Представлені результати, а також особливості проведених теоретичних та експериментальних досліджень з впровадження стикованого виконання хребтової балки вагонів-окатишевозів. Практична реалізація запропонованих рішень дозволяє знизити собівартість виготовлення таких вагонів на 10 %. Отримані результати повністю підтверджують конструктивну ефективність стикованого виконання хребтової балки вагонів-окатишевозів на протязі встановленого п'ятнадцятирічного строку їх служби

Ключові слова: транспортна механіка, вантажні вагони, несучі системи, ресурсозбереження, напружено-деформований стан, випробування

Представлены результаты, а также особенности проведенных теоретических и экспериментальных исследований по внедрению стыкованного исполнения хребтовой балки вагонов-окатышевозов. Практическая реализация предложенных решений позволяет снизить себестоимость изготовления таких вагонов на 10 %. Полученные результаты полностью подтверждают конструктивную эффективность стыкованного исполнения хребтовой балки вагонов-окатышевозов на протяжении установленного пятнадцатилетнего срока их службы

Ключевые слова: транспортная механика, грузовые вагоны, несущие системы, ресурсосбережение, напряженно-деформированное состояние, испытания

EXPERIMENTAL CONFIRMATION OF THE THEORY OF IMPLEMENTATION OF THE COUPLED DESIGN OF CENTER GIRDER OF THE HOPPER WAGONS FOR IRON ORE PELLETS

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1. Introduction

The railroad is of paramount importance for the development of heavy industry, as well as for meeting public needs for transportation. Wagon fleet is the largest and important part of railroad assets [1]. The level of technical and economic indicators of wagons directly affects efficiency of fulfilling the main task of railroad transport – delivery of goods and passengers. In this case, the key source of income in railroad transportation is the use of freight wagons [2]. At the same time, current technical condition of the wagon fleet [3] necessitates its renovation with competitive models.

According to a generally accepted classification [4], freight wagons are divided into universal and specialized. Recognized trends in the development of railways [5, 6] indicate economically-substantiated expediency of increasing the proportion of transportation by specialized wagons. Given this, of particular relevance and importance are the issues related to development of specialized wagons in order to reduce costs in their lifecycle. Some of the most sought after specialized wagons on the railroads of the CIS countries are hopper wagons for iron ore pellets (Fig. 1), which are used for the transportation of hot iron ore pellets and agglomerate. The efficiency of using these wagons is very important, both for railroad transport and metallurgy.

In this case, the rated operation time of such wagons is limited to 15 years, which predetermines the need for their continuous renewal. This in turn renders special relevance and importance to the issue of reducing the cost of manufacturing and repair of such wagons. One of the decisive factors in bringing down the specified cost is the development and implementation of elements whose production will require less material means. A promising way to achieve the desig-

nated objective is the development and implementation of effective solutions on resource-saving. Significant share in the cost of manufacturing hopper wagons for iron ore pellets (as well as other modern freight wagons) is allocated for purchasing specialized expensive wagon profiles. Such profiles may include, for example, wagon I-beam sections of the center girder (Fig. 2) of a hopper wagon for iron ore pellets. Because the specialized wagon rolled steel requires appropriate equipment with its tuning during manufacturing [4, 7], but the demand for it is not constant over time, selling it implies certain specificity. Particularly, when buying it, it is required to purchase similarly overpriced (multiplied by additional coefficient) rolled steel, included in the whole batch, of a length less than actually needed. Consequently, given the impossibility of using such rolled steel for wagon structures, it can be utilized only for a plant's general needs or, alternatively, be sold as scrap metal.
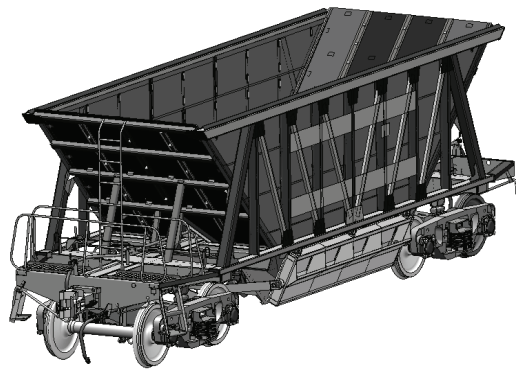

