The object of this study is the processes related to the emergence, perception, and redistribution of loads in the improved structure of a passenger car frame. The scientific and applied task tackled in this paper is to ensure the strength of the supporting structure of a passenger car under operating loads. In this regard, it is proposed to improve the frame of a passenger car by constructing a girder beam from two rectangular pipes filled with material with energy-absorbing properties. The regularities of the frame load have been determined by taking into consideration the proposed solutions. It was found that the maximum equivalent stresses in the frame, taking its improvement into account, are 11.2 % lower than in the structure without filler, and 11.7 % lower than in the typical design. The results reported here are explained by the fact that the use of rectangular pipes filled with energy-absorbing material contributes to an increase in the moment of resistance of the frame, and, accordingly, reduces stresses.

In addition, the study has determined the natural oscillation frequencies of the frame. The results of the calculation of the strength of the weld in the zone of interaction of the girder beam with the pivot beams are given.

A feature of the results obtained is that the improvement in the strength of the frame is achieved not by strengthening its components but reducing the load.

The scope of practical application of the reported results concerns railroad transportation, as well as other sectors of mechanical engineering. The conditions for the practical use of these findings are the introduction of closed profiles in the structure of vehicles at the stage of their design and modernization.

This study could help reduce the cost of maintaining passenger cars and improve the efficiency of their operation. In addition, the research might prove useful for designing modern railroad car structures

Keywords: girder beam, energy-absorbing filler, frame with filler, energy-absorbing frame concept

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DETERMINING FEATURES OF THE STRESSED STATE OF A PASSENGER CAR FRAME WITH AN ENERGY-ABSORBING MATERIAL IN THE GIRDER BEAM

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1. Introduction

One of the most important components of the successful functioning of the economy of European countries is the development of transport infrastructure. For a long time, rail transport has been the most priority and competitive component of the industry [1, 2]. To further maintain the position of the primacy of railroad transport in the segment of transportations, it is important to reform it. One of the key points, in this case, is the introduction of energy-saving technologies in railroad transport [3, 4], ensuring its environmental friendliness [5, 6], as well as the introduction of highly efficient rolling stock into operation [7–9]. These solutions should be implemented not only in relation to freight but also passenger rolling stock.

The most loaded component of the passenger car is the supporting structure [10, 11], which includes the body and frame. Depending on the type of passenger car, the frame can be made with or without a through girder beam. During operation, the frame of the passenger car is exposed to the constant effect of vertical and longitudinal loads, which are due to the operating modes of the car. The cyclical nature of the action of these loads causes damage to the frame, including the emergence of cracks, deformations, etc. The presence of such defects on the way of the train threatens not only the safety of its movement but also poses a threat to the lives of passengers. Therefore, to improve the strength of the supporting structures of passenger cars, it is important to devise measures aimed at their improvement.

2. Literature review and problem statement

The issues of improvements in the supporting structures of passenger cars at the stage of their design, as well as modernizations to ensure strength and increase operating efficiency are quite relevant. Thus, the features of improving the supporting structure of a passenger car are covered in [12]. A new design methodology for the components of the supporting structure of passenger cars is given, which meets the requirements for loads and production. However, this improvement applies to the supporting structure of a passenger car without a girder beam and is not appropriate for frame structures with a through girder beam.

Improvement of the supporting structure of the passenger car is considered in paper [13]. Options for reducing the weight of the supporting structure of a passenger car are proposed. The results of determining its strength are given. However, this advancement does not contribute to improving the strength indicators of the supporting structure of a passenger car but only reducing its sprung mass.

In [14], the analysis of modern designs of passenger cars is carried out. The requirements that they must meet in operation are determined. These requirements are possible to apply not only in the design of cars but also in their modernization. It is important to say that the authors did not indicate the need to improve the supporting structures of cars to improve the efficiency of their operation.

In [15], the concept of improving the supporting structures of railroad vehicles using standard thin-walled steel profiles is considered. Examples of the application of the proposed concept are given. The expediency of its use in the production of modern vehicles is indicated. However, the application of the proposed concept in relation to the supporting structure of the passenger car was not considered in the cited work.

Features of improving the frame of a railroad car are covered in [16]. The results of determining the stressed state of the frame structure under the main operating modes inherent in the 1435 mm track are given. The effectiveness of the proposed improvement has been proven. At the same time, the research has been conducted on the frame of the freight car.

To ensure the strength of passenger cars during operation in railroad-water transportation, work [17] investigates their loading during transportation by railroad ferry. To that end, a mathematical model was built that makes it possible to determine the dynamic loads acting on its supporting structure. To verify it, the authors conducted a computer simulation of the dynamic load of the supporting structure of a passenger car. However, in the cited article, the authors limited themselves to determining the loads that act on a passenger car when transported by sea. That is, the work does not propose solutions for improving the supporting structures of passenger cars to improve their strength in operation, including when transporting on railroad ferries in international traffic.

