. This technical solution allows to provide the necessary strength of the pivot beam in the zone of interaction with the spine under operating loads.

In order to ensure the fastening of the longitudinal beams with the transverse ones, the latter have special cutouts, 1 mm deep in the longitudinal beams that are enclosed (Fig. 1.5).

The node of interaction of the longitudinal beams of the flat wagon with the transverse ones is shown in Fig. 1.6.

In order to study the strength of a flat wagon with an improved design, a calculation is made using the finite element method. A computer model of the strength of a flat wagon in the conditions of the design mode is shown in Fig. 1.7.

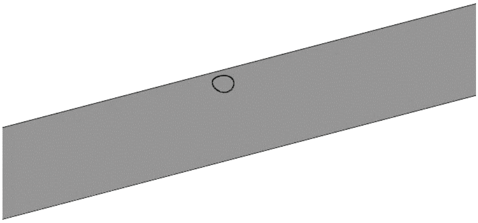


Fig. 1.5 Cross beam of a flat wagon

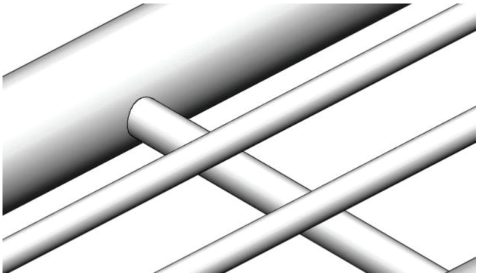


Fig. 1.6 Interaction node of longitudinal beams with transverse

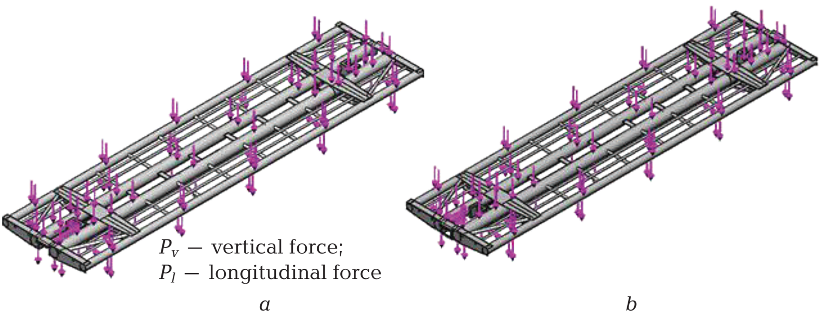


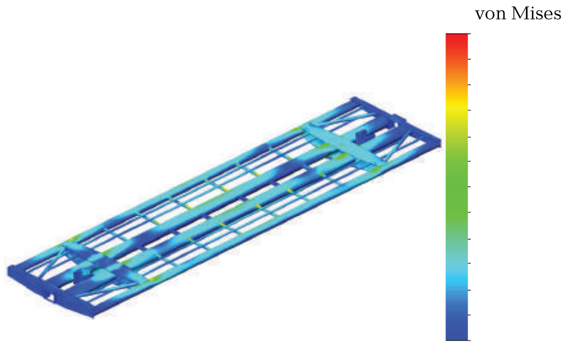
Fig. 1.7 — Computer model of the strength of a flat wagon of an improved design under I design mode: *a* — hit-compression; *b* — stretch-jerk

The optimal number of mesh elements is determined using the graphical-analytical method. Isoparametric tetrahedrons are used as finite elements. The number of grid elements is 2.821.871, nodes is 797860. The maximum size of a grid element is 15 mm, the minimum is 3 mm, the maximum aspect ratio of elements is 105.9, the percentage of elements with an aspect ratio of less than three is 89, more than ten is 0.197.

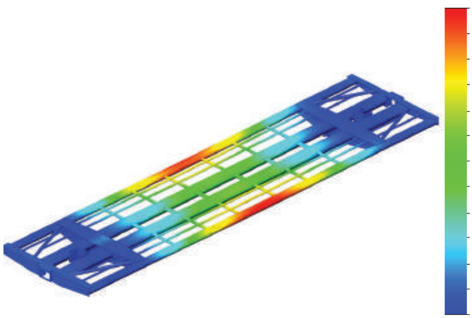
The limitations of the model are the absence of a difference in the levels of the bodies of automatic couplers of platform wagons interacting with each other.

The fixing of the model is carried out at center plates and side beams of the pivot beams of the bearing structure of the flat wagon.

The results of calculating the strength of the bearing structure of a flatcar under the I design mode (impact) are shown in Fig. 1.8.



a



b

Fig. 1.8 The results of the strength calculation of the platform wagon of the improved design in the conditions of the I design mode (impact): *a* — stressful state; *b* — movement in nodes

In this case, the maximum equivalent stresses arise in the lower zone of interaction of the pivot beam with the center beam and are about 320 MPa, the maximum displacements in the structural nodes are recorded in the middle part of the main longitudinal frame beams and are 26.5 mm, the maximum deformations are $1.96 \cdot 10^{-3}$.

The distribution of equivalent stresses along the length of the center beam is shown in Fig. 1.9. In this case, stress and deformations are recorded with the lower part of the spine pipe.

The Fig. 1.9 shows that the maximum stresses arise in the cantilever parts of the center beam of the flat wagon. In the console located on the side where the shock load is applied, the stresses are greater than on the opposite side by almost 40 %.

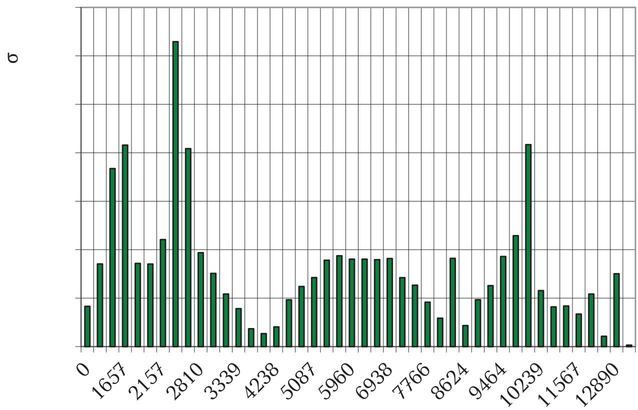


Fig. 1.9 Distribution of equivalent stresses over length of the center beam

The maximum equivalent stresses at the I design mode (jerk, tension) arise in the lower zone of interaction of the pivot beam with the center beam and are about 300 MPa, the maximum displacements in the nodes of the structure are 27.8 mm, the maximum deformations are.

Under conditions of «compression» beyond the design mode I, the maximum equivalent stresses are about 280 MPa, the maximum displacements in the nodes are 26.5 mm, and the deformations are $18 \cdot 10^{-3}$.

The maximum equivalent stresses under IRS in the design mode (impact, compression) are about 250 MPa, the maximum displacements in the nodes of the structure are 25 mm, the maximum deformations are $1.88 \cdot 10^{-3}$.

With «jerk-tension» the maximum equivalent stresses are about 240 MPa, the maximum displacements in the nodes of the structure are 27.7 mm, the maximum deformations are.

The developed platform wagon design is designed for fatigue strength in the CosmosWorks software environment [12]. The test base in this case is made up of cycles [13]. The results of the calculation made it possible to conclude that the fatigue strength of the bearing structure of the flat wagon is provided.

In order to determine the design service life of a flat wagon, the method described in [14] is used:

$$T_p = \frac{\left(\sigma_p/[n]\right)^m \cdot N_0}{B \cdot f_e \cdot \sigma_a^m}, \tag{1.3}$$

where σ_p – average value of the endurance limit of the part, MPa; n – permissible safety factor; m – indicator of the degree of the fatigue curve; N_0 – test base; B – coefficient characterizing the time of continuous operation of the object in seconds; f_e – effective frequency of dynamic stresses, s; σ_a – amplitude of equivalent dynamic stresses, MPa.

The coefficient characterizing the time of continuous operation of the object is determined by the formula:

$$B = \frac{365 \cdot 10^3 \cdot L_d}{\vartheta_{av}(1+0.34)}, \tag{1.4}$$

where L_d – average daily run of the wagon, km (km [14]); ϑ_{av} – average value of the speed of the wagon, m/s; 0.34 – empty run ratio.

The effective frequency of dynamic stresses is determined by the formula:

$$f_5 = \frac{1{,}1}{2\pi}\sqrt{\frac{g}{f_{st}}}, \tag{1.5}$$

where f_{st} – static deflection of the spring suspension, mm.

When carrying out the calculations, the following input parameters are taken: the average value of the endurance limit of the bearing structure is determined as a material (steel grade 09G2D, O9G2S) and amounted to 245 MPa; test base – cycles (recommended test base for steel [13]); the time of continuous operation of the bearing structure at $\vartheta_{av} = 33.3$ m/s is 6514.37 s; the effective frequency of dynamic stresses is determined taking into account the parameters of the spring suspension of the model 18-100 bogie and amounted to 2.7 Hz; the permissible safety factor is 2; the index of the degree of the fatigue curve for the welded structure is taken equal to 4; the amplitude of the equivalent dynamic stresses is determined on the basis of the performed calculations of the stress-strain state of the wagon's bearing structure and amounted to 50.6 MPa.

On the basis of the calculations, it is established that the design service life of the bearing structure of the improved flat wagon is more than 32 years, that is, not less than the life cycle of the wagon. The improved bearing structure of the flat wagon (Fig. 1.10 [15]) has a container of about 6.2 tons, which is 4 % less than the container of the prototype wagon (6.5 tons).

Fig. 1.10 Flat wagon made of round pipes

It is important to note that taking into account the reduced packaging of the bearing structure of the platform wagon, its strength is ensured under operational loads. In addition, the developed design provides greater convenience in servicing the braking equipment of the wagon, in contrast to the prototype wagon.

1.3 Optimization of the bearing structure of a gondola wagon

To substantiate the expediency of introducing round pipes as bearing elements of gondola wagon bodies, a model 12-757 gondola wagon, built by PJSC «KRCBW», is chosen (Fig. 1.11).

To determine the strength reserves of the gondola wagon, its spatial model is built in the SolidWorks software environment. Strength analysis is carried out in the CosmosWorks software package using the finite element method.

Fig. 1.11 Bearing structure of a gondola wagon model 12-757

The numerical values of the forces acting on the gondola in operation are shown in Table 1.5. In this case, it is taken into account that the full bearing capacity of the wagon is used. Coal is accepted as a bulk cargo, since this type of cargo is the most common for transportation on the territory of Ukraine.

Table 1.5 Numerical values of the forces acting on the model 13-401 flat wagon in operation

Force type	I d.m.	III d.m.
Vertical static, kN	829.926	829.926
Vertical dynamic, kN		99.59
Centrifugal, kN		156.35
Framed, kN		202.94
Wind, kN		17.1

Efforts to spread bulk cargo on the side walls and end doors of the gondola wagon body, determined by the method described in [10]. According to this method, it is assumed that the load of the bulk load on the side walls of the wagon body is distributed according to the law of a triangle with a maximum at its base, and on the end wall according to the law of a trapezoid.

The maximum loads at the foundations of the side wall struts are determined by:

$$q_1 = 0.5 \cdot p_a \cdot l_1, \tag{1.6}$$

$$q_2 = 0.5 \cdot p_a \cdot (l_1 + l_2), \tag{1.7}$$

$$q_3 = 0.5 \cdot p_a \cdot (l_2 + l_3), \tag{1.8}$$

$$q_4 = 0.5 \cdot p_a \cdot (l_3 + l_4), \tag{1.9}$$

where p_a — active (static) pressure of the expansion of the bulk cargo, which is per unit area of the surface of the vertical wall at the floor level, kPa; l_1 — distance from the final frame beam to the geometric axis of the center plate of the wagon, m; l_2 — distance from the geometric axis of the center plate of the wagon to the second body post, m; l_3 — distance from the second body post to the third, m; l_4 — distance from the third body post to the vertical geometric axis of the wagon body, m.

The active expansion pressure of the bulk cargo is determined by the formula:

$$p_a = \gamma \cdot g \cdot H \cdot \mathrm{tg}^2\left(\frac{\pi}{4} - \frac{\varphi}{2}\right), \tag{1.10}$$

where γ — bulk cargo density, t/m; H — side wall height, m; φ — cargo repose angle, rad; g — acceleration of gravity, m/s.

The calculated values of the efforts of the expansion of the bulk cargo on the side walls of the gondola wagon body are shown in Table 1.6.

Table 1.6 Numerical value of the expansion pressure of the bulk cargo on the elements of the side wall of the gondola wagon body

Side wall post	Bulk pressure, kPa
corner:	
I d.m.	7.083
III d.m.	23.6
the first from the console side:	
I d.m.	13.03
III d.m.	43.42
the second from the console side:	
I d.m.	11.895
III d.m.	39.64
the third from the console side:	
I d.m.	8.92
III d.m.	29.73

The pressure of the unevenly distributed load applied to the leaf of the end door is determined by the formula:

$$p = p_a + p_p, \tag{1.11}$$

where p_p — passive pressure of the bulk cargo, which is determined by the formula (1.10), in which the square of the tangent of the difference between the two angles is replaced by the square of the tangent of their sum and taking into account the coefficient of vertical dynamics, as well as the angle of repose.

The intensity of the trapezoidal load on the corner post is determined by:

$$q^l_{T1} = 0.5\left(p_a + p_p\right)\cdot b_1; \tag{1.12}$$

$$q^t_{T1} = 0.5\cdot p_p \cdot b_1; \tag{1.13}$$

on the intermediate post:

$$q^l_{T2} = 0.5\left(p_a + p_p\right)\cdot\left(b_1 + b_2\right); \tag{1.14}$$

$$q^t_{T2} = 0.5\cdot p_p \cdot\left(b_1 + b_2\right); \tag{1.15}$$

on the middle post:

$$q^l_{T3} = 0.5\left(p_a + p_l\right)\cdot b_2; \tag{1.16}$$

$$q^t_{T3} = 0.5\cdot p_l \cdot b_2. \tag{1.17}$$

The calculated values of the efforts of the expansion of the bulk cargo on the end wall of the gondola wagon body are shown in Table 1.7.

Table 1.7 Numerical value of the expansion pressure of the bulk cargo on the elements of the end door of the gondola wagon body

End door element	Bulk pressure, kPa
corner post:	
I d.m.	50.06* 44.98**
III d.m.	35.39* 18.45**
intermediate post:	
I d.m.	100.12* 89.96**
III d.m.	70.78* 36.9**
middle post:	
I d.m.	50.06* 44.98**
III d.m.	35.39* 18.45**

**pressure on the post bottom; ****pressure on the post top*

The optimal number of grid elements is determined using the graphical-analytical method. The number of grid elements is 473652, nodes — 154365. The maximum size of a grid element is 80 mm, the minimum is 16 mm, the maximum aspect ratio of elements is 566.7 percent of elements with an aspect ratio of less than three — 25, more than ten — 27.4.

The calculation results are shown in Table 1.8.

Table 1.8 Results of strength analysis of the gondola wagon bearing structure

Strength index	Load mode				
	hit	compression	jerk-stretch	hit-compression	jerk-stretch
Tension, MPa	320	280	280	250	260
Displacement in knots, mm	8.72	6.64	6.6	6.2	6.8
Deformations	$1.15 \cdot 10^{-3}$	$1.1 \cdot 10^{-3}$	$1.13 \cdot 10^{-3}$	$1.03 \cdot 10^{-3}$	$1.1 \cdot 10^{-3}$

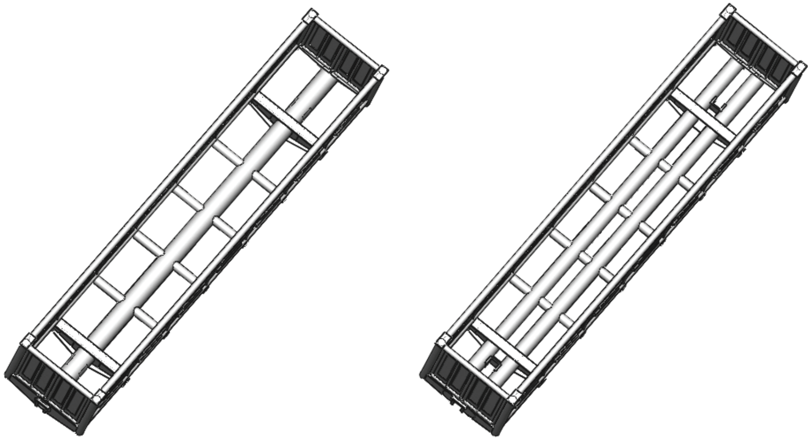
The results of determining the optimal parameters of the sections of the elements of the bearing structure made of round pipes of the gondola wagon model 12-757 are shown in Table 1.9.

Taking into account the data given in Table 1.9, the bearing structure of the body of the universal gondola wagon has been optimized. This takes into account two options for the design of the frame. The first design option provides for the implementation of the center beam of the frame from one pipe (Fig. 1.12, *a*), the second — two pipes (Fig. 1.12, *b*).