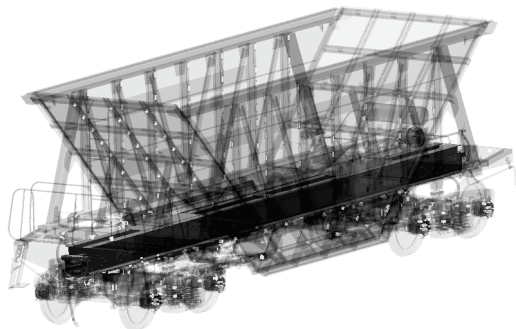Fig. 1. Hopper wagon for iron ore pellets

Fig. 2. Computer model of a hopper wagon for iron ore pellets of standard design, given in transparency mode, highlighting the center girder

The above-described situation predetermines relevance of conducting scientific-research and experimental-design work related to the implementation into load-bearing structures of hopper wagons for iron ore pellets of the coupled profiles (connected by welding) rather than the solid rolled profiles. In this case, high-efficiency realization of such direction can be achieved through modernization of the most material-intensive nodes. For the hopper wagons for iron ore pellets, such a node is the center girder (Fig. 2).

2. Literature review and problem statement

Contemporary studies that deal with reducing the cost of freight wagons are mostly aimed at improving their designs by improving procedures for their design or implementation of new materials. Article [4] reviews, in particular, innovations proposed for the designs of semi-wagons for a «1520 mm railroad space», as well as special features during designing them, but it fails to outline possibilities of applying such innovations for the hopper semi-wagons. The authors of paper [5] define and address promising directions in the design of railroad semi-wagons' bodywork in order to improve technical and economic indicators, but they did not reveal the economic potential of commercial use of the specialized expensive wagon rolled sheet with a non-standard length. Articles [8, 9] present new approaches towards improvement of the dynamic calculations of wagon structures, and report more accurate results based on them. Thus, paper [8] describes appropriate features and results of dynamic characteristics of the platform wagons. Article [9] focuses on presentation of the proposed methods for determining dynamic characteristics for various types of carrying wagon designs. A number of current publications address designs of the new-generation freight wagon engineering, which are developed using advanced materials and technologies. The authors of [5], for example, show results of computer simulation of the prototype of a freight wagon with the basic bearing elements executed without redundant connections. The approaches, presented in [5, 8, 9], however, are focused on utilizing solid profiles and do not allow precise calculation of the implementation of coupled variants of beams designs in different nodes of freight wagons. Paper [6] features innovations, proposed by the authors, in the running gear module, and demonstrates their effect on the body module, however, it fails to consider any variation in its designs. Article [7] published results of work on the generation of promising configurations of profiles that can be used in the production of different types of rolling stock. The authors, however, do not give data on the possibility of creating promising profiles for designs coupled along the length. Paper [10] highlights proposed methods for analysis of railroad structures of the future and techniques to extend their functionality, but it lacks, similar to article [5], data on improving working functionality of non-standard specialized profiles. Important role in today's wagon-building belong to appropriate approaches to designing; article [11], for example, presented a devised methodological apparatus for making optimal decisions. Yet again, it does not provide comprehensive opportunities for the formation of conjugated beams that would combine both optimal configuration and length.

Given the above, we can conclude that the results of analysis of information sources on the examined subject indicate the absence of sufficient methodological and practical materials on the introduction of coupled variants of beam designs to the center girder of hopper wagons for iron ore pellets.

3. The aim and objectives of the study

The aim of present study is experimental strength confirmation of the feasibility of theoretically-substantiated implementation of the coupled design of center girder for the hopper wagons for iron ore pellets.

To achieve the specified aim, the following tasks have been set and solved:

– to work out an engineering-technological solution for the implementation of the coupled design of center girder for hopper wagons for iron ore pellets;

– to estimate the loads acting on the improved design, according to the norms of wagon calculations of wagons;

– to create an adequate spatial finite-element computer model of the improved design of hopper wagons for iron ore pellets, and, based on it, simulation computer modeling of different estimated cases of work in terms of load acceptance by the structure;

– to devise a procedure of experiment to confirm results derived theoretically, and to conduct it.