Paper [18], which considers the issue of improving the passenger car by optimizing its design, is of scientific interest. The study was conducted in the environment of the ANSYS (ADPL) software package. The authors provide a comparable analysis of the design of a passenger car before and after improvement. The application of the proposed optimization procedure in the implementation of the optimal design of the supporting structure of a passenger car is substantiated. It is important to note that at the same time, no attention is paid to the issue of improving the frame of a passenger car as the most loaded structural unit in operation.

In [19], the features of the improved supporting structure of the passenger car are highlighted. The purpose of improvement was to reduce the mass of the passenger car while ensuring the conditions of strength of the components of its supporting structure. To justify the proposed solutions, calculations for strength were carried out. In this case, regulatory standards of Japan and Europe were applied. The results of the calculations confirmed the feasibility of the proposed improvement. At the same time, the considered solutions do not contribute to improving the strength indicators of the supporting structure of a passenger car.

Measures to improve the strength of the supporting structures of passenger cars are considered in [20]. In this case, the improvement of the strength of the body is achieved not by improving it but by using elastic spring devices, that is, by increasing the oscillation frequencies. It must be said that in the case of rearrangement of the car from the bogies of one track to another, which takes place during its operation in international traffic, such measures lose their expediency. Therefore, the issue of improving the supporting structure of a passenger car to ensure its strength in operation is more rational.

To improve the strength of the passenger car body by reducing its workload, work [21] proposed a new concept. In this case, the authors propose the use of panels that are built according to the «sandwich» type and welded together. In addition, the work proposed optimization of the design of the shape of the edge and the thickness of the panel. The studies were conducted on the example of a passenger car. The results of the calculations proved the feasibility of this solution. At the same time, the proposed optimization concerns the body. That is, the authors did not consider the possibility of optimizing the frame of a passenger car.

Our review of literary sources [12–21] makes it possible to conclude that the issues of improvements in the supporting structures of passenger cars require further research to improve the efficiency of their operation. This is proved by the lack of technical solutions aimed at improving the strength of their supporting structures by reducing the dynamic load during operation.

3. The aim and objectives of the study

The aim of this study is to improve the frame of a passenger car by constructing a girder beam from closed profiles filled with energy-absorbing material. This will help improve the strength of the supporting structures of passenger cars, reduce maintenance costs, and increase operating efficiency.

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

– to investigate the strength of the typical design of the passenger car frame;

– to determine the regularities of the load of the passenger car frame, taking into consideration its improvement;

– to determine the strength of the weld in the zone of interaction of the girder beam with the pivot beam.

4. The study materials and methods

The object of this study is the processes related to the emergence, perception, and redistribution of loads in the improved design of the passenger car frame.

The main hypothesis of the study assumes that improving the strength of the passenger car frame is possible by reducing the load under operating modes. This can be achieved by using fillers with energy-absorbing properties in the most loaded components of the frame [22, 23].

To determine the most loaded components of a typical frame design, a strength calculation was carried out. For this purpose, its spatial model was built (Fig. 1). Graphic work was conducted using the software package SolidWorks (France).

Fig. 1. Spatial model of the passenger car frame

As a prototype, a passenger non-compartment car of model 61–821 was chosen. The main technical characteristics of the car are given in Table 1.

Table 1

Passenger non-compartment car 61–821

Parameter name	Value
Tare, t	51
Length, mm	24537
Width, mm	3105
Height, mm	4377
Speed of movement, km/h	160
Dimensions	0-T

When constructing the frame model, structural elements that interact rigidly with each other are taken into consideration. In this case, the model does not take into consideration the welding seams between the individual components, that is, it is monolithic. To determine the main indicators of frame strength, the finite element method was used, which is implemented in the SolidWorks Simulation software package (France) [24–26]. This method was chosen as an estimation method because it is the most used to determine the strength of cars at car-building enterprises. An alternative method, in particular the method of forces, can currently be used at the design stage. For example, to determine the performance profiles of the components of the car. This method does not make it possible to investigate the volumetric stressed state of the object and is rather cumbersome regarding its use in railroad car structures.

The calculation was carried out according to the Mieses criterion [27–29]. It is known that this criterion is based on the theory of shape change energy.

In this case, the maximum equivalent stresses are determined by

$$\sigma_{eq} = \sqrt{\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}}, \qquad (1)$$

where σ_1, σ_2, σ_3 are the main stresses, MPa.

The finite-element frame model is formed by spatial isoparametric tetrahedrons [30–32]. The optimal number of finite elements is determined using the options of the software package. There are other methods that make it possible to do this, for example, graph-analytical [33, 34]. However, it requires multivariate calculations. In this case, the finite element model has 15231 elements and 5573 nodes. The maximum size of the element was 175.1 mm, the minimum – 35.0 mm. The maximum aspect ratio was 296.41. It must be said that the percentage of such elements is insignificant. The number of elements in the circle is 9. The ratio of the increase in the size of the elements is 1.6.