Table 1.9 — Determination of the optimal parameters of the sections of the elements of the bearing structure of the gondola wagon model 12-757

Frame element	Mass 1 m, kg	Length, m	n	σ_{eq}, MPa	I_x, cm^4	I_y, cm^4	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3	Optimal pipe parameters			Mass of 1 m round pipe, kg
											W, cm^3	D, mm	S, mm	
Central beam	154.05	12.8	1.1	312.4	41512.19	62884.88	1317.95	2566.7	1186.1	2310.03	2330.12	406.4	21.0	199.6
Intermediate cross beam	37.2	25.4	2.8	70.7	1274.0	22954.96	124.9	1010.12	44.6	360.8	378.15	244.5	9.0	52.27
Bottom fitting	19.48	33.2	1.12	308	784.0	239.0	156.8	30	140.0	26.8	153.57	168.0	8.0	31.57
Top fitting	24.7	33.2	3.2	107.8	1090.49	3063.89	99.14	523.74	30.98	163.7	173.39	177.8	8.0	33.5
Corner post	85	9.6	8.4	41.3	6179.3	23983.45	294.95	2694.8	35.1	320.81	328.15	219.0	10.0	51.54
Post	28.7	48.4	6.0	57.6	347.44	868.18	118.2	119.3	19.7	19.8	20.04	76.0	5.5	9.56
End door intermediate post	28.7	8.1	7.32	47.1	228.88	625.9	54.2	145.6	7.4	19.89	20.04	76.0	5.5	9.56
End door middle post	28.7	4.0	4.1	84.2	460.83	1934.4	69.3	154.4	16.9	37.66	38.18	102.0	5.5	13.09
End door top fitting	31.45	5.72	3.7	94.01	745.19	1397.15	240.89	248.4	65.1	67.14	74.69	152.0	4.5	16.37
End door bottom fitting	15.15	5.72	1.8	189.1	278.29	333.82	51.54	69.55	28.63	38.64	40.77	114.0	4.5	12.15
End post	56.32	11.0	3.42	100.9	2067.2	11958.88	99.38	1272.22	29.1	372.0	383.0	219.0	12.0	61.26

In order to ensure the possibility of mounting and dismantling the automatic coupler harness, the diameter of the center beam pipe when it is executed from one profile, increased to 530 mm.



a *b*

Fig. 1.12 Optimized bearing structure of the body of a gondola wagon model 12-757: *a* — center beam is represented by one round pipe; *b* — center beam is represented by two round pipes

One of the design features of the optimized gondola wagon model is the difference in height between the pivot and cross beams of the frame (Fig. 1.13).

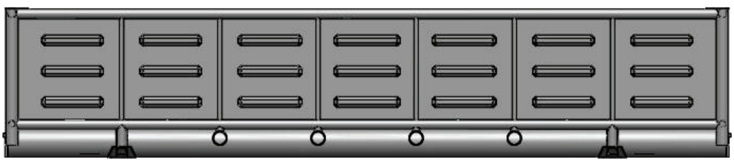


Fig. 1.13 Optimized bearing structure of gondola wagon model 12-757 (side view)

Due to the fact that in the model the Ω-shaped profile of the vertical post has been changed to a round pipe, the location of the vertical sheets of the pivot beam has undergone changes. It is proposed to carry out their fan-shaped placement, that is, in the interaction zone of the pivot beam with the center beam, the distance between the vertical sheets remained the same as in the prototype wagon, and in the zone of its interaction with the vertical post, the distance between the vertical sheets is reduced to 164 mm. It is important to note that the known designs of pivot beams of gondola wagons with a similar arrangement of vertical sheets [16].

The vertical body post in the interaction zone with the cross beam has an increased diameter compared to its main part (Fig. 1.14). This technical solution is due to the fact that a significant portion of the load acting on the post is perceived by the node pinching it with the cross beam.

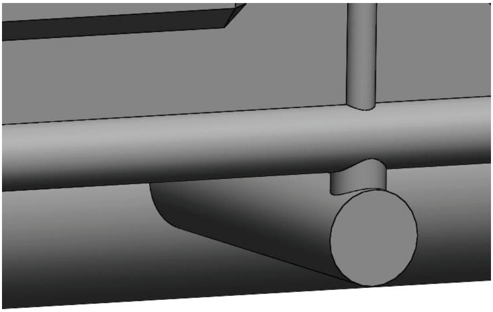


Fig. 1.14 Pinching assembly of a vertical post with a transverse beam of the frame

In order to test the strength of the optimized bearing structure of the gondola wagon body according to the first embodiment of the center beam, a calculation is performed using the finite element method.

The number of grid elements is determined using the graphic-analytical method and amounted to 633837, nodes — 201679. The maximum size of a grid element is 85 mm, the minimum is 17 mm, the maximum aspect ratio of elements is 695.86, the percentage of elements with an aspect ratio of less than three is 28.1, more than ten — 12. When generating the mesh, simplifications of the model are used in the areas where the rounding and holes are located, the coefficient of simplification of the model is 0.2 with the number of mesh division iterations equal to two.

The strength model of the gondola wagon body of optimized design is shown in Fig. 1.15.

The computer model of strength does not take into account the difference in the levels of gondola wagon couplers interacting with each other.

The fixing of the model is carried out at center plates and side beams of the pivot beams of the wagon's bearing structure.

The results of calculating the strength of the bearing structure of a gondola wagon at the I design mode (impact) are shown in Fig. 1.16.

The maximum equivalent stresses arise in the interaction zone of the lower part of the strut with the transverse beam of the frame and are about 320 MPa, the maximum displacements in the nodes of the structure are 5.72 mm, the maximum deformations are $1.15 \cdot 10^{-3}$.

The results of calculating the strength of the bearing structure of the gondola wagon with an optimized design under the I design mode (jerk) show that the maximum equivalent stresses arise in the interaction zone of the lower part of the vertical post with the transverse beam of the frame and are about 270 MPa, the maximum displacements in the structural units are 8.44 mm, the maximum deformations are $1.85 \cdot 10^{-3}$.

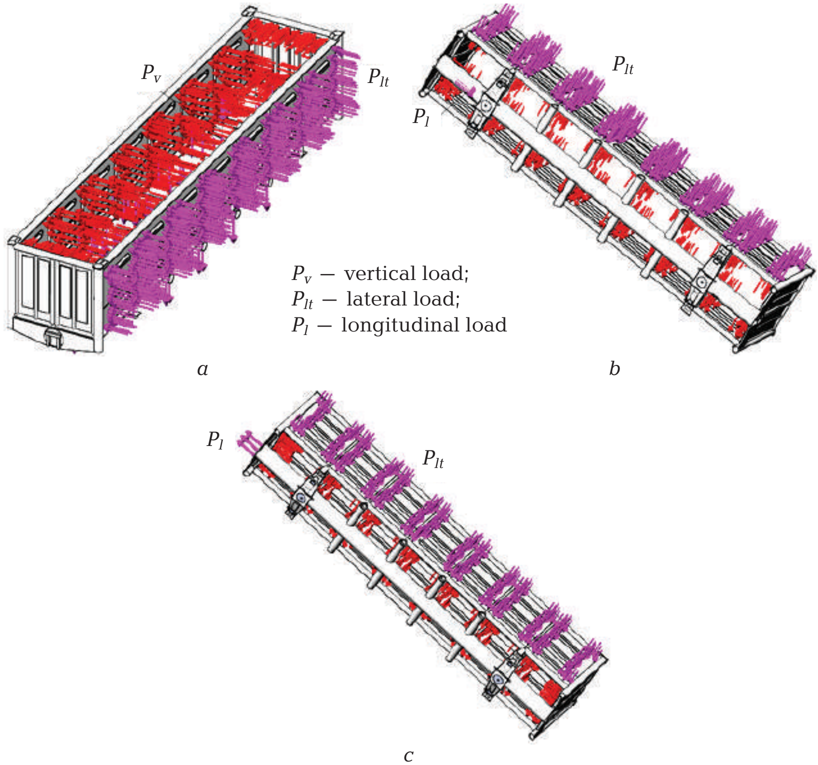
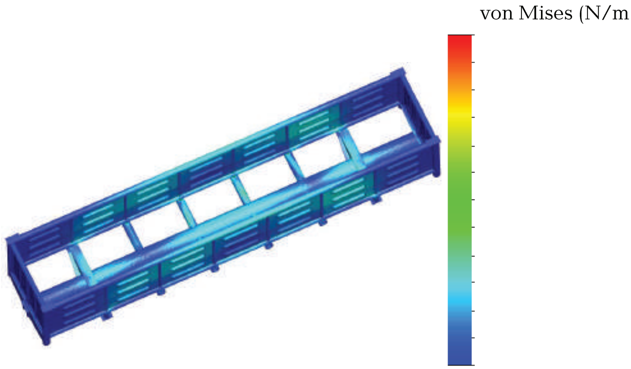


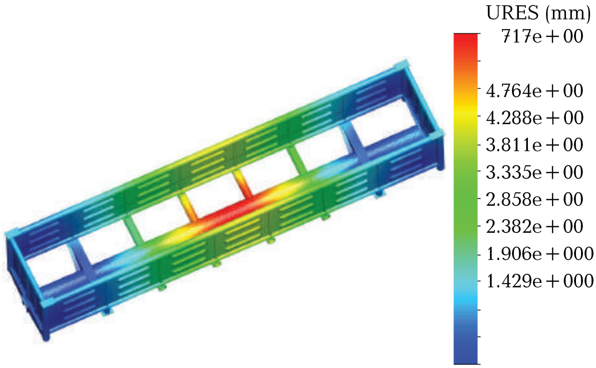
Fig. 1.15 — Computer model of the strength of the optimized bearing structure of a gondola wagon model 12-757: *a* — side view; *b* — bottom view (hit, compression); *c* — bottom view (stretch, jerk)

In the design mode (compression), the maximum equivalent stresses are about 280 MPa, the maximum displacements in the structural units are 15.1 mm, and the maximum deformations are $5.13 \cdot 10^{-3}$.

The results of calculating the strength of the bearing structure of a gondola wagon under III the design mode (impact, compression) made it possible to conclude that the maximum equivalent stresses are about 250 MPa, the maximum displacements in the structural units are 15 mm, and the maximum deformations are $4.53 \cdot 10^{-3}$. In this case, the vertical dynamic force that acts on the gondola wagon body when moving relative to the rail track is taken into account in quasi-static.



a



b

Fig. 1.16 The results of strength analysis of the bearing structure of a gondola wagon model 12-757 under I design mode (impact): *a* — stressful state; *b* — movement in the nodes of the structure

The results of calculating the strength of the gondola wagon bearing structure under the conditions of III design mode (jerk, stretching) show that the maximum equivalent stresses are about 260 MPa, the maximum displacements in the structure nodes are 8.3 mm, and the maximum deformations are $1.83 \cdot 10^{-3}$.

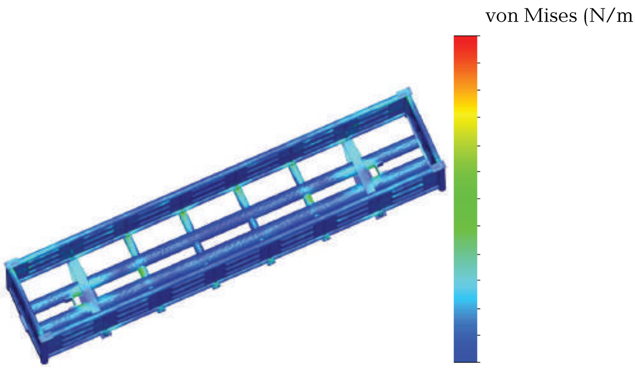
From the studies carried out, it can be concluded that the maximum equivalent stresses in the optimized bearing structure of the gondola wagon arise under III design mode under impact conditions, but they do not exceed the permissible [7–9].

In order to test the strength of the optimized gondola wagon body structure, taking into account two round pipes in the center beam structure, a strength calculation is performed in the CosmosWorks software environment using the finite element method.

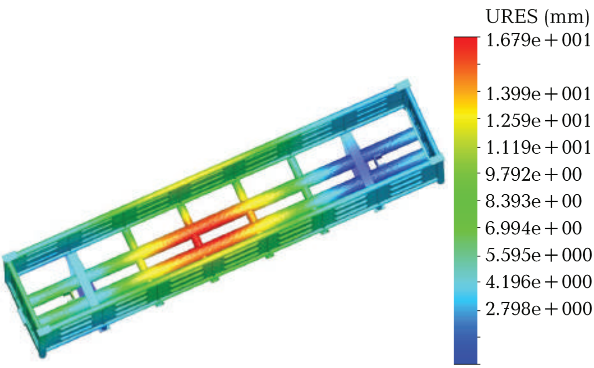
The results of calculating the strength of the bearing structure of the gondola wagon under I design mode (impact) are shown in Fig. 1.17.

The maximum equivalent stresses arise in the zone of interaction of the lower part of the strut with the transverse beam of the frame and are about 335 MPa, the maximum displacements in the nodes of the structure are 16.8 mm, the maximum deformations — $2.52 \cdot 10^{-3}$.

The results of calculating the strength of the bearing structure of the gondola wagon with an optimized design under the conditions of the I design mode (jerk) show that the maximum equivalent stresses arise in the zone of interaction of the lower part of the vertical post with the transverse beam of the gondola wagon frame and are about 295 MPa, the maximum displacements in the structural units are 14.8 mm, the maximum deformations are $5.0 \cdot 10^{-3}$.



a



b

Fig. 1.17 The results of strength analysis of the bearing structure of a model 12-757 gondola wagon under I design mode (impact): *a* — stressful state; *b* — movement in the nodes of the structure

Under I design mode (compression), the maximum equivalent stresses are 284.7 MPa, the maximum displacements in the nodes of the structure are 15.1 mm, and the maximum deformations are.

The results of calculating the strength of the gondola wagon bearing structure under the conditions of IRS of the design mode (impact, compression) allowed to conclude that the maximum equivalent stresses are about 250 MPa, the maximum displacements in the structural units are 15 mm, and the maximum deformations are $5.13 \cdot 10^{-3}$.

The results of calculating the strength of the gondola wagon bearing structure under the conditions of IRS of the design mode (jerk, tension) show that the maximum equivalent stresses are about 260 MPa, the maximum displacements in the structural units are 15 mm, and the maximum deformations are $4.68 \cdot 10^{-3}$.

The studies carried out allow to conclude that the maximum equivalent stresses in the optimized bearing structure of the gondola wagon arise under I design mode under impact conditions, but they do not exceed the permissible ones [7–9].

It is important to note that the mass of the first version of the body structure is lighter than the typical one by 0.6 tons, and the second — by 1 ton, which is 6 % and 9.5 %, respectively.

The second variant of the body structure is more optimal in terms of material consumption, as well as repair and maintenance technology.

The distribution of the maximum equivalent stresses along the length of the center beam of the gondola wagon body is shown in Fig. 1.18.

On the basis of Fig. 1.19, it can be concluded that the maximum stresses arise in the zones of the body bearing on the bogies.

The spatial model of a gondola wagon made of round pipes is shown in Fig. 1.19 [17].

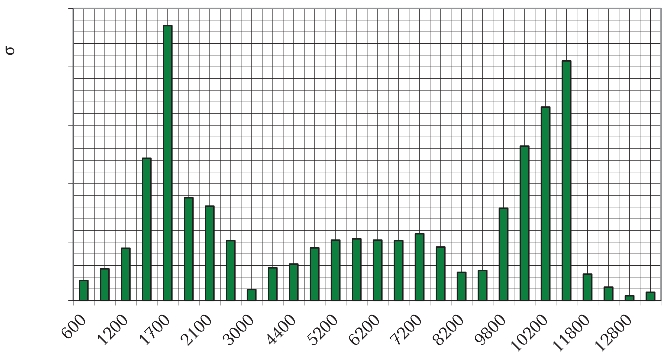


Fig. 1.18 Distribution of equivalent stresses along the length of the gondola wagon (in the lower zone of the pipe)

The developed structure of the gondola wagon body is designed for fatigue strength in the CosmosWorks software environment [12]. The test base is then made up of cycles. The calculation results made it possible to conclude that the fatigue strength of the gondola wagon body bearing structure is provided.

The design service life of the gondola wagon is also determined. When carrying out the calculations, the following input parameters are taken: the average value of the endurance limit of the bearing structure is determined as $0.5\sigma_B$ of material (steel grade 09G2D, O9G2S) and amounted to 245 MPa; base of tests — 10^7 cycles [13]); the time of continuous operation of the bearing structure at $\vartheta_{av} = 33.3$ m/s is 6514.37 s; the effective frequency of dynamic stresses is determined taking into account the parameters of the spring suspension of the model 18-100 bogie and amounted to 2.7 Hz; the permissible safety factor is 2; fatigue curve degree index for the welded structure is taken equal to 4; the amplitude of the equivalent dynamic stresses is determined on the basis of the performed calculations of the stress-strain state of the wagon's bearing structure and amounted to about 50 MPa.