4. Theoretical and experimental studies into implementation of the coupled design of a center girder for hopper wagons for iron ore pellets

4. 1. Engineering-technological description of implementation of the coupled design of a center girder for hopper wagons for iron ore pellets

We chose as the wagon-prototype to consider a possibility of implementation of the coupled variant of beam designs in a center girder (Fig. 2, 3) of hopper wagons for iron ore pellets their standard structure, namely, model 20-9749 [12] made by GP «Ukrspetswagon» (Ukraine).
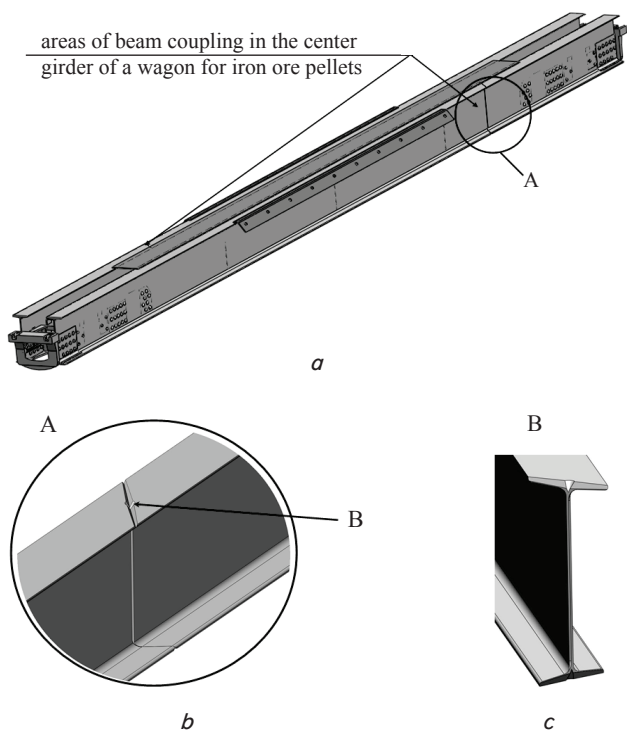
a

b

c

Fig. 3. Coupled variant of beam design in the center girder of a hopper wagon for iron ore pellets:
a — isometric view of the estimated computer spatial model of improved beams in the center girder of hopper wagons for iron ore pellets; *b* — beams of the center girder of improved design in a hopper wagon for iron ore pellets, conjugated for a welding coupling; *c* — edge of the I-beam prepared (with worked-out edges) for butt joint

Implementation of the coupled variant of beams in a center girder will allow efficient use of costly rolled wagon profiles in the appropriate design. Preliminary economic calculations showed that such solutions would have a positive impact on the cost of manufacturing and repair of improved

wagons, namely, create conditions for bringing down the cost by 10 %. In order to accomplish the set goal, we developed engineering and technological solutions (Fig. 3).

In this case, we substantiated places for the implementation of butt joints in the center girder between a pivot beam and a slab (Fig. 3, *a*). Before welding and coupling, it is required to work out the edges (details are in Fig. 3, *b*, *c*).

4. 2. Strength analysis of the center girder using a simulated computer modeling

In this case, the possibility to implement such modernization should be confirmed by the results of theoretical and experimental studies. First, we substantiated the possibility of implementation of the coupled variant of beams theoretically, namely by computer modeling methods using the software complex Cosmos Works based on ECM of GP «Ukrspetswagon». In this case, the studies were conducted in two main areas: strength analysis of the improved center girder node (Fig. 3), and the analysis of its fatigue strength (determining its resource). Since a hopper wagon for iron ore pellets is a wholly-welded carrying system, research into acceptance of operational loads by the center girder node considered exclusively a general structure.

Strength analysis of a hopper wagon for iron ore pellets was conducted in accordance with the applicable acting documents. The wagon is a wholly-welded beam structure.

In the given calculation, we estimated the strained-deformed state of the carrying elements of a wagon caused by basic operating loads using a computing software package that implements the method of finite elements. Configuring the adequacy of the estimated model was carried out based on field tests of the hopper wagons of model 20-9749. The structure was simulated by finite elements. The number of finite elements is 31.423, of nodes – 24.693.

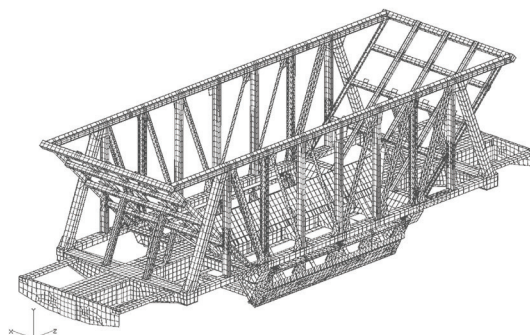The derived finite-element model is shown in Fig. 4.

Fig. 4. Finite-element model

Load-bearing structural elements and reinforcing elements are made of high strength steel. Mechanical characteristics of the materials used for a center girder node are given in Table 1.