When building the estimation scheme of the frame, it is taken into consideration that it is affected by the vertical load P_v, including the vertical static P_v^{st} and dynamic P_v^{d} component, and the longitudinal load P_l, applied to the front (jerk, stretching) or rear (impact-compression) stops of the automatic coupling (Fig. 2).

The dynamic component of the vertical load was determined in accordance with DSTU 7774:2015. Passenger cars of main locomotive traction. General technical standards for the calculation and design of mechanical parts of cars. That is,

$$P_v^d = P_v^{st} \cdot k_{dv}, \qquad (2)$$

where k_{dv} is the coefficient of vertical dynamics.

The coefficient of vertical dynamics k_{dv} is considered as a random function with a probable distribution of the form:

$$P(k_{dv}) = 1-\exp\left(-\frac{\pi}{4}\cdot\frac{k_{dv}^2}{\overline{k_{dv}^2}}\cdot\beta^2\right), \qquad (3)$$

where $\overline{k_{dv}}$ is the average probable value of the coefficient of vertical dynamics; β – distribution parameter, which is determined by experimental data. The value of this parameter under existing operating conditions can be taken as β=1.

The coefficient k_{dv} is defined as the quantile of expression (3) at the calculated one-way probability $P(k_{dv})$:

$$k_{dv} = \frac{\overline{k_{dv}}}{\beta}\sqrt{\frac{4}{\pi}\cdot\ln\frac{1}{1-P(k_{dv})}}. \qquad (4)$$

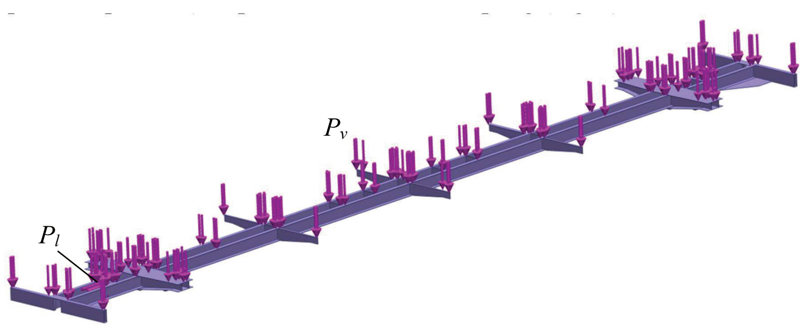


Fig. 2. Estimation scheme of the frame

When performing calculations under mode III, lateral loads, in particular centrifugal and wind, were also taken into consideration.

The frame was fastened in the areas of its support on the chassis. The structural material is the low-alloy structural steel of grade 09G2S. It must be said that this steel grade can also be used in supporting structures of narrow-gauge passenger cars. In addition, other materials can be used in the supporting structures of such cars [35, 36].

To reduce the load of the passenger car frame, it is proposed to manufacture the most loaded element of its design, namely the girder beam, from rectangular pipes (Fig. 3), filled with material with energy-absorbing properties.

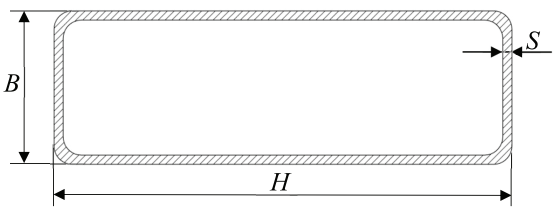


Fig. 3. Cross-section of a rectangular pipe: *H* – cross-sectional height; B – cross-sectional width; *S* – cross-sectional wall thickness

In order to determine the optimal, from the point of view of the minimum material intensity, profile of the girder beam, the method of optimization by strength reserves was applied [37, 38]. In this case, the objective function of optimization is to minimize the material intensity of the profile of the girder beam. The limitation of the problem is to maintain the maximum equivalent stresses in the girder beam within the permissible limits.

To determine the strength of the improved frame design, a calculation was performed using the finite element method. When assembling the finite element model of the frame, the maximum size of the element was 175 mm, the minimum – 35 mm. The number of nodes of the model is 6060, and the elements – 18769. The maximum aspect ratio is 387.93. At the same time, the percentage of such elements is quite insignificant. The number of elements in the circle is 9. The ratio of the increase in the size of the elements is 1.7.

To reduce the load of the frame under operational modes, the use of energy-absorbing material in the girder beam is proposed. As an example, the calculation was carried out for the case of using foam aluminum as such material. To simulate foam aluminum, rectangular elements were installed in the girder beam, which have the characteristics inherent in foam aluminum. It is assumed that this material is isotropic. This assumption is due to the fact that at this stage of the study the task is to justify the use of a filler with energy-absorbing properties in the frame. That is, this solution is considered as a concept. It is taken into consideration that foam aluminum has the following characteristics: modulus of elasticity – $5.3 \cdot 10^9$ Pa, Poisson coefficient – 0.3, mass density – 800 kg/m^3, tensile strength – $5.0 \cdot 10^6$ Pa, yield strength – $1.05 \cdot 10^6$ Pa.