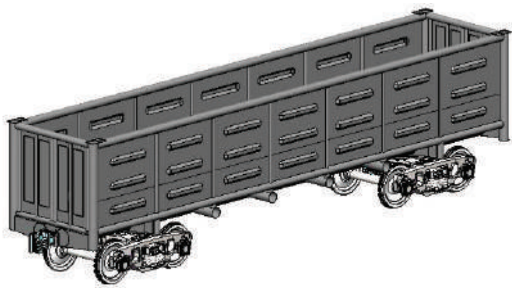


Fig. 1.19 Gondola wagon made of round pipes

On the basis of the calculations, it is established that the design service life of the bearing structure of the improved gondola wagon is more than 32 years, that is, not less than the life cycle of the wagon.

1.4 Optimization of the bearing structure of a covered wagon

To substantiate the feasibility of introducing round pipes as bearing elements of the bodies of covered wagons, a model 11-217 wagon (Fig. 1.20), built (JSC «Altayvagon») is chosen.

To determine the safety reserves of a covered wagon, its spatial model is built in the SolidWorks software environment. Strength calculation is carried out in the CosmosWorks software package [12] by the finite element method.

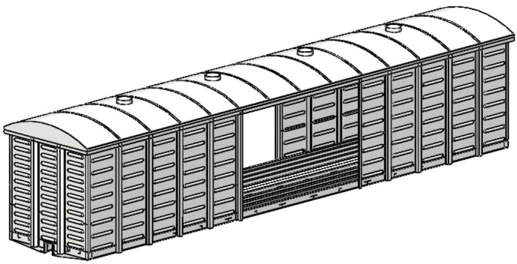


Fig. 1.20 Bearing structure of a covered wagon model 11-217

The numerical values of the forces acting on the covered wagon in operation are shown in Table 1.10. In this case, it is taken into account that the full bearing capacity of the wagon is used.

Table 1.10 Numerical values of the forces acting on the bearing structure of the body of a covered wagon model 11-217 in operation

Force type	I d.m.	III d.m.
Vertical static, kN	810.306	810.306
Vertical dynamic, kN		
Centrifugal, kN		152.66
Wind, kN		13.75
Bulk cargo spacing, kN:		
corner post	5.24	0.988
the first post from the console	4.99	0.941
second to fourth from the console post	4.74	0.895
door post	11.57	0.218

The optimal number of mesh elements is determined using the graphical-analytical method. The number of grid elements is 637.520, nodes — 225092. The maximum size of a grid element is 100.0 mm, the minimum is 20.0 mm, the maximum aspect ratio of elements is 525.19, the percentage of elements with an aspect ratio of less than three is 11.1, more than ten — 46.7. The minimum number of elements in a circle is 22, the ratio of the element size increase is 1.8, the model simplification factor in the areas of fillet placement and holes is 0.4.

The calculation results are shown in Table 1.11.

The results of the calculation made it possible to conclude that the maximum equivalent stresses in the constituent elements of the frame are much less than the permissible ones and have a significant margin of safety. Therefore, in order to reduce the material consumption of the bearing structure of a covered wagon, it is necessary to optimize it to ensure rational safety margins by using their surplus.

Table 1.11 Results of strength analysis of the bearing structure of a covered wagon

Strength index	Load mode				
	hit	compression	jerk-stretch	hit-compression	jerk-stretch
Tension, MPa	330	310	316.6	269.5	270
Displacement in knots, mm	11.0	11.0	10.6	9.44	9.2
Deformations	$4.04 \cdot 10^{-3}$	$4.04 \cdot 10^{-3}$	$4.05 \cdot 10^{-3}$	$5.05 \cdot 10^{-3}$	$4.3 \cdot 10^{-3}$

The results of determining the optimal parameters of the cross-sections of the elements of the bearing structure made of round pipes of the model 11-217 covered wagon are shown in Table 1.12.

Taking into account the data given in Table 1.12, a spatial model of the bearing structure of a covered wagon with the optimal parameters of the structural elements is built (Fig. 1.21).

In order to study the strength of the body of a covered wagon with an improved design, a calculation is made using the finite element method. A computer model of the strength of the bearing structure of a covered wagon in the conditions of the design mode is shown in Fig. 1.22.

The limitations of the model are the absence of a difference in the levels of the covered wagon couplers interacting with each other.

The fixing of the model is carried out at center plates and side beams of the pivot beams of the bearing body structure.

The number of grid elements is 1614576, nodes — 553114. The maximum size of a grid element is 80.0 mm, the minimum is 16.0 mm, the maximum aspect ratio of elements is 1523.8, the percentage of elements with an aspect ratio of less than three is 31.9, more than ten — 20.6. The minimum number of elements in a circle is 22, the ratio of increasing the element size is 1.8, the simplification factor of the model in the rounding areas and holes is 0.4.

The results of the strength calculation of the bearing structure of the covered wagon body under the I design mode (impact) are shown in Fig. 1.23.

In this case, the maximum equivalent stresses arise in the lower zone of interaction of the pivot beam with the center beam and are about 332 MPa, the maximum displacements in the structural nodes are recorded in the middle part of the longitudinal frame beams and are 10.4 mm, the maximum deformations are $2.37 \cdot 10^{-3}$.

Table 1.12 Determination of the optimal parameters of the sections of the elements of the supporting structure of the body of a covered wagon model 11-217

Frame element	Mass 1 m, kg	Length, m	n	σ_{eq}, MPa	I_x, cm^4	I_y, cm^4	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3	Optimal pipe parameters			Mass of 1 m round pipe, kg
											W, cm^3	D, mm	S, mm	
Center beam	125.67	13.87	1.05	330.0	6144.75	57796.39	202.13	3634.99	192.5	3461.9	3494.94	530	17.5	221.18
Side beam	18.4	13.87	6.8	51.0	89.98	1226.47	23.68	122.65	3.5	18.04	18.59	76.0	5.0	8.75
Intermediate crossbeam	9.75	2.79	6.9	50.0	83.26	211.04	20.82	42.21	3.02	6.12	6.53	57.0	3.0	4.0
Longitudinal beam	9.75	10.0	8.3	41.6	83.26	211.04	20.82	42.21	2.51	5.1	6.53	57.0	3.0	4.0
Main crossbeam	28.53	2.79	6.5	52.8	649.89	1248.88	56.025	249.8	8.62	38.43	39.83	108.0	5.0	12.7
Thrust	12.3	1.82	7.8	44.3	52.45	486.66	18.09	69.52	2.32	8.91	9.38	63.5	3.5	5.18
Short longitudinal console beam	9.75	1.265	12.1	28.5	83.26	211.04	20.82	42.1	1.72	3.5	.53	57.0	3.0	4.0
Long longitudinal beam	9.75	1.885	4.7	73.0	83.26	211.04	20.82	42.1	4.43	8.96	9.38	63.5	3.5	5.18
End beam	16.24	2.79	2.03	170.0	188.2	1003.43	27.28	100.34	13.44	49.43	51.23	127.0	4.5	13.59
body post	19.52	2.737	6.86	50.28	783.14	1198.42	68.1	191.75	9.93	27.95	28.38	89.0	5.5	11.33
door post	17.91	2.737	7.03	49.1	46.19	395.34	9.24	52.7	1.31	7.5	7.7	60.0	3.2	4.48
corner post	7.21	2.737	29.2	11.83	23.17	91.45	5.8	22.86	0.2	0.8	6.53	57.0	3.0	4.0
End wall post	19.52	2.737	4.6	75.0	783.14	1198.42	68.1	191.75	14.8	41.68	44.7	114.0	5.0	13.44
Roof arch	3.6	19.8	86.25	4.0	12.7	28.3	5.08	9.43	0.06	0.1	6.53	57.0	3.0	4.0

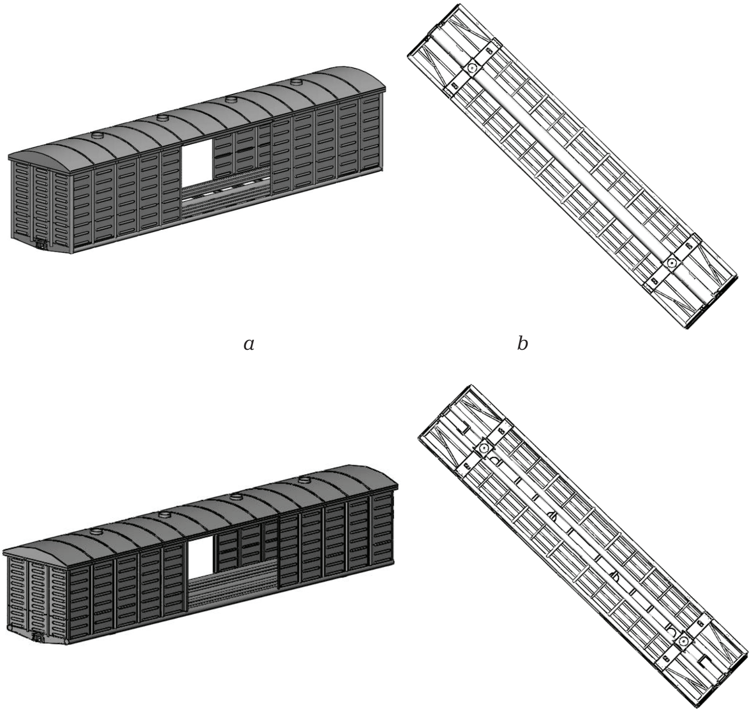
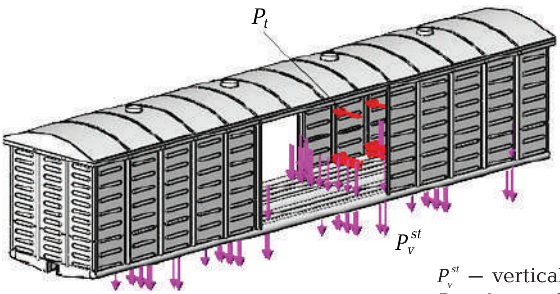


Fig. 1.21 Spatial model of the bearing structure of the covered wagon body: *a* – center beam consists of one pipe; *b* – center beam consists of one pipe (bottom view); *c* – center beam consists of two pipes; *d* – center beam consists of two pipes (bottom view)



P_v^{st} – vertical static force;
P_t – thrust effort of the bulk cargo

Fig. 1.22 Computer model of the strength of the covered wagon body with an improved design under the I design mode

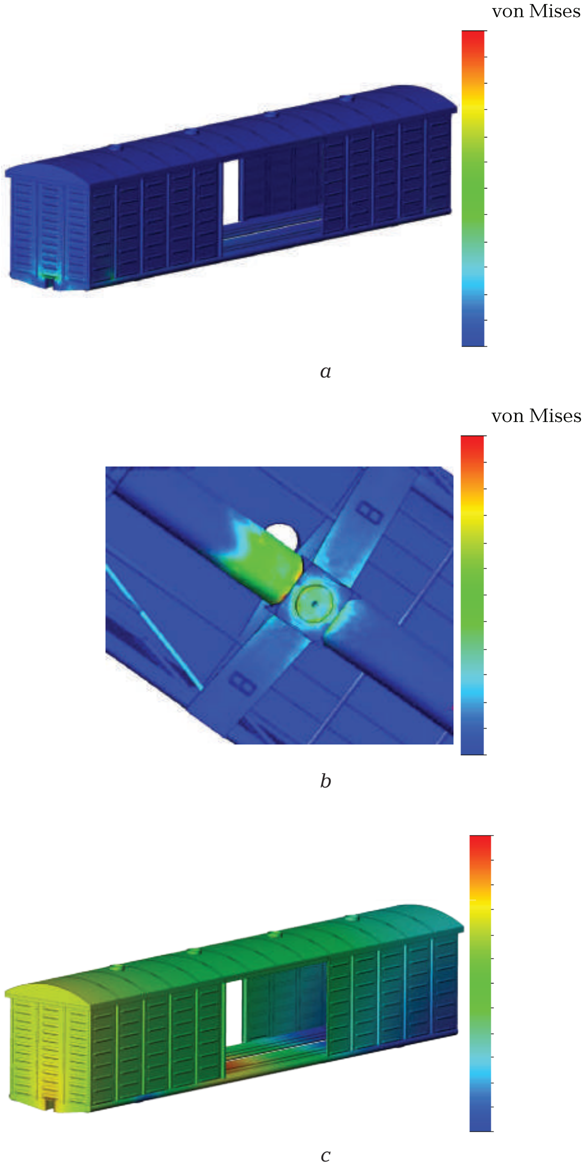


Fig. 1.23 The results of the strength calculation of the body of a covered wagon with an improved design under the conditions of I design mode (impact): *a* — stress state (side view); *b* — stress state (bottom view); *c* — displacement in nodes

The distribution of equivalent stresses along the length of the center beam is shown in Fig. 1.24. In this case, the value of the equivalent stresses is recorded with the lower part of the center beam pipe.

The results of calculating the strength of the bearing structure of the covered wagon body under the I design mode (tension-jerk) show that the maximum equivalent stresses in this case arise in the lower zone of the center beam and are about 300 MPa, the maximum displacements in the structural nodes are 8.9 mm, the maximum deformations are $3.43 \cdot 10^{-3}$.

Under the I design mode (compression), the maximum equivalent stresses arise in the lower zone of the center beam and are about 310 MPa, the maximum displacements in the structural nodes are 10.1 mm, and the maximum deformations are $2.03 \cdot 10^{-3}$.

The results of calculating the strength of the bearing structure of the gondola wagon body under the III design mode (shock-compression) made it possible to conclude that the maximum equivalent stresses in this case are about 290 MPa, the maximum displacements in the nodes of the structure are 7.2 mm, and the maximum deformations are $6.8 \cdot 10^{-4}$.

The results of calculating the strength of the bearing structure of the covered wagon body under the design conditions (tension-jerk) show that the maximum equivalent stresses in this case are about 270 MPa, the maximum displacements in the nodes of the structure are 5.6 mm, and the maximum deformations are $1.37 \cdot 10^{-3}$.

Also, the strength of the bearing structure of a covered wagon with a center beam of two round pipes is determined. The calculation results confirmed the feasibility of the adopted technical solutions.

A spatial model of a covered wagon made of round pipes is shown in Fig. 1.25 [18].

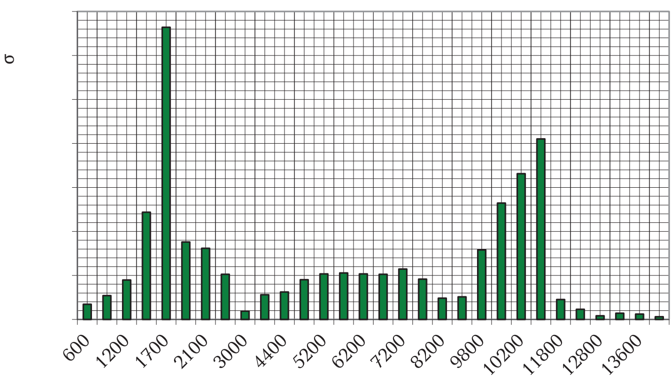


Fig. 1.24 Distribution of equivalent stresses along the length of the center beam under the I design mode (impact)

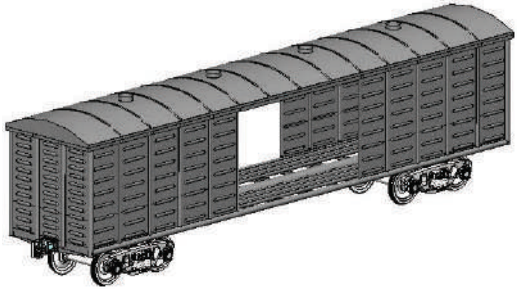


Fig. 1.25 Covered wagon made of round pipes

The proposed measures make it possible to reduce the weight of the bearing structure of the covered wagon body in comparison with the prototype wagon by 4 % (one pipe) and 4.84 % (two pipes).

The bearing structure of the covered wagon body has been designed for fatigue strength in the CosmosWorks software environment. The test base is then made up of cycles. The calculation results made it possible to conclude that the fatigue strength of the bearing structure of the covered wagon body is ensured.

The design service life of a covered wagon made of round pipes has been determined.