In accordance with [3, 4, 8], a wagon frame is estimated for strength under the first (impact, jerk, compression, and stretching) and third (impact, jerk, compression, and stretching) calculation modes.

Table 1

Mechanical characteristics of the materials studied

Name	σ_v –temporal resistance, MPa	σ_t –material's yield limit, MPa	δ – relative elongation, %
Steel 09G2S	430	295	21

When estimating under the first calculation mode, the following combination of loads acting on the frame during impact and jerk is accepted:
– frame gravity force;
– frame longitudinal inertia force;
– gravity force of the load;
– load longitudinal inertia force;
– bogies longitudinal force;
– the force acting on a pivot node;
– longitudinal force of impact or jerk;
– the force applied to the coupler supports;
– vertical force at an off-center interaction between couplers;
– vertical dynamic addition from the action of longitudinal force of the load inertia.

When estimating under the first calculation mode, the following combination of loads acting on the frame during compression and stretching is accepted:
– frame gravity force;
– gravity force of the load;
– longitudinal force of compression or stretching;
– vertical force at an off-center interaction between couplers;
– lateral forces of interaction between wagons in curves.

When estimating under the third calculation mode, the following combination of loads acting on the frame during impact and jerk is accepted:
– frame gravity force;
– frame longitudinal inertia force;
– gravity force of the load;
– load longitudinal inertia force;
– bogies longitudinal force;
– the force acting on a pivot node;

– vertical dynamic force;
– force of the load accepted to be 7.5 % of the gravity force of the load;
– vertical force at an off-center interaction between couplers;
– longitudinal force of impact or jerk, applied to the coupler's supports.

When estimating under the third calculation mode, the following combination of loads acting on the frame during compression and stretching is accepted:
– frame gravity force;
– gravity force of the load;
– longitudinal force of compression or stretching;
– the force, applied to the coupler's supports.
– vertical force at an off-center interaction between couplers;
– vertical dynamic force;
– lateral force of the load accepted to be 7.5 % of the gravity force of the load.

Permissible stress limit values for the center girder were taken in accordance with [3, 4, 8] and are given in Table 2.

Distribution of maximal stresses in the general form of a wagon – in line with the estimated cases – is shown in Fig. 5.

Table 2

Permissible stress limit values for the center girder

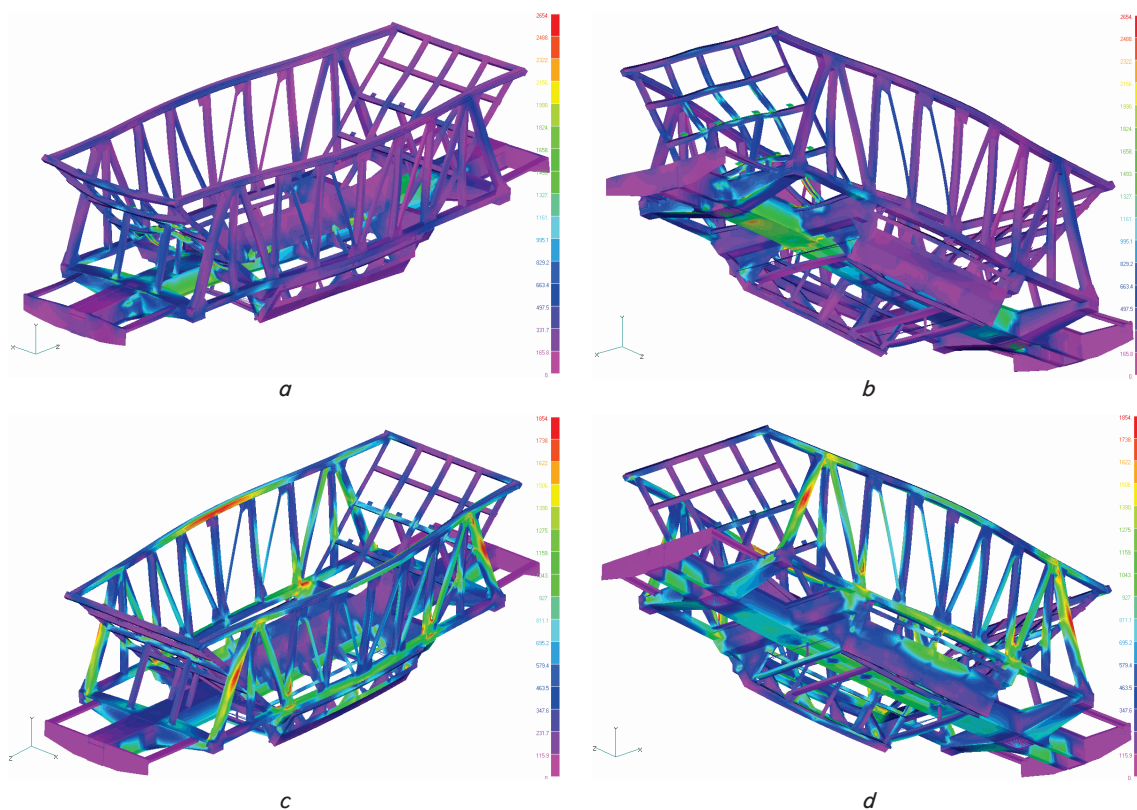| Title	Yield limit,	Permisible stresses	
	$\sigma_{\lim t}$ – material's yield limit, MPa	$\sigma_{\lim v}$ – permissible stress limit values, MPa	
		mode I	mode III
Center girder	295	251	177