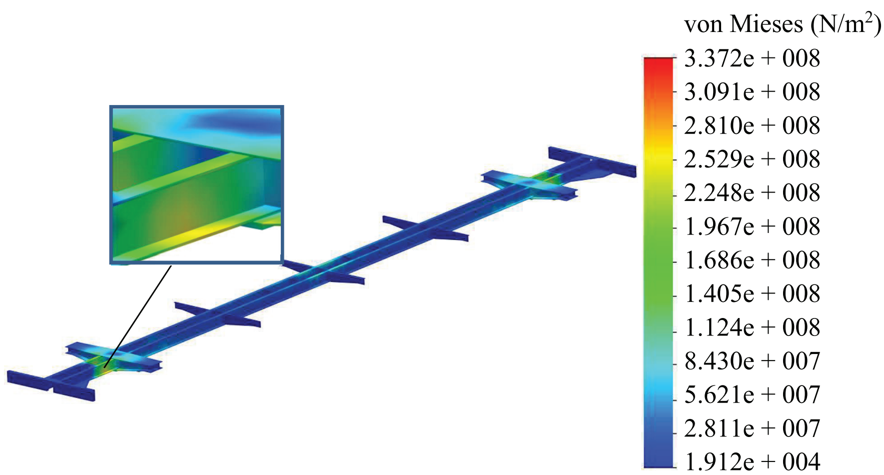
The finite-element model of the frame with foam aluminum in the girder beam has 18200 elements and 4828 nodes. The maximum size of the element was 210 mm, the minimum – 42 mm. The maximum aspect ratio was 933.93. It is important to say that the percentage of such elements is insignificant. The number of elements in the circle is 9. The ratio of the increase in the size of the elements is 1.7.

To determine the strength of welding seams in the most loaded areas of the frame, the classical method of material resistance was used [39, 40].

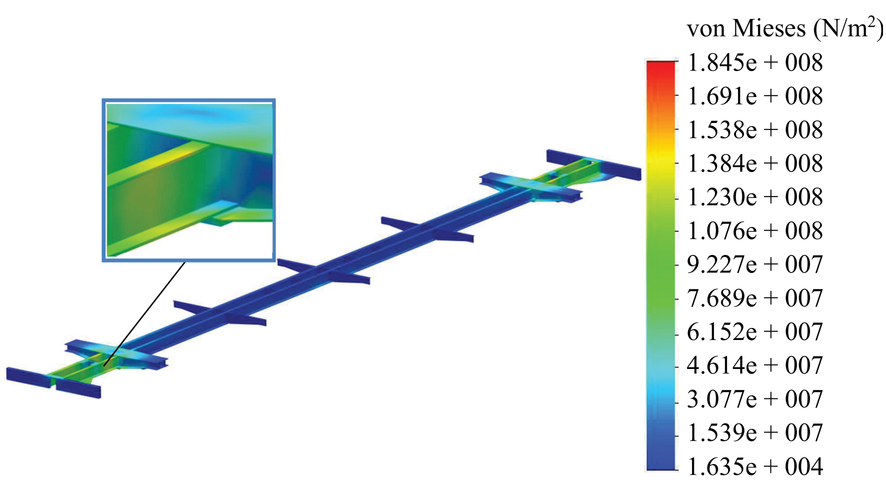
5. Results of studying the stressed state of the frame with energy-absorbing material in the girder beam

5. 1. Studying the strength of a typical passenger car frame structure

The results of the calculation of the frame under mode I "impact-compression", as well as "jerk", are shown in Fig. 4.



a



b

Fig. 4. Stressed state of the frame: *a* – "impact-compression"; *b* – "jerk"

The maximum equivalent stresses are registered in the girder beam in areas located closer to pivot beams (Fig. 5). In this regard, this element of the frame is selected for improvement.

With «shock-compression», the maximum equivalent stresses were 337.2 MPa, and with a «jerk» – 184.5 MPa.

The results of calculating the frame for strength under other estimation modes are given in Table 2.

Table 2

The results of the calculation of the frame for strength

Estimation mode	Maximal equivalent stresses
Estimation mode I	
Impact-compression	337.2
Jerk	283.5
Stretching	264.3
Estimation mode III	
Impact-compression	184.5
Jerk-Stretch	168.6

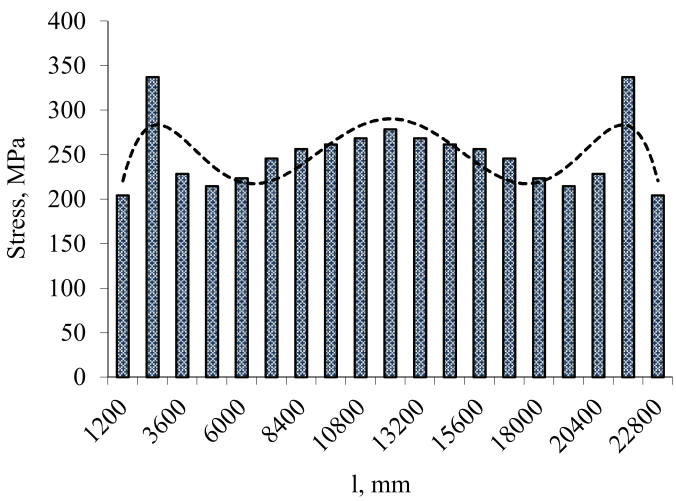


Fig. 5. Distribution of stresses by the length of a typical girder beam

The results that are given in Table 2 show that the received stresses are lower than permissible (DSTU 7774: 2015. Passenger cars of main locomotive traction. general technical standards for the calculation and design of mechanical parts of cars). An international analog of the specified standard is EN 12663-1:2010, Railroad applications – Structural requirements of railroad vehicle bodies – Part 1: Locomotives and passenger rolling stock (and alternative method for freight wagons).