When carrying out the calculations, the following input parameters are taken: the average value of the endurance limit of the bearing structure is determined as $0.5\sigma_B$ of the material (steel grade 09G2D, O9G2S) and amounted to 245 MPa; test base — 10^7 cycles [13]; the time of continuous operation of the bearing structure at $\vartheta_{av} = 33.3$ m/s is 6514.37 s; the effective frequency of dynamic stresses is determined taking into account the parameters of the spring suspension of the model 18-100 bogie and amounted to 2.7 Hz; the permissible safety factor is 2; the index of the degree of the fatigue curve for the welded structure is taken equal to 4; the amplitude of the equivalent dynamic stresses is determined on the basis of the performed calculations of the stress-strain state of the wagon's bearing structure and amounted to 52.82 MPa (one pipe), 50.6 MPa (two pipes).

Based on the calculations, it is established that the design service life of the improved bearing structure of the covered wagon is more than 32 years, that is, not less than the life cycle of the wagon.

1.5 Optimization of the bearing structure of the hopper wagon

To study the possibility of optimizing the bearing structure of a hopper wagon, its spatial model is created in SolidWorks software (Fig. 1.26).

The hopper wagon model 20-9749, built by the State Enterprise «Ukrspetsvagon» (Ukraine), is selected for the prototype wagon.

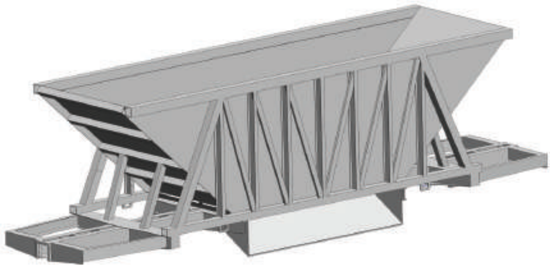


Fig. 1.26 Spatial geometric computer model of the bearing structure of a hopper wagon model 20-9749

To determine the strength reserves of a covered wagon, a strength calculation is carried out in the CosmosWorks software package [13] by the finite element method.

The numerical values of the forces acting on the hopper wagon in operation are shown in Table 1.13. In this case, it is taken into account that the full bearing capacity of the wagon is used.

Table 1.13 Numerical values of forces acting on the model 20-9749 hopper wagon in operation

Force type	I d.m.	III d.m.
Vertical static, kN	829.926	829.926
Vertical dynamic, kN		99.59
Centrifugal, kN		156.35
Framed, kN		202.94
Wind, kN		7.95

Strength analysis is carried out using the finite element method in the CosmosWorks software environment. The optimal number of mesh elements of the finite element model is determined using the graphical-analytical method. The number of grid elements is 376670, nodes — 126221. The maximum size of a grid element is 60 mm, the minimum is 12 mm, the maximum aspect ratio of elements is 1298.6, the percentage of elements with an aspect ratio of less than three is 7.43, more than ten is 32.5.

The fixing of the model is carried out at center plates and side beams of the pivot beams of the bearing structure of the hopper wagon.

The calculation results are shown in Table 1.14.

Table 1.14 Results of strength analysis of the bearing structure of a hopper wagon

Strength index	Load mode				
	hit	compression	jerk-stretch	hit-compression	jerk-stretch
Tension, MPa	220	186.6	175.4	196.8	169.5
Displacement in knots, mm	4.5	3.8	3.7	4.1	3.6
Deformations	$3.6 \cdot 10^{-5}$	$3.4 \cdot 10^{-5}$	$3.3 \cdot 10^{-5}$	$3.5 \cdot 10^{-5}$	$3.2 \cdot 10^{-5}$

The obtained results allow to conclude that the maximum equivalent stresses in the bearing structure of the hopper wagon arise in the design mode under impact conditions. At the same time, in the constituent elements of the frame, the maximum equivalent stresses are much less than the permissible ones and have a significant margin of safety. Therefore, in order to reduce the material consumption of the bearing structure of the hopper wagon, it is necessary to optimize it to ensure rational safety margins by using their surpluses.

When choosing the optimal parameters for the pipes of the centerline and main longitudinal beams of the frame, it is taken into account that the longitudinal and middle parts of the beams must have the same wall thickness with the manufacturability of the structure.

The results of determining the optimal parameters of the cross-sections of the elements of the bearing structure made of round pipes of the model 20-9749 hopper wagon are shown in Table 1.15.

Taking into account the data given in Table 1.15, a spatial model of the bearing structure of a hopper wagon with the optimal parameters of the structural elements is built (Fig. 1.27).

To determine the strength of the bearing structure of the hopper wagon made from round pipes, a calculation is performed using the finite element method. The design diagram of the bearing structure of a hopper wagon during a shunting collision is shown in Fig. 1.28.

The number of grid elements is determined using the graphical analytical method and amounted to 2480364, nodes — 808646. The maximum size of a grid element is 20 mm, the minimum is 4 mm, the maximum aspect ratio of elements is 8, the percentage of elements with an aspect ratio of less than three is 37.8, more than ten — 1.51. The design diagram of the bearing structure of the hopper wagon is shown in Fig. 1.27. The designations of the loads adopted in the design scheme are identical to those considered when calculating a typical bearing structure of a hopper wagon.

It is taken into account that the bearing structure of the wagon is subject to a vertical static load P_v^{st} due to the weight of the cargo. Also, the expansion forces P_e from the bulk cargo act on the body structure. A shock load P_{sh} acts on the vertical surface of the backgauge, the numerical value of which in accordance with regulatory documents is 3.5 MN.

Table 1.15 Determination of the optimal parameters of the sections of the elements of the bearing structure of the hopper wagon body model 20-9749

Frame element	Mass 1 m, kg	Length, m	n	σ_{eq}, MPa	I_x, cm^4	I_y, cm^4	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3	Optimal pipe parameters			Mass of 1 m round pipe, kg
											W, cm^3	D, mm	S, mm	
Center 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
Intermediate cross 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
top fitting	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 fitting	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
Post	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
Thrust	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 thrust	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
End channel beam	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 knee	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

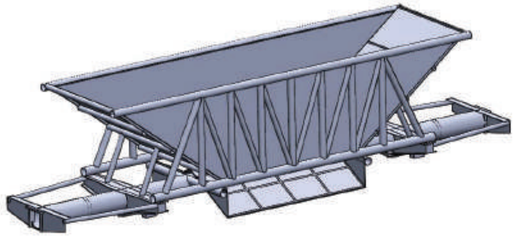


Fig. 1.27 Spatial model of the bearing structure of the covered wagon body

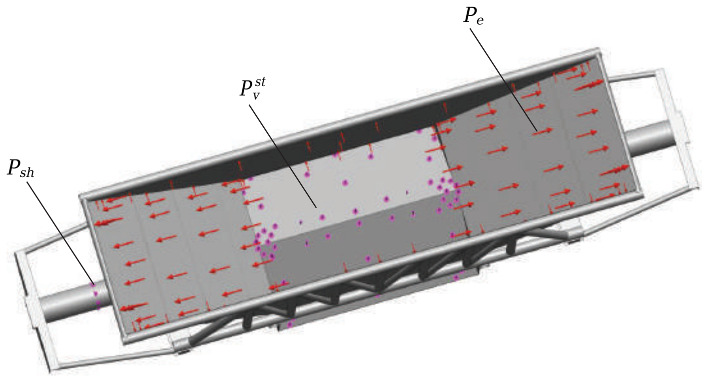


Fig. 1.28 Design diagram of the bearing structure of a hopper wagon during a shunting collision

The model is fastened in the areas where the body rests on the chassis. Construction material — 09G2S steel. The calculation results are shown in Fig. 1.29. The distribution of stresses along the length of the center beam of a hopper wagon during a shunting collision is shown in Fig. 1.30.

At the same time, the maximum equivalent stresses in the bearing structure of the hopper wagon are about 270 MPa and are concentrated in the zone of interaction of the center beam with the pivot, but do not exceed the permissible ones [7–9].

The maximum displacements in the bearing structure of the hopper wagon occur in the unloading bins and are about 5.2 mm. The maximum deformations in the bearing structure of the hopper wagon are $5.7 \cdot 10^{-5}$.

The results of calculating the strength of the bearing structure of the body of hopper wagons under the I design mode (tension-jerk) show that the maximum equivalent stresses are about 215.3 MPa, the maximum displacements in the structural nodes are 3.7 mm, and the maximum deformations are $6.1 \cdot 10^{-5}$.

In the design mode (compression), the maximum equivalent stresses are about 234.2 MPa, the maximum displacements in the structural nodes are 4.4 mm, and the maximum deformations are $5.9 \cdot 10^{-5}$.

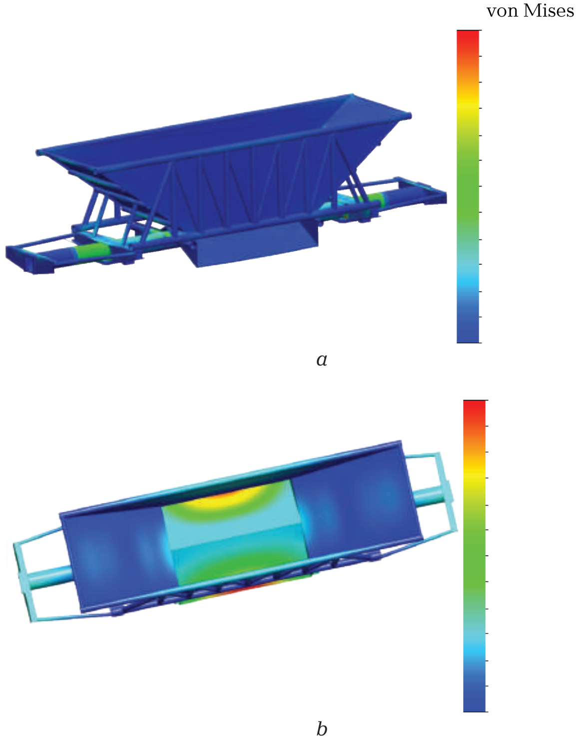


Fig. 1.29 The results of the strength calculation of the hopper wagon body of an improved design in the conditions of the design mode (impact): *a* — stressful state; *b* — movements in nodes

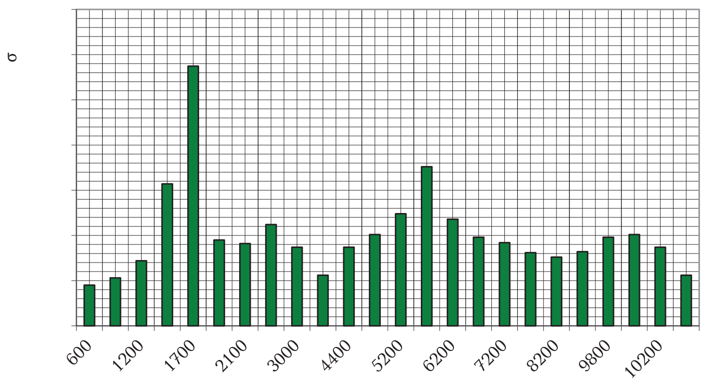


Fig. 1.30 Distribution of stresses along the length of the center beam of the hopper wagon

The results of calculating the strength of the bearing structure of the hopper wagon body in the design mode (shock-compression) make it possible to conclude that the maximum equivalent stresses are about 246.8 MPa, the maximum displacements in the structural nodes are 4.8 mm, the maximum deformations are $5.8 \cdot 10^{-5}$.

The results of calculating the strength of the bearing structure of the hopper wagon body in the design mode (tension-jerk) show that the maximum equivalent stresses are about 203.5 MPa, the maximum displacements in the structural nodes are 3.2 mm, and the maximum deformations are $6.3 \cdot 10^{-5}$.

The proposed measures make it possible to reduce the weight of the bearing structure of the hopper wagon body in comparison with the prototype wagon by almost 5 %.

The bearing structure of the hopper wagon body is designed for fatigue strength in the CosmosWorks software environment. The test base is then made up of cycles. The results of the calculation make it possible to conclude that the fatigue strength of the bearing structure of the hopper wagon body is ensured.

A spatial model of a hopper wagon made of round pipes is shown in Fig. 1.31 [19].

Fig. 1.31 Hopper wagon made of round pipes

The design service life of a hopper wagon made of round pipes has been determined. When carrying out the calculations, the following input parameters are taken: the average value of the endurance limit of the bearing structure is determined as $0.5\sigma_B$ of material (steel grade 09G2D, O9G2S) and amounted to 245 MPa; test base — 10^7 cycles [13]; the time of continuous operation of the bearing structure at $\vartheta_{av} = 33.3$ m/s is 6514.37 s; the effective frequency of dynamic stresses is determined taking into account the parameters of the spring suspension of the model 18-100 bogie and amounted to 2.7 Hz; the permissible safety factor is 2; the index of the degree of the fatigue curve for the welded structure is taken equal to 4; the amplitude of the equivalent dynamic stresses is determined on the basis of the performed calculations of the stress-strain state of the wagon's bearing structure and amounted to 57.3 MPa.

Based on the calculations, it is established that the design service life of the improved bearing structure of the hopper wagon is more than 32 years, that is, not less than the life cycle of the wagon.

1.6 Optimization of the bearing structure of the tank container

To reduce the material consumption of the frame of the tank container, it is proposed to replace the elements of the bearing structure made of square pipes with circular pipes. A tank container model TK25, built by JSC «Zarechensk Chemical Engineering Plant» (Fig. 1.32), is chosen as the base one. The specified tank container has a standard size according to ISO – 1CC and is intended for transportation of: FRM, gasoline, diesel fuel, engine oil, cooling mixture, oil solvent, nefras, foaming agent.

The limitations of this optimization model are:

1. Geometric dimensions of the fitting.
2. Design loads must be less than the permissible ones:

$$\sigma_{eq} < \sigma_y, \tag{1.18}$$

where σ_{eq} – equivalent stresses in the structure, MPa; σ_y – material yield stress, MPa.

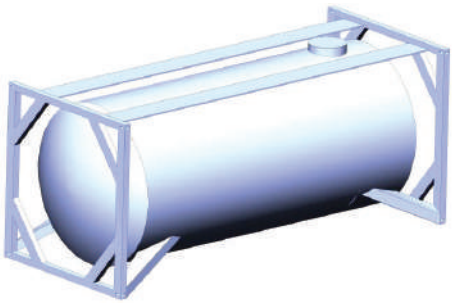


Fig. 1.32 Tank container of 1CC standard size

The diameter of the vertical pipe is selected based on the geometric features of the fitting and is equal to $D = 152$ mm. The variation occurs based on the change in the wall thickness S of the pipe. The pipe wall thickness varies from 3 to 5.5 mm. The inner diameter of the pipes ranges from 146.5 to 149 mm. The calculated main dimensions, static characteristics and mass of pipes with a circular cross-section are given in Table 1.16.

In order to optimize the design of the tank container, the spatial purpose of the model is built using the SolidWorks software, the strength calculation of which is carried out by the finite element method. The results of the research are shown in Table 1.17.

Table 1.16 Dimensions, static characteristics and weight of 1m of pipes

Pipe dimensions, mm		Cross-sectional area F, cm^2	Static characteristics for the x and y axes		Weight of 1 m pipe M, kg
Outer diameter D	Wall thickness S		Section moment of inertia, I_{xy}, cm^4	Section resistance moment W_{xy}, cm^3	
152	3.0	14.04	389.87	51.30	11.02
	3.2	14.96	414.21	54.50	11.74
	3.5	16.33	450.35	59.26	12.82
	3.8	17.69	486.04	63.95	13.89
	4.0	18.60	509.59	67.05	14.60
	4.5	20.85	567.61	74.69	16.37
	5.0	23.09	624.43	82.16	18.13
	5.5	25.31	680.06	89.48	19.87

Table 1.17 Results of studies on the strength of spatial models

Outer diameter, mm	Wall thickness, mm	Stress value	Mass, kg	Deformations
152	3.0	318.32	1055.61	$1.701 \cdot 10^{-3}$
	3.2	316.466	1061.69	$1.726 \cdot 10^{-3}$
	3.5	315.482	1070.77	$1.758 \cdot 10^{-3}$
	3.8	314.525	1057.39	$1.790 \cdot 10^{-3}$
	4.0	313.639	1063.38	$1.858 \cdot 10^{-3}$
	4.5	311.555	1078.33	$1.927 \cdot 10^{-3}$
	5.0	311.664	1093.17	$1.948 \cdot 10^{-3}$
	5.5	310.8	1093.17	$1.983 \cdot 10^{-3}$
	5.5[1)]	316.374	1075.17	$1.868 \cdot 10^{-3}$

[1)] a pipe of circular cross-section is taken along the entire height of the vertical post of the tank container

In order to reduce the material consumption of the bearing structure of the tank container, it is proposed to improve its frame by replacing square pipes, which are used in a typical design, with round pipes while ensuring the conditions of strength and operational reliability.

When choosing the diameter of round pipes from which it is proposed to manufacture the frame of the tank container, the geometric dimensions of the fittings are taken into account [20]. Therefore, pipes with an outer diameter of 152 mm are chosen as the basic element [21].