a

b

c

d

Fig. 5. Obtained images of the stressed-deformed states of a wagon under estimated modes I and III:
a – estimated mode I – impact; *b* – estimated mode I – impact; *c* – estimated mode III – jerk; *d* – estimated mode III – jerk

The calculation showed that the maximal stresses in a center girder in the area of presumed joint (between a pivot beam and a slab) occur at collision (mode I) and jerk (mode III), in the area of a pivot beam at the bottom; they do not exceed permissible values.

The node of presumed joint, as well as safety coefficients, are given in Table 3.

Table 3

Maximal values of stresses in a center girder

Loading modes		$\sigma_{\lim v}$ – permissible stress limit values, MPa	$\sigma_{\max}$ – maximal estimated equivalent stresses, MPa	η – safety coefficient
I	impact	251	220	1.14
III	jerk	177	85	2.08

Performed calculations for strength of the hopper wagon for transporting hot iron ore pellets and agglomerate of model 20-9749 demonstrated that the design met durability standards. In this case, the joint of center girders in the center girder node is to be made in a zone closer to the slab.

4. 3. Analysis of fatigue strength of the center girder node (determining its resource) conducted experimentally

Estimation of fatigue strength was conducted in accordance with [3, 8] based on fatigue resistance factor:

$$n = \frac{\sigma_{a,N}}{\sigma_{a,e}} \geq [n],$$ (1)

where $\sigma_{a,N}$ is the safety limit (by amplitude) for control area during symmetrical cycle and established mode of loading at a basic number of cycles of $N_0 = 10^7$; $\sigma_{a,e}$ is the magnitude of dynamic stress of conditional symmetrical cycle, reduced to the basic number of cycles N_0, equivalent to damaging action of real operating stresses over the rated service life; $[n]$ is the minimal permissible fatigue resistance factor over selected operation period.

Here the estimated value $\sigma_{a,N}$ was determined from formula:

$$\sigma_{a,N} = \bar{\sigma}_{a,N}(1 - z_p \cdot v_{\sigma_{a,N}}),$$ (2)

where $\bar{\sigma}_{a,N}$ is the average (median) value of fatigue limit of control area; z_p is the quantile of distribution $\sigma_{a,N}$, corresponding to a one-side probability of 95 % ($z_p = 1.645$); $v_{\sigma_{a,N}}$ is the coefficient of variation of fatigue limit of the material ($v_{\sigma_{a,N}} = 0.07$).

The value of $\bar{\sigma}_{a,N}$ was determined from formula:

$$\bar{\sigma}_{a,N} = \frac{\bar{\sigma}_{-1}}{(\bar{k}_\sigma)_k},$$ (3)

where $\bar{\sigma}_{-1}$ is the average (median) value of fatigue limit of a smooth standard sample from the material (in line with GOST 25.502-79) at a symmetrical bending cycle based on $N_0 = 10^7$ cycles; (for the rolled sheet 09G2S, $\bar{\sigma}_{-1} = 200$ MPa); $(\bar{k}_\sigma)_k$ is the average value of total reduction ratio of fatigue limit in the selected control zone relative to the safety limit of a smooth standard sample. The value of $(\bar{k}_\sigma)_k$ corresponds to 1.7 for the joint of a center girder; when determining $\sigma_{a,e}$ we employed formula:

$$\sigma_{a,e} = \sqrt[m]{\frac{N_c^I}{N_0}\sum_{i=1}^{n}(\sigma_{ai}^I)^m p_i + \frac{N_c^{II}}{N_0}\sum_{i=1}^{n}(\sigma_{ai}^{II})^m p_i + \frac{N_c^{III}}{N_0}\sum_{i=1}^{n}(\sigma_{ai}^{III})^m p_i,}$$ (4)

where m is the degree exponent in the equation of fatigue curve in amplitudes; N_c^I is the total number of cycles of vertical dynamic forces over the estimated lifetime of a wagon; N_c^{II} is the total number of cycles of longitudinal impact force over the estimated lifetime of a wagon; N_c^{III} is the total number of loading and unloading cycles over the estimated lifetime of a wagon; N_0 is the basic number of cycles; σ_{ai}^I is the level of stress amplitudes caused by the action of vertical dynamic forces in the interval i; σ_{ai}^{II} is the level of stress amplitudes caused by the action of longitudinal impact force in the interval i; σ_{ai}^{III} is the level of stress amplitudes caused by the action of loading and unloading forces in the interval i; p_i is the frequency of action of amplitude level; n is the number of intervals of stresses.

The magnitude of degree exponent m was determined from formula:

$$m = \frac{18}{(k_\sigma)_k}.$$ (5)

Levels of stress amplitudes caused by the action of vertical dynamic forces were determined from formula:

$$\sigma_{ai}^I = \sigma_{static} \cdot \bar{k}_{qbi},$$ (6)

where σ_{static} is the static stress from the gravity force of wagon gross weight; $\bar{k}_{qbi}$ is the average coefficient of vertical dynamics over interval i.

Static stresses caused by the gravity force of wagon gross weight were derived from calculation by the finite element method of the frame (only the gravity forces of frame and load were applied to the finite element model). Based on calculation data on the center girder between a pivot beam and a slab, they amounted to $\sigma_{static} = 40$ MPa.

Values of $\bar{k}_{qbi}$ were derived from formulae:
at $V_i < 15$ m/s

$$\bar{k}_{qbi} = a\frac{V_i}{15},$$ (7)

at $V_i \geq 15$ m/s

$$\bar{k}_{qbi} = a + 3.6 \cdot 10^{-4} b \frac{V_i - 15}{f_{static}},$$ (8)

where a is a coefficient equal to 0.05 for body parts; b is the coefficient that takes into account effect of the number of axles n in the bogie under one end of the carriage, shall be determined from formula $b = \frac{n+2}{2 \cdot n}$; V_i is the design motion speed, m/s; f_{static} is the static deflection of spring suspension, m.

We accepted as V_i the mean speeds of velocity distribution intervals. The magnitudes of V_i, $\bar{k}_{qbi}$ and p_i are given in Table 4.

The total number of cycles of vertical dynamic forces over operation time is determined from formula:

$$N_c^{\mathrm{I}} = 365 f_e \frac{\bar{L}_c}{V_T} T_p, \tag{9}$$

where f_e is the central (effective) change frequency of dynamic stresses, Hz; a is the coefficient for the body, $a = 1.4$; g is the free fall acceleration, $g = 9.81$ m/s²; f_{static} is the static deflection of spring suspension of the loaded wagon, m; T_p is the wagon operation time, $T_p = 15$ years; $\bar{L}_c$ is the mean daily loaded mileage of wagon, $\bar{L}_c = 220$ km/day; V_T is the average technical motion speed of a wagon, $V_T = 22.4$ m/s.

Table 4

Magnitudes of V_i and p_i

Velocity interval	Mean speed V_i, m/s	$\bar{k}_{qbi}$	Frequency, p_i
$0 \div 12.5$	6.25	0.021	0.03
$12.5 \div 15.0$	13.75	0.046	0.07
$15.0 \div 17.5$	16.25	0.059	0.09
$17.5 \div 20.0$	18.77	0.078	0.12
$20.0 \div 22.5$	21.25	0.096	0.16
$22.5 \div 25.0$	23.75	0.115	0.19
$25.0 \div 27.7$	26.25	0.134	0.16
$27.5 \div 30.0$	28.75	0.152	0.10
$30.0 \div 32.5$	31.25	0.171	0.06
$32.5 \div 35.0$	33.75	0.189	0.02

During impact interaction [3, 8] between a hopper wagon and neighboring wagons there arise quickly-damped oscillatory processes. Levels of stress amplitudes caused by the action of longitudinal impact force can be determined by recalculating the stresses:

$$\sigma_{ai}^{\mathrm{II}} = \frac{\sigma_{i.f.}}{N} F_{ai}^y, \tag{10}$$

where $\sigma_{i.f.}$ is the amplitude of stresses, calculated using the finite element method, during impact interaction between a wagon and the neighboring wagons, MPa; N is the longitudinal impact force acting on the wagon, corresponding to the obtained estimated data; F_{ai}^y is the level of amplitude of the impact longitudinal effort on a coupler over interval i, MPa.