The permissible value of stresses underestimation mode I is assumed to be 345 MPa, that is, the yield strength of the material, and under mode III – 190 MPa.

At the same time, under the conditions of cyclic alternating or excess loads on the frame, the stresses in it may exceed the permissible values. This circumstance necessitates devising measures aimed at improving the strength of the supporting structure of a passenger car.

5. 2. Determining loading patterns in the passenger car frame taking into consideration its improvement

To determine the optimal parameters of the profiles for the girder beam of the frame, the calculation was carried out. The results of the calculations are given in Table 3.

In this case, Table 3 contains the following designations: σ_{eq} – maximum equivalent stresses; W_i – the moment of cross-sectional resistance relative to the corresponding axis; $[W_i]$ – the permissible moment of cross-sectional resistance relative to the corresponding axis.

Analyzing the data given in Table 3, it should be noted that the determined parameters of the profile of the girder beam are not rational from a technological point of view, namely the placement and fastening of elements of an automatic coupling device. The height of the profile must be at least the typical one. In this case, it is advisable to use a rectangular pipe with the parameters given in Table 4 (DSTU 8940:2019 Profile steel pipes. Specifications). There are international analogs of this standard, but the dimensions of the profiles indicated in them may differ slightly compared to the above standard. For example, in the standard “BS EN 10219 – Cold Formed Welded Structural Hollow Sections of Non-alloy and Fine Grain Steels”, the closest to the selected profile is a pipe with a height-to-width ratio of 300/200 mm. Therefore, when using such standards as basic, it is necessary to take this into consideration.

At the same time, the mass of such a pipe is higher than the typical profile of the girder beam. Therefore, it is proposed to use this pipe taking into consideration the reduction in the thickness of its walls while ensuring the conditions of strength.

To determine the optimal, from the point of view of the minimum mass, pipe thickness while ensuring the conditions of its strength, variational calculations were carried out, the results of which are shown in Fig. 6. In this case, $\sigma=f(t)$ indicates the dependence of stresses on the wall thickness, and $m=f(t)$ – the dependence of the mass of 1 m of the pipe on the wall thickness. From Fig. 6, it can be seen that the optimal wall thickness is 4.5 mm. At the same time, its mass is 27.45 kg, and the moments of resistance relative to the X and Y axes, respectively, are W_x=667.58 cm^3 and W_y=3796.88 cm^3. It is important to say that taking into consideration the proposed solutions for improvement, the frame weight is reduced by 4.6 % compared to the typical design.

To substantiate the proposed solutions, a calculation was performed on the strength of the car frame. At the first stage, the strength of the frame without filler in it was determined. The spatial model of the frame is shown in Fig. 7.

The calculation of strength was carried out according to the scheme shown in Fig. 2. The results of the calculation are illustrated in Fig. 8.

Table 3

The results of determining the parameters of the optimal profile for the girder beam

σ_{eq}, MPa	W_x, cm^3	W_y, cm^3	$[W_x]$, cm^3	$[W_y]$, cm^3	Pipe optimal parameter				Mass of 1 m of pipe, *kg*
					W_x, cm^3	W_y, cm^3	H/B, mm	S, mm	
337.2	23.97	134.98	23.97	134.98	150.98	139.72	150/130	7.0	27.91

Table 4

Parameters of a rectangular pipe for the manufacture of a girder beam

W_x, cm^3	W_y, cm^3	H/B, mm	S, mm	Mass of 1 m of pipe, *kg*
318.45	168.47	300/100	6.0	35.82