In order to obtain an optimal frame structure (a structure with optimal geometric parameters of the cross-sections of the pipes being introduced), optimization studies are carried out in the following sequence: it is found that optimization will be carried out according to the criterion of minimum material consumption (m) while ensuring strength conditions (not exceeding the permissible stress values (σ) and deformations (l)) based on the design features, the boundaries of variation of variable parameters – the outer diameter of the pipe (D) and the wall thickness (S); it is found that mathematical models of changes in indicators (m, σ, l) are described by two-factor generalized mathematical models [22], to determine which nine experiments are carried out on the basis of the corresponding spatial computer models; certain mathematical models on the basis of which an auxiliary graph is built and the optimal geometric parameters of pipes are clarified.

At the initial stage, it is decided to use a round pipe along the entire height of the vertical post of the tank container (Fig. 1.33). The results of the strength calculation show that the stress in the zone of interaction between the paw and the vertical post exceeds the permissible values for the steel grade of the metal structure of the tank container. Therefore, when building a spatial computer model of a tank container, it is taken into account that a vertical post of circular cross-section is placed on a special superstructure (Fig. 1.34). That is, the paw pinch assembly with a vertical post remains unchanged. The diameter of the pipes of the slopes of the end frames, as well as its cross beams, is selected based on the geometric parameters of the fitting.

Using the obtained values of the indicators (m, σ, l), they are approximated in the form of polynomials of the second degree:

$$\begin{aligned} Y = f(x_{t_1}, x_{t_2}, x_{t_3}, x_{t_a}) = a_0 + a_1 x_{t_1} + a_2 x_{t_2} + a_3 x_{t_3} + a_4 x_{t_a} + \\ + a_{11} x_{t_1}^2 + a_{11} x_{t_1}^2 + a_{22} x_{t_2}^2 + a_{33} x_{t_3}^2 + a_{44} x_{t_a}^2 + a_{12} x_{t_1} x_{t_2} + a_{13} x_{t_1} x_{t_3} + \\ + a_{14} x_{t_1} x_{t_a} + a_{23} x_{t_2} x_{t_3} + a_{24} x_{t_2} x_{t_a} + a_{34} x_{t_3} x_{t_a}, \end{aligned} \tag{1.19}$$

where Y – controlled indicator; x_{t_a} – normalized parameters; a_i – coefficients of the generalized mathematical model, the numerical values of which are determined by solving the system of equations from the obtained data.

When compiling generalized mathematical models (GMM), the technique given in [23] is used. GMM determination and the construction of an auxiliary graph for the determination of the optimal parameters of the pipe frame of the tank container is carried out in the software environment [22].

The resulting generalized mathematical models are as follows:

$$\begin{aligned} m = -(7.013E+04) + (9.243E+02)D + 542.416S - \\ - 3D^2 + 2.048S^2 - 3.6DS; \end{aligned} \tag{1.20}$$

$$s = 22383.3 - 286.18D - 181.77S + 0.928858D^2 +$$
$$+ 1.1853S^2 + 1.1146DS; \qquad (1.21)$$

$$l = -0.4537 + 0.00591D + 0.0038S - 1.916D^2 -$$
$$- (2.3E - 05)S^2 - (2E - 05)DS, \qquad (1.22)$$

where D — the outer diameter of the pipe, mm; S — pipe wall thickness, mm; m — structure weight, kg; l — deformation in the structure; σ — stress in the structure, MPa.

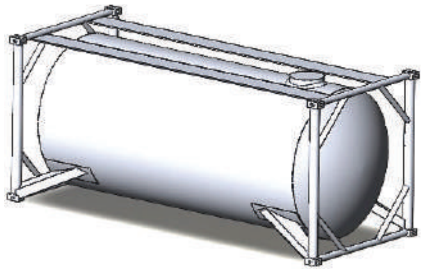


Fig. 1.33 Bearing structure of a tank container using a round pipe along the entire height of the vertical post of a tank container

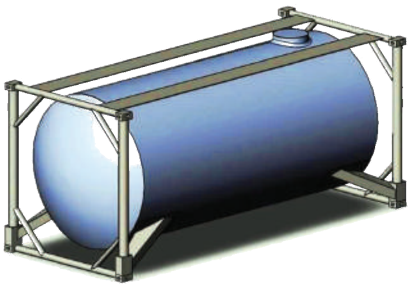


Fig. 1.34 3D model of a tank container with an improved bearing structure

The verification of the adequacy of mathematical models (1.20–1.22) is carried out according to the value of the standard deviations:

$$\sigma = \sqrt{\frac{\sum_{j=1}^{m-k}\left(Y_j - Y_{jp}\right)^2}{m-k}}, \qquad (1.23)$$

where k — the number of coefficients a_i of the generalized mathematical model; m — the number of modes of the mathematical plan.

The performed calculations confirmed the efficiency of the obtained generalized mathematical models. Moreover, the value of the standard deviations does not exceed 3 %.

An auxiliary graph for determining the optimal values of the parameters of the pipe section is shown in Fig. 1.35.

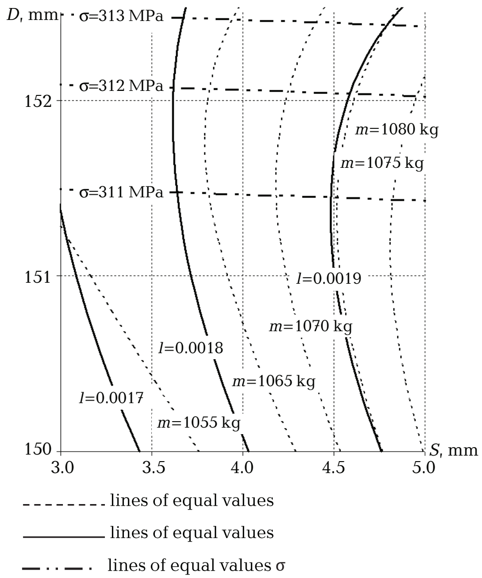


Fig. 1.35 Auxiliary graph for determining the optimal values of the pipe section indicators

Based on the studies carried out, it is concluded that the optimal pipe is with an outer diameter of 152 mm and a wall thickness of 3 mm. Additionally, the correctness of the calculations is confirmed by the results of solving the same optimization problem by the analytical method.

After determining the optimal values of the geometric parameters of the pipes being introduced into the developed spatial computer geometric model, a strength calculation is carried out (Fig. 1.36). The maximum equivalent stresses are 312 MPa. The results of strength calculations have shown the correctness of the studies. The developed design of the tank container is designed for all types of loads specified in [24].

When examining the strength of a tank container, taking into account the loads acting on it during stacking, the maximum allowable number of tank containers in a stack for this type is taken into account, it is four tiers (Fig. 1.37). In this case, the maximum equivalent stresses do not exceed the permissible ones, and are about 181 MPa [25, 26].

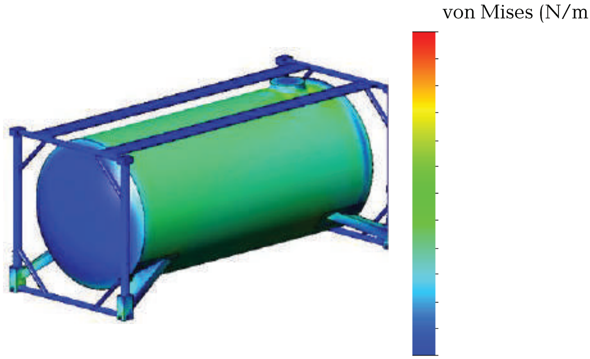


Fig. 1.36 Diagram of statistical nodal stresses of the improved design of the tank container at collision in the loaded state

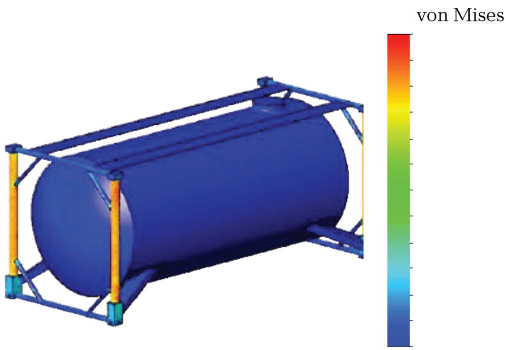


Fig. 1.37 Diagram of statistical nodal loads when stacked in 4 tiers

The results of strength calculations make it possible to conclude that the stress in the bearing structure of the tank container does not exceed the permissible values. Taking into account the measures to improve the bearing structure of the tank container, it becomes possible to reduce its weight by 40 % in comparison with the prototype design, which will make it possible to achieve a significant economic effect [27, 28].

Conclusions to chapter 1

1. The general procedure for optimizing the bearing structures of wagons by using round pipes as the main bearing elements is considered. This procedure can be applied to the optimization of other vehicles and engineering structures.

2. The use of round pipes as bearing elements of the main types of wagons, as well as tank containers is substantiated.

It is established that the implementation of round pipes into the bearing structures of wagons makes it possible to reduce their tare compared to prototype wagons, a flat wagon — 4 %, a gondola wagon (single-pipe center beam) — 6 %, a gondola wagon (double-pipe center beam) — 9.5 %, covered wagon (single-pipe center beam) — 4 %, covered wagon (double-pipe center beam) — 4.84 %, hopper wagon — 5 %.

3. The results of calculating the strength of the bearing structures of the main types of wagons, taking into account the use of round pipes as bearing elements, make it possible to conclude that the maximum equivalent stresses for all design load schemes do not exceed the permissible ones. The determination of the fatigue strength of the bearing structures of wagons made of round pipes is carried out on the basis of tests 10^7. The calculation results confirm the expediency of the decisions taken. Calculation of the design service life of the bearing structures of the wagons made of round pipes show that the standard service life is at least 32 years, that is, at least for the life cycle of the wagon.

4. The substantiation of the implementation of round pipes as bearing elements of the frame of the tank container is carried out. The calculations are carried out with respect to a 1CC tank container. It is found that taking into account the measures to improve the bearing structure of the tank container, it becomes possible to reduce its weight by 40 % compared to the design of the prototype.

So, the implementation of round pipes as bearing elements of vehicles is a reasonable decision and will significantly reduce the cost of their manufacture in the conditions of wagon enterprises.

Chapter 2
Implementation of a fitting device concept in the bearing structures of wagons made of round pipes

2.1 Determination of the dynamic loading of a wagon during a shunting collision

To reduce the dynamic loads acting on the bearing structures of the bodies of railway wagons, it is proposed to eliminate the automatic coupling devices from the structures and transfer their functions to absorb energy, arises from the action of operational loads on the vertebral beam, as well as the upper and lower fittings of the side walls, which are proposed to be performed from round pipes and fill with material with damping and anti-corrosion properties, which will reduce the material consumption of the wagon and, accordingly, increase its bearing capacity and the loading volume of the body, as well as extend its maintenance-free service life.

To determine the dynamic loads acting on the wagon bodies during shunting collisions, as in the case of the highest load on their bearing structures in operation, the mathematical model given in [29] is used. This model is designed to determine accelerations as a component of the dynamic load acting on a flat wagon with tank containers during a shunting collision. Therefore, it has been modified to determine accelerations as a component of the dynamic load, which is perceived by the wagon when the longitudinal impact force is applied.

$$M'_W \cdot \ddot{x}_W + M' \cdot \ddot{\varphi}_W = S_a; \tag{2.1}$$

$$I_W \cdot \ddot{\varphi}_W + M' \cdot \ddot{x}_W - g \cdot \varphi_W \cdot M' = l \cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1 - \operatorname{sign}\dot{\Delta}_2\right) + l\left(C_1 - C_2\right); \tag{2.2}$$

$$M_W \cdot \ddot{z}_W = C_1 + C_2 - F_{FR}\left(\operatorname{sign}\dot{\Delta}_1 - \operatorname{sign}\dot{\Delta}_2\right), \tag{2.3}$$

where

$$M'_W = M_W + 2 \cdot m_C + \frac{n \cdot I_{WS}}{r^2}; \quad M' = M_W \cdot h; \quad C_1 = k_1 \cdot \Delta_1; \quad C_2 = k_2 \cdot \Delta_2; \tag{2.4}$$

$$\Delta_1 = z_W - l\cdot\varphi_W; \quad \Delta_2 = z_W + l\cdot\varphi_W, \tag{2.5}$$

where M_W — mass of the wagon bearing structure; I_W — moment of inertia of the wagon relative to the longitudinal axis; S_a — value of the longitudinal force of impact into the automatic coupler; m_c — cart mass; I_{WS} — moment of inertia of the wheelset; r — radius of the mid-worn wheel; n — the number of bogie axles; l — half of the wagon base; F_{fr} —absolute value of the dry friction force in the spring set; k_1, k_2 — stiffness of the spring suspension of the wagon bogies; x_W, φ_W, z_W — coordinates corresponding to the longitudinal, angular around the transverse axis and vertical movement of the wagon, respectively.

Differential equations are solved in the MathCad software environment [30, 31]. Moreover, they are reduced to the Cauchy normal form, and then integrated using the Runge-Kutta method.

Initial displacements and speeds caused by being equal to zero. The input parameters of the mathematical model are the technical characteristics of the wagon body, the parameters of the spring suspension, as well as the value of the force of the longitudinal impact into the automatic coupling.

As an example, below are the results of studies of the longitudinal dynamics of the body of a gondola wagon model 12-757, as one of the most common types of wagons in operation, taking into account the manufacture of its bearing structure from pipes of circular cross-section. The parameters of the spring suspension, which are taken into account in the calculations, are taken equal to those that are characteristic of typical cargo carts of model 18-100.

The longitudinal impact force acting on the vertical surface of the rear stop of the automatic coupler is taken equal to 3.5 MN [7, 8].

The research results allow to conclude that the acceleration that acts on the bearing structure of the wagon during a shunting collision is about 40 m/s^2 (Fig. 2.1, *a*)).

To determine the accelerations acting on the bearing structure of the wagon body, taking into account the filling of round pipes with an elastomeric material with damping and anti-corrosion properties, the above mathematical model is reduced to the form:

$$M'_W\cdot\ddot{x}_W + M'\cdot\ddot{\varphi}_W = S_a - \beta\cdot\dot{x}_W, \tag{2.6}$$

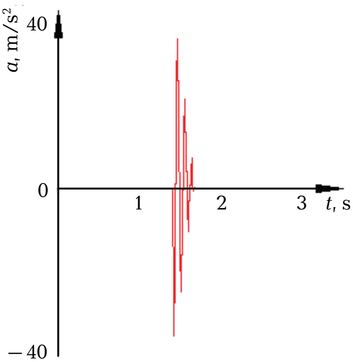
$$I_W\cdot\ddot{\varphi}_W + M'\cdot\ddot{x}_W - g\cdot\varphi_W\cdot M' = l\cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1 - \operatorname{sign}\dot{\Delta}_2\right) + l\left(C_1 - C_2\right), \tag{2.7}$$

$$M_W\cdot\ddot{z}_W = C_1 + C_2 - F_{FR}\left(\operatorname{sign}\dot{\Delta}_1 - \operatorname{sign}\dot{\Delta}_2\right), \tag{2.8}$$

where β — coefficient of viscous resistance of the material with which the elements of the wagon bearing structure are filled.

The study of accelerations acting on the bearing structure of the wagon body in the presence of a substance with viscous properties in its bearing elements led to the conclusion that the effectiveness of this solution is achieved when the viscous resistance coefficient does not exceed 120 kN·s/m [32–34].

The research results, taking into account the value of the coefficient of viscous resistance of the substance 120 kN·s/m, make it possible to conclude that the acceleration that acts on the bearing structure of the wagon during a shunting collision is about 34 m/s^2 (Fig. 2.1, *b*)), and is lower by 10 % of the value of accelerations obtained under a typical scheme of the load perception by the center beam of the wagon.



a *b*

Fig. 2.1 Acceleration acting on a wagon during a shunting collision:
a — typical scheme of load perception by the center beam of a wagon;
b — load is perceived by the center beam, which is filled with a substance with viscous properties

It is important to note that the resulting acceleration value can be reduced by selecting the most optimal parameters of binders used in industry.

In order to implement the proposed scheme for the perception of longitudinal loads by the center beam of the wagon, it is planned to change the design features of automatic couplers, namely, the elimination of harness devices, as well as rear stops and shifting their functions to elements that are less complex in design (Fig. 2.2), which will significantly reduce the manufacturing cost and repair of such gondola wagons [35].

The automatic coupler includes a typical case 1 CA-3, which interacts with an intermediate adapter 2, consisting of a thrust part on which a base plate of a typical design is located. The thrust part of the adapter is connected through the rod to the piston in which there are two throttle valves — inlet and outlet. The binder is placed on the left and right sides of the piston. To create the pressure of the binder during the movement of the piston during the perception of the shock load, a bottom 5 is provided in the center beam 4. To limit the movements of the adapter during «jerk-stretching», a limiter 6 is provided.