The equivalent stresses caused by the action of longitudinal impact force $N = 3.5$ MN were obtained by calculating the frame using the finite element method. Based on calculation data, in the center girder node between a pivot beam and a slab (upper superstructure of the center girder node), they amounted to $\sigma_{i.f.} = 110$ MPa.

Static distribution of the impact force amplitudes is given in Table 5.

The total number of cycles of longitudinal forces per a coupler over the rated operation time is:

$$N_c^{\mathrm{II}} = N_{year} \cdot T_p, \tag{11}$$

where N_{year} is the number of impact forces per a coupler per year, $N_{year} = 20200$; T_p is the wagon operation time, $T_p = 15$ years.

The levels of stress amplitudes caused by the action of loading and unloading forces were determined from formula:

$$\sigma_{ai}^{\mathrm{III}} = \frac{\sigma_{static}}{2}. \tag{12}$$

Table 5

Static distribution of the impact force amplitudes

Force interval	Cycle amplitude F_{ai}^y, MPa	Frequency, p_i	Number of cycles per year
$0.1 \div 0.4$	0.25	0.1258	2565
$0.4 \div 0.8$	0.6	0.2852	5818
$0.8 \div 1.2$	1	0.2802	5716
$1.2 \div 1.6$	1.4	0.1832	3737
$1.6 \div 2.0$	1.8	0.0772	1569
$2.0 \div 2.4$	2.2	0.0359	586
$2.4 \div 2.8$	2.6	0.0098	163
$2.8 \div 3.2$	3	0.0023	39
$3.2 \div 3.6$	3.4	0.0003	5
$3.6 \div 4.0$	3.8	0.0001	2
–	–	$\Sigma = 1$	$\Sigma = 20200$

The total number of loading and unloading cycles over the estimated service time is:

$$N_c^{\mathrm{III}} = N_{l.unl.} \cdot T_p, \tag{13}$$

where $N_{l.unl.}$ is the number of loading and unloading forces per year, $N_{l.unl.} = 100$.

Substituting σ_{ai}^{I}, $\sigma_{ai}^{\mathrm{II}}$ and $\sigma_{ai}^{\mathrm{III}}$ into the formula, we shall obtain:

$$\sigma_{a,e} = \sqrt[m]{\frac{N_c^{\mathrm{I}}}{N_0} \sum_{i=1}^{n} \left(\bar{k}_{qbi} \cdot \sigma_{static} \right)^m p_i + \frac{N_c^{\mathrm{II}}}{N_0} \sum_{i=1}^{n} \left(\frac{\sigma_{i.f.}}{N} \cdot F_{ai}^y \right)^m p_i + \frac{N_c^{\mathrm{III}}}{N_0} \sum_{i=1}^{n} \left(\frac{\sigma_{static}}{2} \right)^m p_i}. \tag{14}$$

Results of calculation of stress amplitudes and fatigue resistance safety factor n are given in Table 6.

Table 6

Results of calculation of stress amplitudes and fatigue resistance safety factor n

Area	In the center girder between a pivot beam and a slab
$\sigma_{a,e}$	8.9
$\sigma_{a,N}$	104
n	11.6

It is accepted that the minimal permissible safety factor for fatigue resistance of wagon structures is $[n] = 1.8$.

5. Discussion of results of experimental study into the theory of implementation of a coupled design of the center girder for hopper wagons for iron ore pellets

Calculation results have shown that the condition for strength $n \geq [n]$ for the area of a center girder between a pivot beam and a slab is satisfied. Fatigue resistance evaluation results have demonstrated that the safety factors of fatigue of the most stressed section of a frame exceed permissible fatigue safety factor of $[n] = 1.8$. Strength in the center girder node of a frame of the wagon of model 20-9749, equipped for

the transportation of hot iron ore pellets, is sufficient over the entire estimated lifetime of wagon of 15 years.