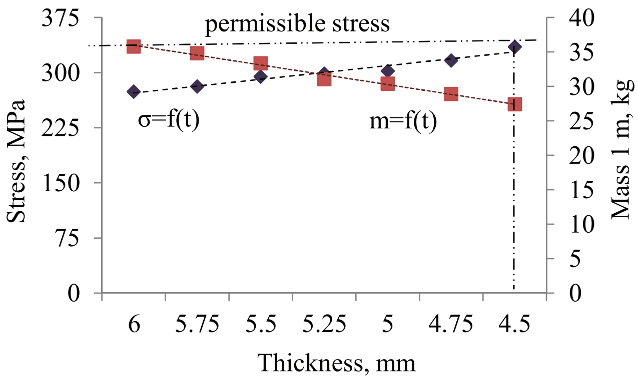


Fig. 6. The dependence of stresses in the girder beam on its parameters

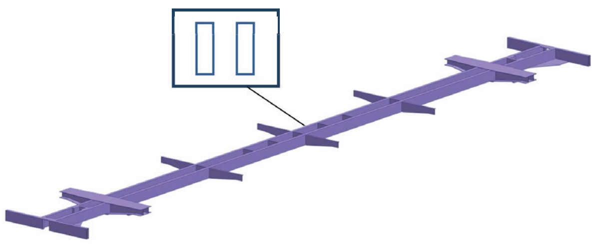


Fig. 7. Spatial frame model

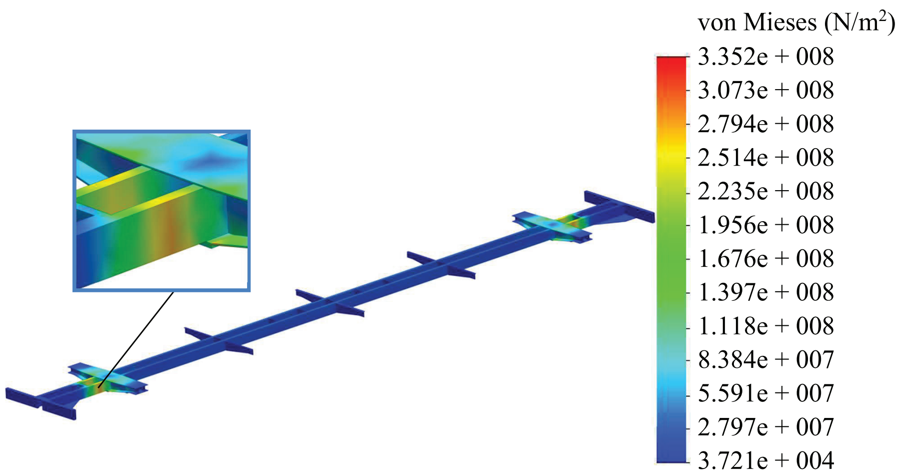


Fig. 8. Stressed state of the frame

The calculation of strength was carried out in relation to other modes. The results of the calculation are given in Table 5.

Table 5

Results of calculation on the strength of the frame with a girder beam of two rectangular pipes

Estimation mode	Maximal equivalent stresses, MPa
Estimation mode I	
Impact-compression	335.5
Jerk	282.1
Stretching	259.4
Estimation mode III	
Impact-compression	179.6
Jerk-Stretch	161.1

Consequently, the strength of the frame with a girder beam of two rectangular pipes is ensured.

The maximum equivalent stresses are registered in the middle part of the pipes in the areas between the rear stops of the couplings and the pivot beams. However, these stress values do not exceed the permissible ones.

At the next stage of the study, the strength of the frame was determined, taking into consideration the presence of energy-absorbing material in the girder beam.

The results of the calculation of the strength of the frame are shown in Fig. 9. In this case, the maximum equivalent stresses are 297.6 MPa, which is 11.2 % lower than that of a structure without energy-absorbing material and 11.7 % lower than in a typical frame design. The safety margin of the frame is n=1.14, which is 12.3 % higher than in a typical design.

The distribution of stresses along the length of the girder beam of the improved design is shown in Fig. 10. Maximum stresses occur behind the rear stops of the auto couplings, and then their decrease is observed.

The calculation of the strength of the frame with energy-absorbing material in the girder beam was carried out in relation to other estimation modes. The results of the calculation are given in Table 6.

Table 6

The results of the calculation of the strength of the frame with energy-absorbing material in the girder beam

Estimation mode	Maximal equivalent stresses, MPa
Estimation mode I	
Impact-compression	297.6
Jerk	252.5
Stretching	229.8
Estimation mode III	
Impact-compression	158.4
Jerk–Stretch	142.7

Therefore, the strength of the frame, taking into consideration the proposed solutions for improvement, is ensured.