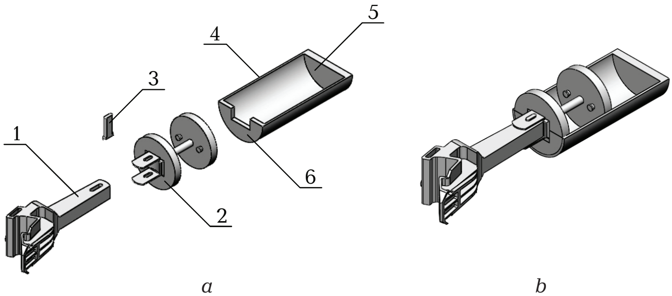


Fig. 2.2 Design features of an automatic coupler device based on the introduction of viscous substances as shock absorbers of longitudinal forces in the bearing structure of the wagon body: *a* — assembly with the separation of elements; *b* — assembly (general view); 1 — automatic coupler body; 2 — adapter; 3 — wedge; 4 — center beam made of round pipe; 5 — bottom

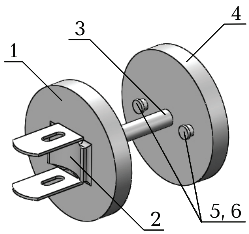


Fig. 2.3 Design features of the adapter: 1 — thrust part; 2 — thrust plate; 3 — stock; 4 — piston; 5, 6 — throttle valves

The design features of the adapter are shown in Fig. 2.3. An auto-coupling device based on the introduction of viscous substances as shock absorbers of longitudinal forces in the bearing structure of the wagon body works as follows (Fig. 2.4). The load through the coupler body 1 is transferred to the thrust part of the adapter 2 through the shank. Under the action of the load, the adapter moves in the longitudinal direction and compresses the binder placed in the cavity between the piston 3 and the bottom 4. This causes it to flow through the throttle valve 5 into the cavity between the thrust part of the adapter and the piston. With the reverse movement of the piston, the overflow process is carried out through another throttle valve 6.

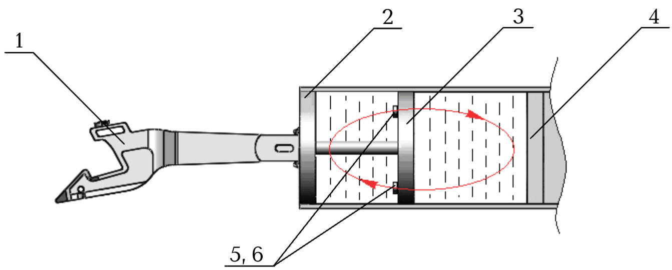


Fig. 2.4 Operation process of an automatic coupler device based on the introduction of viscous substances as shock absorbers of longitudinal forces in the bearing structure of the wagon body

The energy generated in this process is dissipated into the environment. That is, the process of the binder is similar to the work of a hydraulic vibration damper.

In order to ensure the efficiency of the proposed wagon harness system, it is necessary to ensure its tightness (no possible leakage of the binder from the center beam).

2.2 Computer modeling of dynamic loading of a wagon taking into account the use of the concept of a fitting

For theoretical verification of the obtained results of mathematical modeling and study of the distribution of acceleration fields in the bearing structure of railway wagons made of circular pipes, they built spatial computer models in the SolidWorks software environment, followed by modeling the dynamics in CosmosWorks.

Determination of accelerations is carried out by the finite element method.

Let's consider modeling the process of dynamic loading of the bearing structure of a railway wagon using the example of a gondola wagon.

The strength model of the perspective bearing structure of the gondola wagon body is shown in Fig. 2.5, *a*).

In this case, it is taken into account that the full bearing capacity of the wagon is used. Coal is accepted as a bulk cargo, since this type of cargo is the most common for transportation on the territory of Ukraine.The calculation of the expansion forces of the bulk cargo is carried out according to the methodology given in [10].

The longitudinal impact force is applied to the vertical surface of the adapter thrust plate (Fig. 2.5, *b*)), and fastening is carried out on the opposite side in order to simulate the maximum compression of the bearing structure.

Also, the model is fixed in the zones of its bearing on the running gear — center plates and side bars.

To simulate the properties of the binder, a spring-damper link from the software mounts library is used, which is installed between the piston and the bottom.

The viscous resistance coefficient of the binder is taken equal to 120 kN·s/m, and the spring stiffness is taken equal to about 0, since there is no elastic connection in the proposed scheme for perceiving the longitudinal load.

Based on the calculations, it is determined that the displacement of the adapter under the action of the longitudinal impact force kN and the impact speed of 0.03 m/s will be 875 mm.

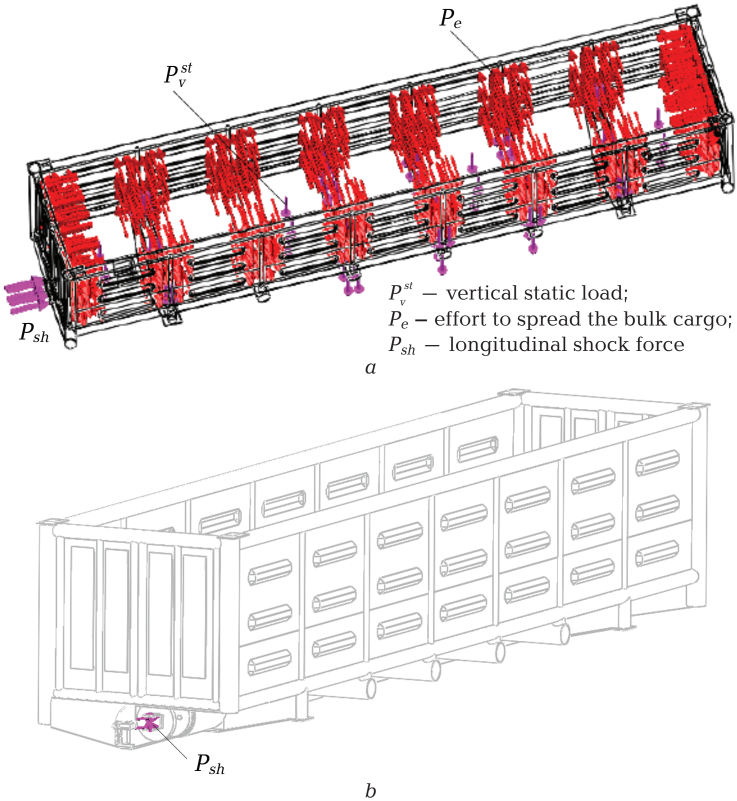
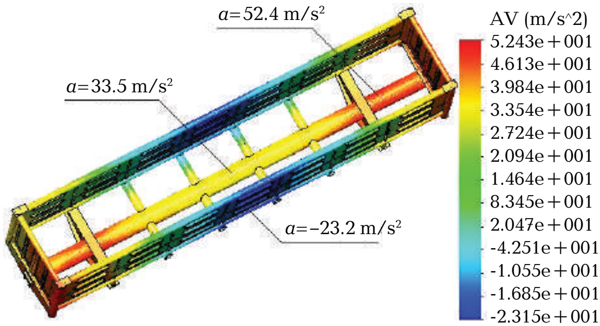


Fig. 2.5 Strength model of a gondola wagon body, taking into account the introduction of viscous substances as shock absorbers of longitudinal forces in the center beam: *a* – general view; *b* – diagram of the application of a longitudinal load to the adapter

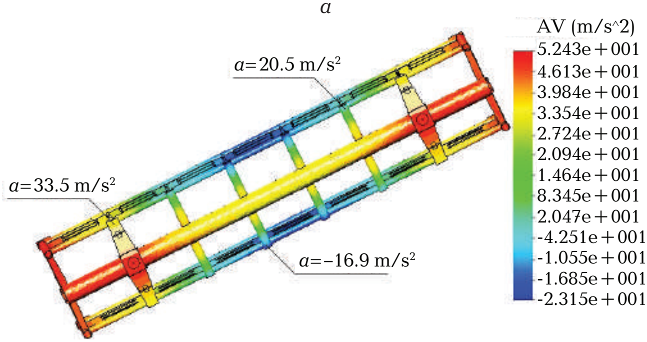
09G2S steel is used as the bearing structure material. When compiling the model of the fortress, the possible difference in the levels of the bodies of the wagon couplers interacting with each other is not taken into account.

The calculation results are shown in Fig. 2.6. From the studies carried out, it can be concluded that the maximum acceleration of the wagon body occurs in the zone from the cantilever part to the zone of interaction of the center beam with the pivot and is about 50 m/s (≈5*g*). In the zone from the pivot beam to the middle part of the center beam, the magnitude of accelerations decreases and amounts to about 30 m/s (≈3*g*).

Also, calculations are carried out for other types of railway wagons. The distribution of accelerations along the center beam length of the frames of railway wagons is shown in Fig. 2.7.



a



b

Fig. 2.6 Distribution of acceleration fields acting on the bearing structure of the gondola wagon body, taking into account the introduction of viscous substances as shock absorbers of longitudinal forces in the center beam: *a* — top view; *b* — bottom view

As an example, Table 2.1 shows a comparative analysis of the accelerations occurring in the frame of the gondola wagon body, as the most loaded element of its structure during a shunting collision, taking into account the presence of a binder in it and in its absence.

So, the maximum accelerations in the gondola wagon frame, taking into account the presence of a binder in it, when exposed to an impact load, arise in the zone from the frontal beam to the pivot beam, and in its absence, in the central zone of the center beam. This is due to the fact that the kinetic energy of the impact is damped by the harness system and is converted into energy of dissipation. In order to check the adequacy of the developed model, the Fisher criterion is used [36–38].

$$F_p = \frac{S_{ad}^2}{S_y^2}, \tag{2.9}$$

where S_{ad}^2 — adequacy variance; S_y^2 — reproducibility variance.

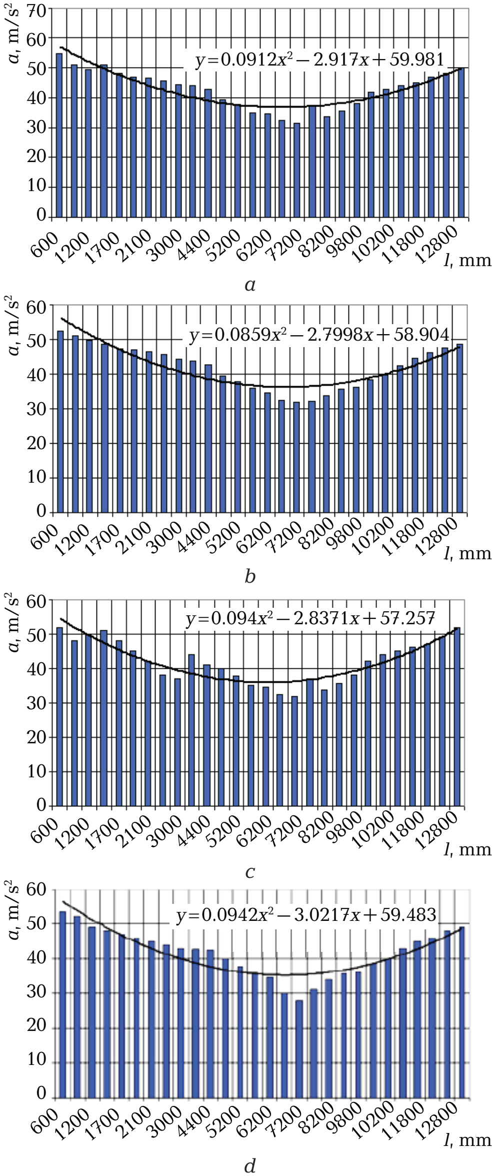


Fig. 2.7 Distribution of accelerations along the length of the center beams of the frames of railway wagons under the action of the longitudinal impact force on the adapter: *a* — flat wagon; *b* — gondola wagon; *c* — covered wagon; *d* — hopper wagon

Table 2.1 Comparative analysis of accelerations acting in the frame of the gondola wagon body

Frame element	Acceleration, m/s^2		Discrepancy, %
	no binder in the center beam	binder in the center beam	
Center beam (central part)	50.3	33.5	33.4
Pivot beam	42.5	38.4	9.6
Frontal beam	21.7	48.3	55.1
Intermediate beam (the first from the pivot beam towards the middle of the center beam)	35.1	33.5	4.6
Intermediate beam (the second from the pivot beam towards the middle of the center beam)	38.2	32.4	15.2

The adequacy variance is found by the formula:

$$S_{ad}^2 = \frac{\sum_{i=1}^{n}\left(y_i - y_i^p\right)}{f_i}, \tag{2.10}$$

where y_i^p — calculated value of the value obtained by modeling; f_i — the number of degrees of freedom.

$$f_i = N - q, \tag{2.11}$$

where N — the number of experiments in the planning matrix; q — the number of coefficients of the equation.

The dispersion of reproducibility is determined by the formula:

$$S_y^2 = \frac{1}{N}\sum_{i=1}^{n} S_i^2, \tag{2.12}$$

where S_i^2 — variance in each row where parallel experiments are carried out.

It is found that the model under consideration is linear and characterizes the change in the acceleration of the wagon from the longitudinal force acting on the adapter. In this case, the number of degrees of freedom at $N = 5$ will be $f_1 = 3$.

When determining the adequacy of the model, it is found that with reproducibility variance $S_y^2 = 1.95$ and adequacy variance $S_{ad}^2 = 2.0$, the actual value of the Fisher criterion $F_p = 1.03$, which is less than the table value of the criterion $F_t = 5.41$. So, with the level of significance $p = 0.05$, the hypothesis about the adequacy of the developed model is not disputed. The approximation error is 8.76 %.

When conducting preliminary exploratory studies, let's additionally consider possible options for filling the lower and upper strapping of the walls of the side gondola wagons with a substance with viscous properties, but today such options did not allow achieving a sufficient level of economic justification.

Let's also consider the option of introducing the concept on wagons, the spinal beams of which consist of two pipes.

The design of the concept is shown in Fig. 2.8, and its placement on the wagon in Fig. 2.9 [6].

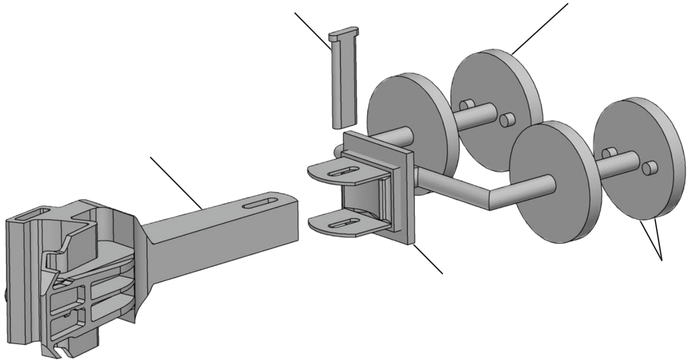


Fig. 2.8 Concept of automatic coupler harness:
1 — automatic coupler body; 2 — wedge; 3 — piston;
4 — throttle valves; 5 — thrust bar

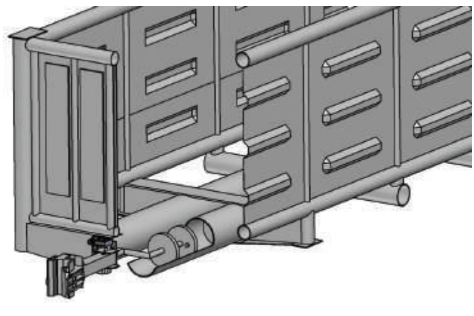


Fig. 2.9 Placement of the harness concept on a gondola wagon

The scheme of the concept is similar to the above constructions (Fig. 2.10).

It is important to note that the presence of a binder in the cantilever parts will also contribute to the anti-corrosion protection of the center beam of the wagon in these zones.

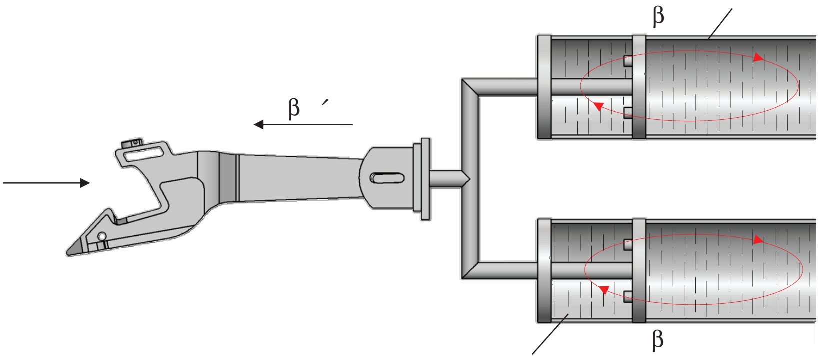


Fig. 2.10 Operation scheme of the harness concept of a gondola wagon

Conclusions to chapter 2

1. To reduce the dynamic loading of wagons in operation, it is proposed to introduce the concept of an automatic coupler harness. It is proposed to exclude automatic coupler harness devices from structures and transfer their functions to absorb energy, arises from the action of operational loads on the center beam, as well as the upper and lower strapping of the side walls, which are proposed to be made of round pipes and filled with material with damping and anti-corrosion properties, which will reduce the material consumption of the wagon and, accordingly, increase its bearing capacity and the loading volume of the body, as well as extend the maintenance-free period of its operation.