Based on the results of theoretical studies, we have confirmed the possibility of using a coupled variant of beams in the center girder of hopper wagons for iron ore pellets. Consequently, the proposed technical solutions have been patented and practically detailed in [13].

At the next stage of research, we conducted full-scale tests of the created physical model of hopper wagons for iron ore pellets (Fig. 6). In this case, we employed methods for determining strength characteristics of the design using the technique of tensometry. Thus, Fig. 7 shows a layout of arranging strain gauges on the elements of body of the pilot hopper wagon for iron ore pellets.

Fig. 6. Wagon for transporting iron ore pellets and agglomerate, model 20-9749
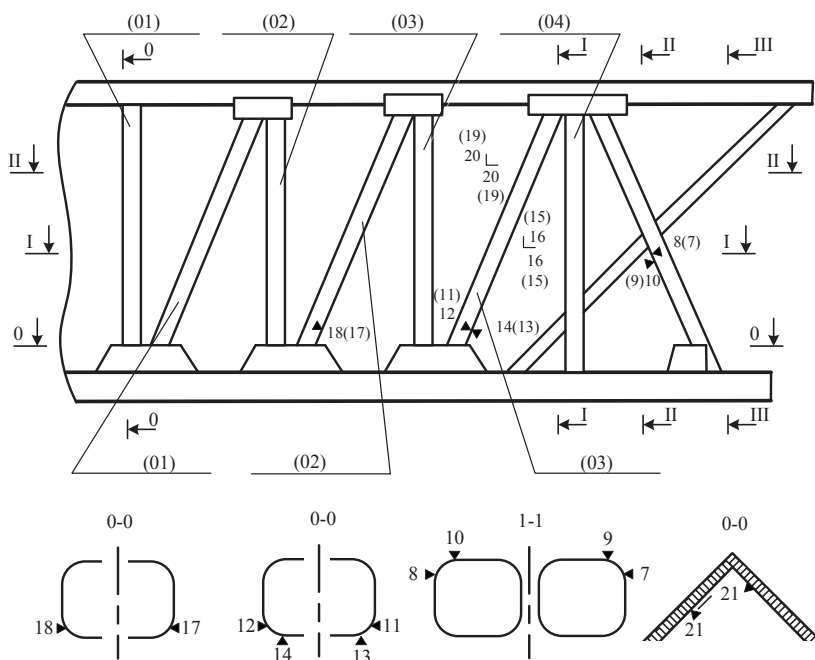

Fig. 7. Layout of arranging strain gauges on the body of a hopper wagon for iron ore pellets

Experimental studies fully confirmed correctness of the performed theoretical calculations and the possibility of industrial implementation of the proposed improvement in a center girder in the framework of a comprehensive (since it includes a number of other designed features) modernization.

5. Conclusions

1. The developed engineering-technological solutions of the coupled (instead of the profile solid along the length, it consists of two parts connected by welding) design of a center girder for hopper wagons for iron ore pellets are efficient and their application is feasible for industrial production.

2. The modified technique (taking into account the coupled design of beams) for calculating the loads acting on the improved structure allowed us to determine with sufficient accuracy their magnitudes, as confirmed by the results of a full-scale experiment.

3. The spatial finite-element computer model of the improved design of hopper wagons for iron ore pellets that we created is adequate (discrepancy between the magnitudes of strength stresses in the structure, obtained on its basis, and those derived experimentally, does not exceed 10 %) and could be used in further scientific-research and design-engineering work. The applied methodology of simulation computer modeling of different estimated cases (it is a combination of loads applied to the structure in different operational situations) in the work on acceptance of loads by a wagon design is effective, which is confirmed by sufficient convergence (the difference does not exceed 7.5 %) between the estimated data and those actually obtained.

4. Results of analysis of the conducted experiment for determining the rated resource of the improved hopper wagons for iron ore pellets allow us to argue about the relevance of the proposed technique and fulfillment of design conditions required for the operation. Thus, we revealed a sufficient agreement between the theoretically derived characteristics and their experimental values. Such indicators include the following: maximal stress at the level of 110 MPa; safety factor of fatigue resistance is 11.6; fatigue limit (by amplitude) for the control area at a symmetrical cycle and established loading mode with a basic number of cycles $N_0 = 10^7$ is equal to 104; the amplitude of dynamic stress of conditional symmetrical cycle is 8.9; all of which will ensure a 15-year-long operation time.

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