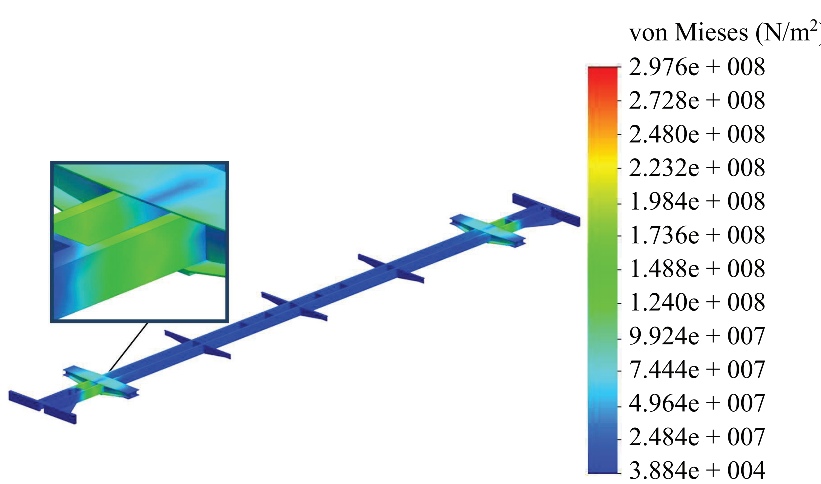


Fig. 9. Stressed state of the frame

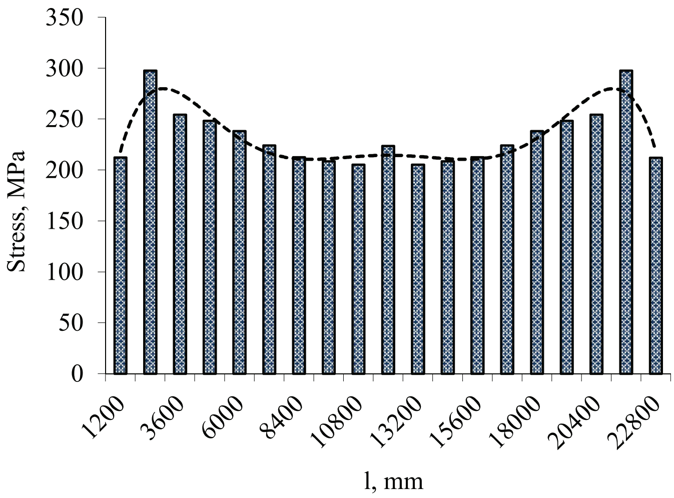


Fig. 10. Distribution of stresses along the length of the improved girder beam

Due to the fact that in this work the frame of the passenger car has been improved, an important stage of the study is to conduct a modal analysis. The purpose of the corresponding calculation is to check the absence of resonance of the car during operation, as well as to ensure the necessary smoothness. In this case, the first oscillation frequency of the supporting structure of the car, in accordance with the above standards, must have a value of at least 8 Hz.

For the modal analysis, we used the estimation scheme shown in Fig. 2. The results of the calculation are given in Table 7.

Table 7

The value of the natural frequencies of oscillations of the car frame

Mode number	Frequency value, Hz
1	24.1
2	25.9
3	59.5
4	59.7
5	60.6
6	65.5
7	69.6
8	79.6
9	114.9
10	122.85

Analyzing the data given in Table 7, we can say that the safety of the movement of the car is ensured since the value of the first natural frequency has a value of 24.1 Hz. The resulting frequency value is more than 2.5 times higher than that inherent in the typical design and is 9.2 Hz.

5. 3. Determining the strength of the weld in the zone of interaction of the girder beam with the pivot beam

Since one of the most loaded areas of the frame during operation is the zone of interaction of the girder beam with the pivot beam, we also calculated the strength of the weld in the places of their interaction. Such a calculation is a critical component of the study since taking into consideration the use of rectangular pipes, the length of the weld in comparison with the typical profile is reduced by more than 140 mm.

It is taken into consideration that the welding seam is exposed to the deformations of "stretching-compression" and "bending". The condition for the strength of the weld, in this case, takes the form:

$$\sigma_S = \frac{3 \cdot M}{(\beta \cdot h_S) \cdot l_S^2} + \frac{N}{F} \le R_Y^S, \tag{5}$$

where M is the bending momentum acting in the cross-section of the seam; $\beta{\cdot}h_S$ – calculated thickness of the seam; β – coefficient of weld boiling depth; l_S – estimated length of the seam; N – calculated force acting on the connection; F – connection area; R_Y^S – calculated resistance of the seam.

It is important to say that the body of the automatic coupling may have deviations in the vertical and horizontal planes when transmitting the longitudinal load on the car. In this regard, it is proposed, in addition to the above deformations, to also take into consideration the tangential stresses from the deformation of the "cut", then the condition of strength will be as follows:

$$\sum \sigma_S = \frac{3 \cdot M}{(\beta \cdot h_S) \cdot l_S^2} + \frac{N}{F} + \frac{Q}{F} \le R_Y^S, \tag{6}$$

where Q is the calculated transverse force acting on the welding seam.

In this case,

$$Q = N \cdot \cos\alpha, \tag{7}$$

where α is the angle of deviation of the auto-coupling housing in the horizontal plane.

Taking into consideration the calculations carried out at N=2500 kN, the value of $\Sigma\sigma_S$=1328 kg/cm^2 was obtained at $R_Y^S = 2000$ kg/cm^2. That is, the condition of strength is met. This also provides a margin of strength of the weld n=1.5. It is important to say that in a typical design the calculated value is n=1.41, which is 6 % lower than in the improved structure.

6. Discussion of results of studying the stressed state of the frame with energy-absorbing material in the girder beam

To reduce the load of the passenger car frame, it was proposed to improve it by constructing a girder beam from closed profiles filled with energy-absorbing material. Foam aluminum was used as an energy-absorbing material. Patterns of frame load are obtained, taking into consideration the proposed solutions. In this case, the maximum equivalent stresses amounted to 297.6 MPa, which is 11.2 % lower than that of a structure without energy-absorbing material and 11.7 % lower than in a typical frame design (Fig. 8, 9). The strength margin of the improved frame design is n=1.14, which is 12.3 % higher than in the typical structure.