2. Mathematical modeling of the dynamic loading of the wagon is carried out taking into account the application of the concept of the harness. It is found that the acceleration that acts on the bearing structure of a wagon equipped with a harness concept during a shunting collision is 10 % lower than the value of accelerations obtained with a typical scheme of loading a wagon center beam.

The effectiveness of this solution is achieved when the value of the viscous resistance coefficient is not higher than 120 kN·s/m;

3. Computer simulation of the dynamic loading of the wagon bearing structure is carried out and the numerical values and fields of acceleration distribution are determined. The adequacy of the developed models is checked according to the Fisher criterion. It is found that the hypothesis of adequacy is not rejected.

The proposed technical solution for the implementation of the harness system of wagons is also advisable to use in structures, both domestic and foreign production, as well as samples of rolling stock.

Chapter 3
Features of creating articulated wagons from round pipes

3.1 Advantages of using articulated wagons

To increase the volume of cargo transportation by rail, articulated wagons have been used. A feature of such wagons is that their design consists of two sections, which are supported by three running parts. The sections interact with each other using a joint assembly. As an example, Fig. 3.1 shows a joint of the SAC-1 type, manufactured by WABTAC [39, 40].

Fig. 3.1 SAC-1 articulation device

At the same time, it becomes possible to use joints from other manufacturers.

Today, platform wagons are the most widespread among such wagons. The variety of articulated wagons is very large (Fig. 3.2—3.6), which is due to a number of advantages over other types:

— reduction in manufacturing costs;
— reduction of loading of frames with vertical loads;
— savings in the transportation of goods, especially on long «shoulders»;
— reduction of the required wagon fleet;
— reducing the cost of the life cycle of the wagon, etc.

Fig. 3.2 Articulated platform wagon

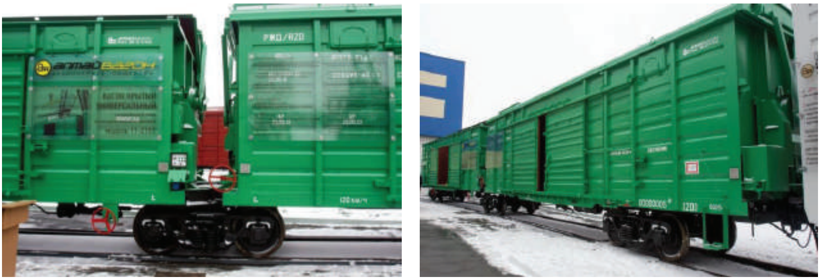


Fig. 3.3 Articulated covered wagon

Fig. 3.4 Articulated gondola wagon

Fig. 3.5 Articulated hopper wagon

Fig. 3.6 Articulated tank wagon

The main distinguishing features of articulated wagons are their design, loading method, type of undercarriage, articulation units, etc.

3.2 Design features of articulated wagons made of round pipes

To increase the efficiency of wagon operation, it is proposed to create articulated wagons on the basis of the developed wagons from round pipes.

Let's consider the design features of an articulated wagon using the example of a flat wagon, as the most common type of articulated wagon in operation (Fig. 3.7). The wagon bearing structure consists of two sections supported by three bogies. The interaction of the sections with each other is carried out through the SAC-1 articulation device.

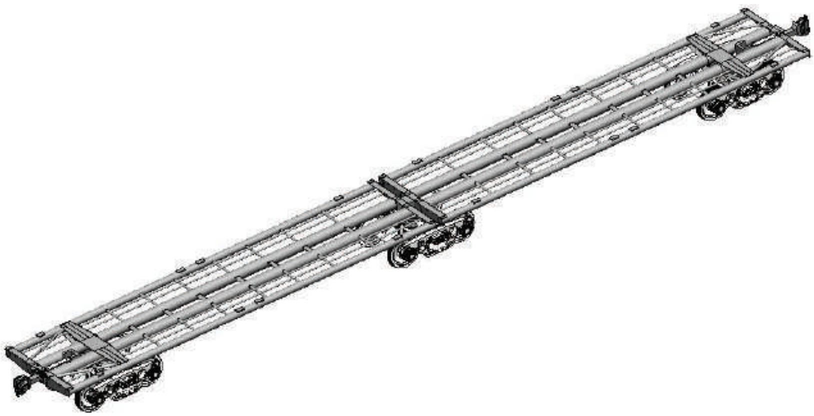


Fig. 3.7 Articulated flat wagon made of pipes of circular cross-section

The pivot beam has a structure identical to that used on the prototype wagon. From the side of the support of the sections to the middle trolley, the pivot beam is replaced by a beam of circular cross-section.

For the possibility of transporting containers in a flat wagon, it is provided for the setting of folding fitting stops in the middle part of the sections, which makes it possible to transport containers of various standard sizes.

To study the dynamic loading of an articulated flat wagon, a mathematical model developed by prof. G. Bogomaz. This model is refined by taking into account the displacements of the two sections under operating load conditions. Also in the model, elastic connections between containers and the bearing structure of a flat wagon are canceled [41–44].

It is taken into account that the flat wagon is loaded with containers of standard size 1CC. Investigation of oscillations of a flat wagon with containers is carried out in a flat coordinate system.

When compiling a mathematical model, it is taken into account that each section of the flat wagon has its own degree of freedom. This assumption is based on the fact that the design features of the articulation device allow such movements.

The design diagram of an articulated flat wagon with containers under the action of a longitudinal force on the bearing structure is shown in Fig. 3.8.

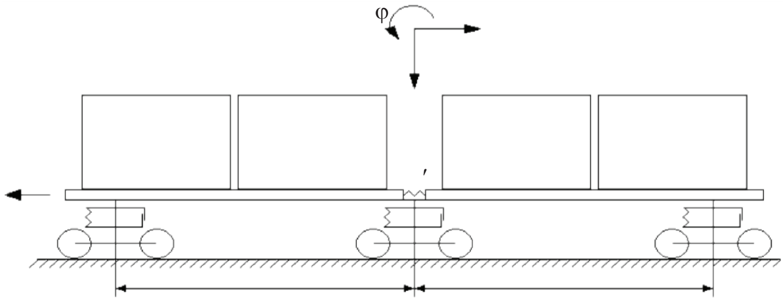


Fig. 3.8 Design diagram of an articulated flat wagon

$$M'_{F_1} \cdot \ddot{x}_{F_1} + M_{F_1} \cdot h \cdot \ddot{\varphi}_{F_1} + k'\left(x_{F_1} - x_{F_2}\right) = P_l, \tag{3.1}$$

$$\begin{gathered} I_{F_1} \cdot \ddot{\varphi}_{F_1} + M_{F_1} \cdot h \cdot \ddot{x}_{F_1} - g \cdot \varphi_{F_1} \cdot M_{F_1} \cdot h = \\ = l \cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_1} - \operatorname{sign}\dot{\Delta}_2^{F_1}\right) + l\left(k_1 \cdot \dot{\Delta}_1^{F_1} - k_2 \cdot \dot{\Delta}_2^{F_1}\right), \end{gathered} \tag{3.2}$$

$$M_{F_1} \cdot \ddot{z}_{F_1} = k_1 \cdot \Delta_1^{F_1} + k_2 \cdot \Delta_2^{F_1} - F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_1} - \operatorname{sign}\dot{\Delta}_2^{F_1}\right), \tag{3.3}$$

$$m_i \cdot \ddot{x}_{F_1} + \left(m_i \cdot z_{ci}\right) \cdot \ddot{\varphi}_{F_1} = 0, \tag{3.4}$$

$$I_i \cdot \ddot{\varphi}_{F_1} + \left(m_i \cdot z_{ci}\right) \cdot \ddot{x}_{F_1} - g \cdot \left(m_i \cdot z_{ci}\right) \cdot \varphi_{F_1} = 0, \tag{3.5}$$

$$m_i \cdot \ddot{z}_{F_1} = 0, \tag{3.6}$$

$$M'_{F_2} \cdot \ddot{x}_{F_2} + M_{F2} \cdot h \cdot \ddot{\varphi}_{F_2} - k'\left(x_{F_1} - x_{F_2}\right) = 0, \tag{3.7}$$

$$\begin{aligned} &I_{F_2} \cdot \ddot{\varphi}_{F_2} + M_{F_2} \cdot h \cdot \ddot{x}_{F_2} - g \cdot \varphi_{F_2} \cdot M_{F2} \cdot h = \\ &= l \cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_2} - \operatorname{sign}\dot{\Delta}_2^{F_2}\right) + l\left(k_1 \cdot \dot{\Delta}_1^{F_2} - k_2 \cdot \dot{\Delta}_2^{F_2}\right), \end{aligned} \tag{3.8}$$

$$M_{F_2} \cdot \ddot{z}_{F_2} = k_1 \cdot \Delta_1^{F_2} + k_2 \cdot \Delta_2^{F_2} - F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_2} - \operatorname{sign}\dot{\Delta}_2^{F_2}\right), \tag{3.9}$$

$$m_i \cdot \ddot{x}_{F_2} + \left(m_i \cdot z_{ci}\right) \cdot \ddot{\varphi}_{F_2} = 0, \tag{3.10}$$

$$I_i \cdot \ddot{\varphi}_{F_2} + \left(m_i \cdot z_{ci}\right) \cdot \ddot{x}_{F_2} - g \cdot \left(m_i \cdot z_{ci}\right) \cdot \varphi_{F_2} = 0, \tag{3.11}$$

$$m_i \cdot \ddot{z}_{F_2} = 0, \tag{3.12}$$

where

$$\Delta_1^i = z_{F_i} - l \cdot \varphi_{F_i}; \quad \Delta_2^i = z_{F_i} + l \cdot \varphi_{F_i}, \tag{3.13}$$

M'_{F_i} – gross weight of the first section of the flat wagon; M_{F_i} – weight of the bearing structure and the first section of the flat wagon; I_{F_i} – moment of inertia of the i-th section of the flat wagon; P_l – value of the longitudinal force acting on the automatic coupler; l – half of the base of the platform wagon section; F_{FR} – absolute value of the dry friction force in the spring set; k' – rigidity of the connection between the sections; k_1, k_2 – rigidity of springs of spring sets of bogies of a platform wagon (bogie model 18-100); m_i – container weight; z_{ci} – height of the center of gravity of the container; I_i – moment of inertia of the i-th container; x_i, φ_i, z_i – coordinates that determine the movement of the sections of the flat wagon relative to the corresponding axes.

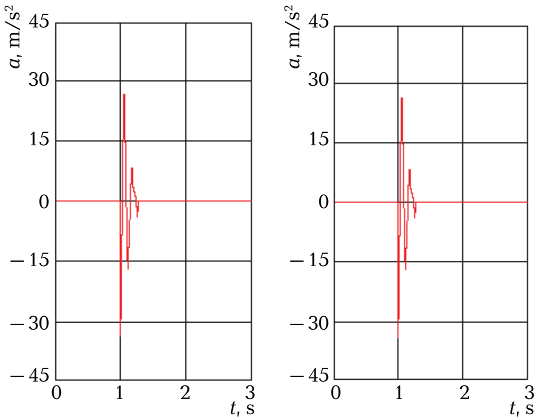
The longitudinal load acting on the bearing structure of the flat wagon is taken equal to 2.5 MN [7, 8]. Differential equations (3.1)–(3.12) are solved in the MathCad environment [30, 31].

The calculation results show that the acceleration acting on the bearing structure of the first from the side of the force of the platform wagon section is 38.2 m/s^2, and the second – about 37.5 m/s^2 (Fig. 3.9).

The next step in this study is to determine the strength indicators of the bearing structure of an articulated flat wagon.

To determine the strength indicators of the bearing structure of a flat wagon made of round pipes, a computer model is compiled. The calculation is carried out using the finite element method in the CosmosWorks software environment [12].

The design diagram of the bearing structure of the platform wagon of design mode I (jerk) is shown in Fig. 3.10. In this case, a longitudinal load of 2.5 MN is applied to the front stops [7, 8].



a *b*

Fig. 3.9 Acceleration acting on the bearing structure of an articulated flat wagon: *a* — the first section of the flat wagon from the side of the action of the longitudinal force; *b* — the second section of the flat wagon from the side of the action of the longitudinal force

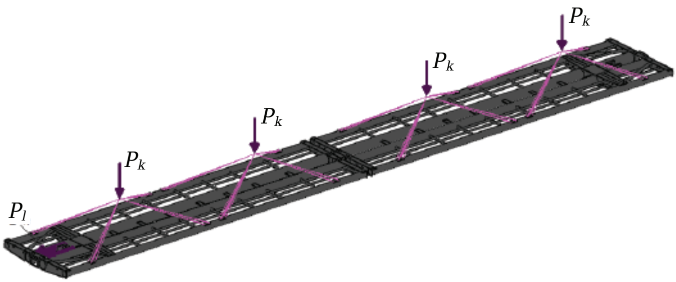


Fig. 3.10 Design diagram of the bearing structure of an articulated flat wagon

It is taken into account that each section of the flat wagon is loaded with two 20-foot containers. The vertical load from the containers is applied to the horizontal surfaces of the fitting stops in the form of a distance load taking into account the center of gravity of the containers.

Spatial isoparametric tetrahedrons are used to compile the finite element model of the platform wagon bearing structure. The optimal number of grid elements is determined using the graphical-analytical method. The number of grid elements is 5406526, nodes — 1538366. The maximum size of the grid element is 15 mm, the minimum is 3 mm, the maximum aspect ratio of the elements is 3078.9; the percentage of elements with an aspect ratio of less than three — 87.6; more than ten — 0.212. The number of elements in a circle is 8. The ratio of increasing the size of an element is 1.7.

The fixing of the model is carried out in the areas of bearing of the bearing structure on the chassis. The results of calculating the strength of the bearing structure of a flat wagon are given below.

In this case, the maximum equivalent stresses arise in the cantilever parts of the center beam and are about 200 MPa, that is, they do not exceed the permissible ones (Fig. 3.11) [7, 8].

The maximum displacements in the nodes of the structure are recorded in the middle parts of the sections, they are 3.8 mm (Fig. 3.12), the maximum deformations are $2.3 \cdot 10^{-3}$.

Also, designs of other types of articulated wagons from round pipes are developed (Fig. 3.13–3.18).