Our results are explained by the fact that the use of rectangular pipes filled with energy-absorbing material contributes to an increase in the moment of resistance of the components of the frame, and, accordingly, reduces stresses.

The results of the modal analysis of the improved frame design showed that the value of the first natural oscillation frequency is 24.1 Hz (Table 7). This value is more than 2.5 times higher than that inherent in the typical frame design.

The strength of the weld in the zone of interaction of the girder beam with the pivot beam has been calculated. In this case, the strength margin of the weld was n=1.5, which is 6 % higher than in a typical structure. This is due to the use of energy-absorbing material in the girder beam, which reduces the loads perceived by the components of the frame.

The advantage of this study compared to [12] is that the proposed frame improvement can be applied both in the design with a through girder beam and not through (stretcher). In contrast to works [13, 18, 19], the improvement of the supporting structure of the car is aimed at improving its strength during operation, rather than reducing the tare. Regarding work [14], it is important to say that one of the key requirements for modern rolling stock is to ensure indicators of its strength during operation, which is not indicated in that paper. The advantage of the results obtained in comparison with studies [15, 16] is that they can be applied not only on passenger cars but also on freight cars. In contrast to work [17], this study not only determined the load of the passenger car frame but also proposed measures for improvement. Compared to work [20], an increase in the natural frequencies of the supporting structure is achieved by improving the frame, rather than using bogies with improved dynamic characteristics. In contrast to [21], the improvement in the strength of the supporting structure of the car is achieved by introducing energy-absorbing material into the frame as the most loaded unit, and not into the side walls.

As a disadvantage of the study, it can be noted that the proposed solutions for improvement are implemented in relation to passenger cars with a girder beam. That is, for those designs of passenger cars, the frame of which is not equipped with a girder beam, it is advisable to consider the possibility of improving subframes.

The limitation of this study is that the strength of the improved girder beam is determined at the standard values of operational loads. When considering excess load modes, it is necessary to take into consideration other characteristics of the energy-absorbing material.

It is important to say that the limitation of the computer models formed as part of the study is that they do not take into consideration the elasticity of the spring suspension of the bogies on which the car rests. In subsequent studies, this point will be taken into consideration.

A further stage of research in this area is to determine the load of the supporting structure of a passenger car with angular movements. In addition, the issues of experimental determining of the strength of the frame of a passenger car with energy-absorbing material in the girder beam require attention. Such studies are planned to be carried out in the laboratory. In this case, it is possible to create a reduced sample of the frame and determine its strength using the method of electrical strain monitoring. This will make it possible to verify the formed models of load of the supporting structure of the car, and, if necessary, clarify them. At the same time, difficulties may arise due to errors that are inherent in the method of similarity. In this case, it is possible to use alternative methods of experimental research.

Our studies will help reduce the cost of maintaining passenger cars and increase the efficiency of their operation. In addition, the results of research can be useful developments in the design of modern competitive structures of passenger cars.

7. Conclusions

1. The strength of the typical design of the passenger car frame has been investigated. The maximum equivalent stresses are registered in the girder beam in areas located closer to the pivot beams. With “impact-compression”, the maximum equivalent stresses were 337.2 MPa, and with a “jerk” – 184.5 MPa. At the same time, under the conditions of cyclic alternating loads, the stresses in the frame may exceed the permissible values.

2. The regularities of the load of the passenger car frame have been determined, taking into consideration its improvement. The results of the calculation of the strength of the frame with a girder beam of rectangular pipes showed that the maximum equivalent stresses are 335.5 MPa. The resulting value of stresses is lower than the permissible ones by almost 3 %. At the same time, these stresses are almost equal to those that occur in a typical structure. Therefore, it is advisable to fill in the profiles of the girder beam with energy-absorbing material.

The maximum equivalent stresses in the frame, taking into consideration the use of energy-absorbing material in the girder beam, were 297.6 MPa. The resulting stress value is 11.2 % lower than that of a structure without energy-absorbing material and 11.7 % lower than in a typical frame design. The strength margin of the frame is n=1.14, which is 12.3 % higher than in a typical structure.

A modal analysis was carried out and the natural oscillation frequencies of the car frame were determined. In this case, the value of the first natural frequency of oscillations is more than 8 Hz. The resulting frequency value is more than 2.5 times higher than that inherent in a typical frame design.

3. The strength of the weld in the zone of interaction of the girder beam with the pivot beam has been determined. The calculation was conducted on the condition that the welding seam is exposed to the deformations of “stretching-compression”, “bending”, and “cutting”. It was established that with a longitudinal load of 2500 kN on the rear stop of the automatic coupling, the strength of the welding seam is observed. The margin of strength of the welding seam is n=1.5, which is 6 % higher than in the typical structure.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

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