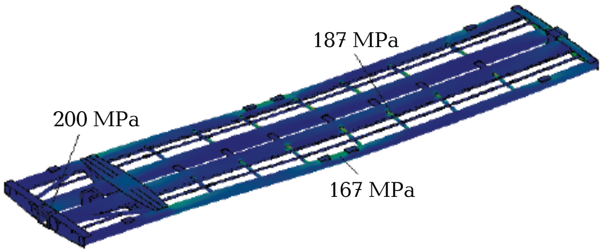


Fig. 3.11 Stress state of a section of an articulated flat wagon

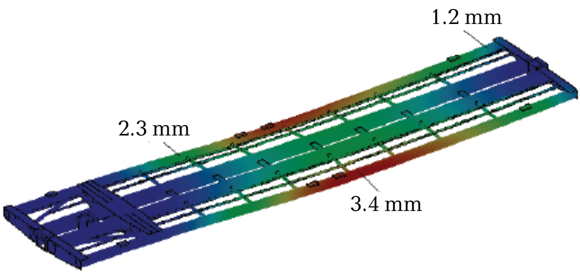


Fig. 3.12 Movement in the nodes of the section of an articulated flat wagon

Fig. 3.13 Articulated gondola wagon

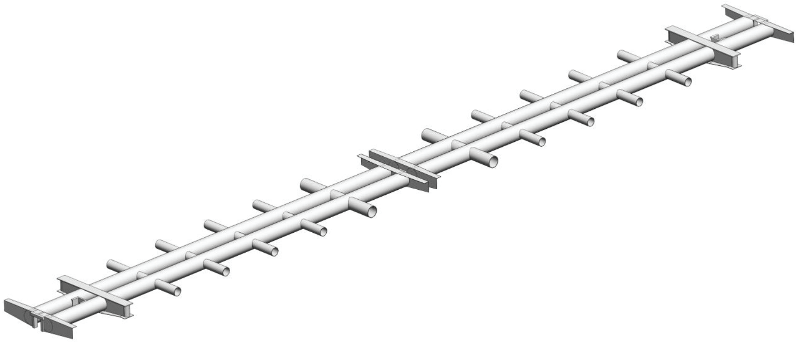


Fig. 3.14 Frame of an articulated gondola wagon

Fig. 3.15 Articulated covered wagon

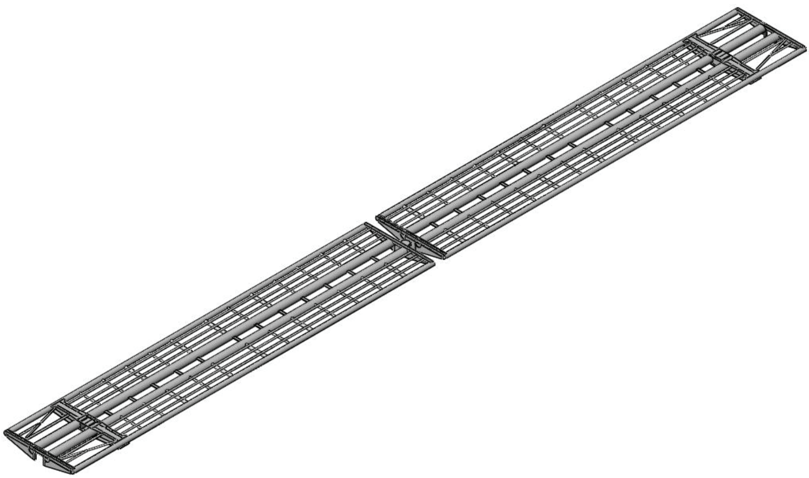


Fig. 3.16 Frame of an articulated covered wagon

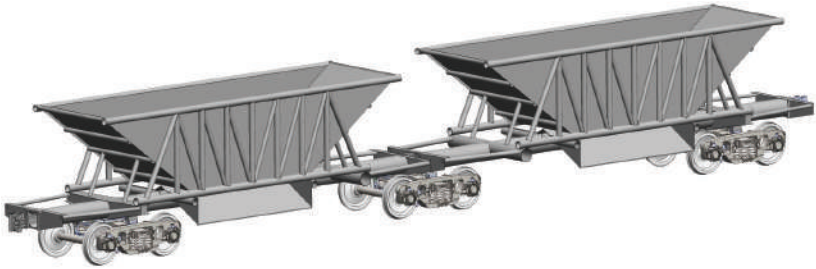


Fig. 3.17 Articulated hopper wagon

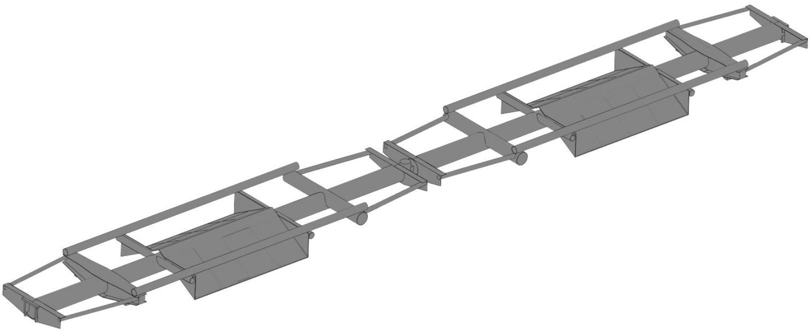


Fig. 3.18 Frame of an articulated hopper wagon

The numerical values of the accelerations acting on the bearing structures of the wagons made of round pipes, determined by mathematical modeling, are shown in Table 3.1.

Table 3.1 Numerical values of accelerations acting on the bearing structures of articulated wagons from round pipes

Wagon type	Acceleration that acts on the first section from the side of the force application	Acceleration that acts on the second section from the side of the force application
Gondola wagon	32.0	31.0
Covered wagon	28.4	27.9
Hopper wagon	38.2	37.5

To reduce the dynamic loading of the bearing structures of articulated wagons, it is proposed to use the concept of a harness device on them. To determine the dynamic loading of the wagon bearing structure, taking into account the use of the harness concept, a mathematical functional description of the dynamic loading process under the action of a longitudinal force on the front stop (tension — jerk) is carried out. The calculation is carried out on the example of an articulated platform wagon, as one of the most common types of articulated wagons in operation.

The design scheme is shown in Fig. 3.19.

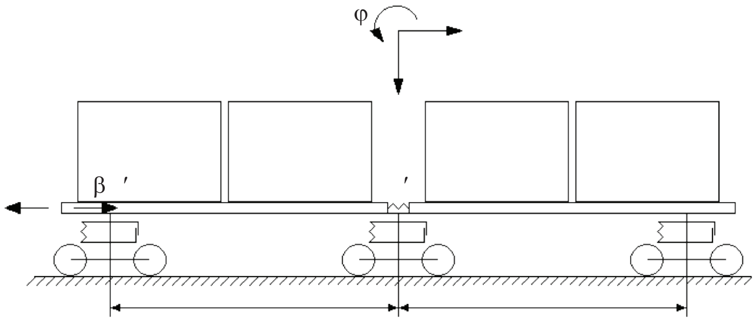


Fig. 3.19 Design diagram of an articulated flat wagon

$$M'_{F_1}\cdot\ddot{x}_{F_1}+M_{F_1}\cdot h\cdot\ddot{\varphi}_{F_1}+k'\left(x_{F_1}-x_{F_2}\right)=P_l-\beta\cdot\dot{x}_{F_1}, \tag{3.14}$$

$$I_{F_1}\cdot\ddot{\varphi}_{F_1}+M_{F_1}\cdot h\cdot\ddot{x}_{F_1}-g\cdot\varphi_{F_1}\cdot M_{F_1}\cdot h=$$
$$=l\cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_1}-\operatorname{sign}\dot{\Delta}_2^{F_1}\right)+l\left(k_1\cdot\dot{\Delta}_1^{F_1}-k_2\cdot\dot{\Delta}_2^{F_1}\right), \tag{3.15}$$

$$M_{F_1}\cdot\ddot{z}_{F_1}=k_1\cdot\Delta_1^{F_1}+k_2\cdot\Delta_2^{F_1}-F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_1}-\operatorname{sign}\dot{\Delta}_2^{F_1}\right), \tag{3.16}$$

$$m_i\cdot\ddot{x}_{F_1}+\left(m_i\cdot z_{ci}\right)\cdot\ddot{\varphi}_{F_1}=0, \tag{3.17}$$

$$I_i\cdot\ddot{\varphi}_{F_1}+\left(m_i\cdot z_{ci}\right)\cdot\ddot{x}_{F_1}-g\cdot\left(m_i\cdot z_{ci}\right)\cdot\varphi_{F_1}=0, \tag{3.18}$$

$$m_i\cdot\ddot{z}_{F_1}=0, \tag{3.19}$$

$$M'_{F_2}\cdot\ddot{x}_{F_2}+M_{F2}\cdot h\cdot\ddot{\varphi}_{F_2}-k'\left(x_{F_1}-x_{F_2}\right)=0, \tag{3.20}$$

$$I_{F_2}\cdot\ddot{\varphi}_{F_2}+M_{F_2}\cdot h\cdot\ddot{x}_{F_2}-g\cdot\varphi_{F_2}\cdot M_{F2}\cdot h=$$
$$=l\cdot F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_2}-\operatorname{sign}\dot{\Delta}_2^{F_2}\right)+l\left(k_1\cdot\dot{\Delta}_1^{F_2}-k_2\cdot\dot{\Delta}_2^{F_2}\right), \tag{3.21}$$

$$M_{F_2}\cdot\ddot{z}_{F_2}=k_1\cdot\Delta_1^{F_2}+k_2\cdot\Delta_2^{F_2}-F_{FR}\left(\operatorname{sign}\dot{\Delta}_1^{F_2}-\operatorname{sign}\dot{\Delta}_2^{F_2}\right), \tag{3.22}$$

$$m_i\cdot\ddot{x}_{F_2}+\left(m_i\cdot z_{ci}\right)\cdot\ddot{\varphi}_{F_2}=0, \tag{3.23}$$

$$I_i\cdot\ddot{\varphi}_{F_2}+\left(m_i\cdot z_{ci}\right)\cdot\ddot{x}_{F_2}-g\cdot\left(m_i\cdot z_{ci}\right)\cdot\varphi_{F_2}=0, \tag{3.24}$$

$$m_i\cdot\ddot{z}_{F_2}=0, \tag{3.25}$$

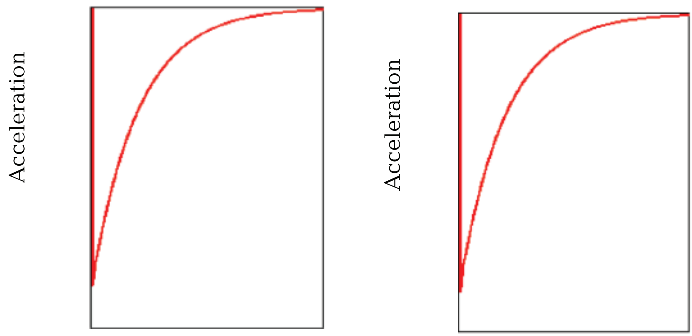
where

$$\Delta_1^i = z_{F_i} - l \cdot \varphi_{F_i}; \quad \Delta_2^i = z_{F_i} + l \cdot \varphi_{F_i}, \tag{3.26}$$

M'_{F_i} — gross weight of the first section of the flat wagon; M_{F_i} — weight of the bearing structure and the first section of the flat wagon; I_{F_i} — moment of inertia of the i-th section of the flat wagon; P_l — value of the longitudinal force acting on the automatic coupler; l — half of the base of the platform wagon section; F_{FR} — absolute value of the dry friction force in the spring set; k' — rigidity of the connection between the sections; k_1, k_2 — rigidity of springs of spring sets of bogies of a platform wagon (bogie model 18-100); m_i — container weight; z_{ci} — height of the center of gravity of the container; I_i — moment of inertia of the i-th container; x_i, φ_i, z_i — coordinates that determine the movement of the sections of the flat wagon relative to the corresponding axes.

The studies are carried out in a flat coordinate system. The interaction of the sections with each other is simulated through an elastic connection. The longitudinal load acting on the bearing structure of the flat wagon is taken to be 2.5 MN [7, 8]. Initial displacements at speed are taken equal to zero. Differential equations (1)–(12) are solved in the MathCad environment by the Runge — Kutta method [30, 31].

The calculation results show that the acceleration acting on the bearing structure of the first section of the flat wagon from the side of the force action is 34.9 m/s^2, and on the second — 34.2 m/s^2 (Fig. 3.20).



a *b*

Fig. 3.20 Acceleration acting on the bearing structure of an articulated flat wagon: *a* — the first section of the flat wagon from the side of the action of the longitudinal force; *b* — the second section of the flat wagon from the side of the action of the longitudinal force

In this case, the coefficient of viscous resistance created by the concept of the harness must be at least 70 kN/s·m.

So, taking into account the use of the concept of a harness device on an articulated flat wagon, it becomes possible to reduce its dynamic load by almost 10 %.

Calculations have been carried out for other types of articulated wagons made of round pipes. The numerical values of the accelerations are given in Table 3.2.

Table 3.2 Numerical values of accelerations acting on the bearing structures of articulated wagons made of round pipes

Wagon type	Acceleration that acts on the first section from the side of the force application	Acceleration that acts on the second section from the side of the force application
Gondola wagon	28.3	27.9
Covered wagon	25.7	25.2
Hopper wagon	28.5	28.1

The conducted studies confirm the feasibility of introducing the concept of a harness device on a rolling stock.

Conclusions to chapter 3

1. To increase the efficiency of wagon operation, it is proposed to create articulated wagons on the basis of the developed wagons from round pipes.

The design features of an articulated wagon are considered on the example of a flat wagon, as the most common type of articulated wagon in operation. In this case, the interaction of the sections of the wagon with each other is proposed to be carried out using a conventional articulation device of the SAC-1 type.

2. Mathematical modeling of the dynamic loading of an articulated flat wagon has been carried out. The studies are carried out in a flat coordinate system. The interaction of the sections with each other is simulated using an elastic connection.

The longitudinal load acting on the bearing structure of the flat wagon is assumed to be 2.5 MN. The calculation results show that the acceleration acting on the bearing structure of the first from the side of the force of the platform wagon section is 38.2 m/s^2, and the second — about 37.5 m/s^2. The resulting accelerations are taken into account when calculating the strength of the bearing structure of an articulated flat wagon. It has been established that the maximum equivalent stresses in the bearing structure do not exceed the standard.

3. To reduce the dynamic loading of bearing structures of articulated wagons, it is proposed to use the concept of a harness device on them. To determine the dynamic loading of the wagon bearing structure, taking into account the use of the harness concept, a mathematical functional description of the dynamic loading process under the action of a longitudinal force on the front stop (tension — jerk) is carried out. In this case, the coefficient of viscous resistance created by the concept of the harness device must be at least 70 kN/s·m.

It has been established that taking into account the use of the concept of a harness on articulated wagons, it becomes possible to reduce their dynamic loading by almost 10 %.

General conclusions

1. An implementation of round pipes as bearing elements of railway wagons and tank containers has been substantiated. Comprehensive strength calculations of bearing elements of railway wagons and tank containers made of round pipes have been carried out.

One of the most promising and new methods is used to optimize the bearing structures of the wagons – optimization by strength reserves. Strength analysis is carried out using the finite element method in the CosmosWorks software environment. The results of the calculation make it possible to conclude that, taking into account the use of round pipes as bearing elements of wagons, the maximum equivalent stresses for all design schemes do not exceed the permissible ones.

Determination of the fatigue strength of the wagons' bearing structures made of round pipes has been carried out. At the same time, the test base is 10^7. The calculation results confirm the feasibility of the decisions taken.

Calculation of the design service life of the bearing structures of the wagons made of round pipes show that the standard service life is at least 32 years, that is, at least for the life cycle of the wagon.

The introduction of round pipes as bearing elements of the frame of the tank container is justified. The calculations are carried out with respect to a 1CC tank container. It has been found that taking into account the measures to improve the bearing structure of the tank container, it becomes possible to reduce its weight by 40 % compared to the design of the prototype.

It has been proven that the introduction of round pipes as bearing elements of vehicles is a reasonable solution and will significantly reduce the cost of their manufacture in the conditions of wagon enterprises.

2. The concept of an automatic coupler harness has been created to reduce the dynamic loading of the bearing structures of wagons made of round pipes under operating conditions. and the lower trims of the side walls, which are proposed to be made of round pipes and filled with material with damping and anti-corrosion properties, which will reduce the material consumption of the wagon and, accordingly, increase its bearing capacity and the loading volume of the body, as well as extend its maintenance-free service life.

3. Mathematical modeling of dynamic loading of bearing structures of wagons made of round pipes equipped with a harness concept has been carried out. It has been found that the acceleration that acts on the bearing structure of a wagon equipped with a harness concept during a shunting collision is 10 %

lower than the value of accelerations obtained with a typical scheme of loading a wagon center beam. The effectiveness of this solution is achieved when the value of the viscous resistance coefficient is not higher than 120 kN·s/m.

4. Computer modeling of dynamic loading of bearing structures of wagons made of round pipes equipped with a harness concept has been carried out. The numerical values of the accelerations and the fields of their distributions relative to the bearing structures of the wagons have been determined. The calculation has been carried out in the CosmosWorks software environment.

5. The adequacy of the developed models has been verified. Fisher's criterion has been used as a calculation. It has been found that the hypothesis of adequacy is not rejected.

The proposed technical solution for the implementation of the harness system of wagons is also advisable to use in structures, both domestic and foreign production, as well as in rolling stock samples.

6. Computer models of the main types of wagons of articulated type made of round pipes have been created. In this case, the interaction of the sections of the wagon with each other is proposed to be carried out using a conventional articulation device of the SAC-1 type. Attention has been drawn to the main types of wagons used on broad gauge railways, flat wagons, gondola wagons, covered wagons and hopper wagons.

7. Mathematical modeling of the dynamic loading of the bearing structures of articulated wagons made of round pipes has been carried out. The studies have been carried out in a flat coordinate system. The interaction of the sections with each other has been simulated using an elastic connection. Attention is drawn to the presence of three degrees of freedom in each section: jerking oscillations, bouncing and galloping.

The longitudinal load acting on the bearing structure of the flat wagon is assumed to be 2.5 MN. The resulting accelerations are taken into account when calculating the strength of the bearing structure of an articulated flat wagon. It has been established that the maximum equivalent stresses in the bearing structure do not exceed the standard.

8. Mathematical modeling of dynamic loading of bearing structures of articulated wagons made of round pipes equipped with a harness concept has been carried out. In this case, the coefficient of viscous resistance created by the concept of the harness device must be at least 70 kN/s·m.

It has been established that taking into account the use of the concept of a harness on articulated wagons, it becomes possible to reduce their dynamic loading by almost 10 %.

The research will contribute to the creation of innovative designs of freight wagons and tank containers, recommendations, as well as a regulatory framework for their design, reduction of manufacturing costs in conditions of wagon-building enterprises, increasing the efficiency of rolling stock operation and maintaining its leading positions in the transport services market.

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