

*This paper reports a study into the dynamic loading and strength of an improved carrying structure of the platform car. A special feature of the car is the presence of elastic elements in the main longitudinal beams of the frame. This could improve the fatigue strength of the carrying structure of the platform car under operational modes.*

*Mathematical modeling was carried out to determine the dynamic load on the carrying structure of the platform car. The fluctuations of bouncing were taken into consideration. It has been established that the vertical acceleration of the carrying structure of the platform car without cargo is about  $2.0 \text{ m/s}^2$  (0.2 g). In this case, the vertical accelerations that act on the carrying structure of the platform car are lower by 15 % as compared to the prototype wagon.*

*The main indicators of strength of the bearing structure of the platform car have been determined. The calculation was carried out using a method of finite elements employing the software suite SolidWorks Simulation (France). When compiling the estimation scheme, it was taken into consideration that the carrying structure of the platform car was loaded with four containers the size of 1SS. In this case, the maximum equivalent stresses occur in the region of interaction between a pivot beam and spreads and are 254.0 MPa. That warrants the strength of the carrying structure of the platform car.*

*The numerical values of acceleration of the carrying structure of the platform car have been determined, as well as the fields of their location, by applying computer simulation.*

*The fatigue strength and oscillation eigenfrequencies in the carrying structure of the platform car have been investigated; their designed service time has been estimated.*

*The reported research would contribute to building innovative structures of platform cars, as well as to improving the efficiency of combined transportation*

*Keywords: platform car, carrying structure, dynamic load, structural strength, fatigue strength, transport mechanics, rail transport*

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# DETERMINING THE LOAD ON THE LONG-BASED STRUCTURE OF THE PLATFORM CAR WITH ELASTIC ELEMENTS IN LONGITUDINAL BEAMS

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## 1. Introduction

The defining direction of economic development of European countries is primarily the successful functioning of the transport industry. Its leading component is railroad transport. The modern railroad network of Europe ensures the coordinated and uninterrupted interaction between all its countries. It is important to note that in order to maintain the leadership positions of railroad transport, it is important to provide the transportation industry with modern highly efficient rolling stock.

It is known that the most popular type of rolling stock in the international connection is platform cars. This type of

wagon is used for the transportation of cargoes that do not require protection from precipitation. Mostly in international traffic, these cars are used for the transportation of containers or tank containers. The main bearing element of the platform car is the frame, which is a welded structure.

To improve the efficiency of platform car operation, long-frame structures that can carry four 20-foot containers at the same time have been commonly used. At the same time, such structures of platform cars are exposed to significant vertical loads due to the pliability of the structure. This causes the accumulation of fatigue stresses in the carrying structure and may contribute to its damage.

In this regard, it is important to introduce new alternative solutions aimed at reducing dynamic load and improving the fatigue strength of the bearing structures of wagons. One of the possible options for reducing the dynamic load on wagons, in particular long-frame ones, is the use of multifunctional and multi-material principles in their design. These principles have proven efficient in various fields of mechanical engineering. However, up to now, they have not been widespread when designing railroad cars. Therefore, based on the positive experience of employing these principles, their integration into the railroad industry is relevant. This would contribute to building innovative structures of wagons with improved technology and economic indicators. That could also improve the competitiveness of the railroad industry and the efficiency of its operation.

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## 2. Literature review and problem statement

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Paper [1] shows the peculiarities of testing the carrying structure of the platform car for twisting and bending. The research was conducted experimentally. To determine the deflection of the carrying structure of the platform car, a method of electrical strain measurement was used. However, the set of tests reported by the authors does not highlight the peculiarities of the calculation of the carrying structure of the platform car for fatigue strength. That might contribute to a significant error as regards the results.

The structural features of the next-generation freight car for intermodal transportation are highlighted in work [2]. The spatial model of the bearing structure of the platform car was designed using the programming package PTC/Creo (USA). The strength calculation employed the software ANSYS (USA) and ADAMS/Rail (USA).

Work [3] considers the structural features of a long-frame platform car for intermodal transportation. The authors reported the results of calculating the strength and dynamic load of the carrying structure of the platform car.

It should be noted that when designing the carrying structures of platform cars, the authors confined themselves to the rated values of loads acting on a wagon during operation. This may be because the gross weight of the projected car and the dynamic indicators of bogies are similar to the prototype wagons. However, for a more accurate assessment of the indicators of dynamics and strength, it is important to take into consideration the actual dynamic load at the stage of car design.

The modeling of the strength of the carrying structure of the platform car for intermodal transportation of various vehicles is reported in [4]. In this case, the calculations are based on PN-EN standards.

Study [5] addresses the design, analysis, and construction of the new-generation rail cars. The study was conducted for Indian Railroads. In this case, the authors applied international standards for different loading conditions, compliance with the optimal dimensions of wagons, etc. However, when designing these structures of cars, they did not take into consideration any measures to reduce fatigue strength under the operating modes of loading. This can be explained by the fact that the values of dynamic loads, underlying the estimated standards of countries-manufacturers of these cars, are lower than those for cars run on a wide track.

A study into the dynamic load of a car's bearing structure is reported in work [6]. The authors took into consideration the case of the use of different types of bogies under cars. They

analyzed the impact of the technical characteristics of bogies on the dynamic performance of the car. At the same time, the work did not study the impact of the technical characteristics of bogies on the fatigue strength of cars' carrying elements.

Paper [7] reports the results of modeling the strength of the carrying structure of a platform car. The car has a relegated middle part of the frame. The girder beam is used only in console parts. The calculation for strength involved a method of finite elements. The reported results were confirmed by experimental studies conducted at the Bulgarian National Research Institute. However, the paper does not indicate whether the fatigue strength of the load-bearing structure of the platform car is improved when taking into consideration the proposed technical solutions.

The improved carrying structure of the platform car, in order to operate it in the international combined transportation, is reported in works [8, 9]. Mathematical modeling was carried out to determine the refined values of dynamic loads. The resulting loads were taken into consideration when calculating the strength of the carrying structure of the platform car. However, the proposed measures to improve the carrying structure of the platform car do not contribute to the improvement of its fatigue strength.

The analysis of literary sources [1–9] allows us to conclude that the issue of improving the fatigue strength of long-base carrying structures of platform cars by introducing elastic elements in them has not been considered before. Therefore, it is important to conduct relevant research in this area.

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## 3. The aim and objectives of the study

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The aim of this study is to determine the loading on a long-frame structure of the platform car with elastic elements in longitudinal beams. This could make it possible to improve the fatigue strength of the long-base carrying structure of the platform car under the action of cyclical loads.

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

- to propose measures to improve the carrying structure of the platform car to increase fatigue strength during operation;
- to investigate the dynamic load on the carrying structure of the platform car;
- to investigate the strength of the carrying structure of the platform car;
- to investigate the fatigue strength and the oscillation eigenfrequencies of the carrying structure of the platform car.

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## 4. Measures to improve the carrying structure of the platform car to increase fatigue strength during operation

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To reduce the dynamic load on the bearing structure of a platform car and increase fatigue strength under operating modes, it is proposed to introduce elastic elements in it. The elastic elements are planned to be arranged in the longitudinal beams of the frame acting as its main carrying elements. To this end, it is proposed to replace the main longitudinal beams of the frame with U-shaped beams (Fig. 1).

Our study was conducted for the platform car of model 13-4024 made by PAT "KVBZ" (Kremenchuk, Ukraine). To this end, we constructed its spatial model in the programming package SolidWorks (Fig. 2).

The cross-section of the main longitudinal beam with elastic elements inside is shown in Fig. 3.

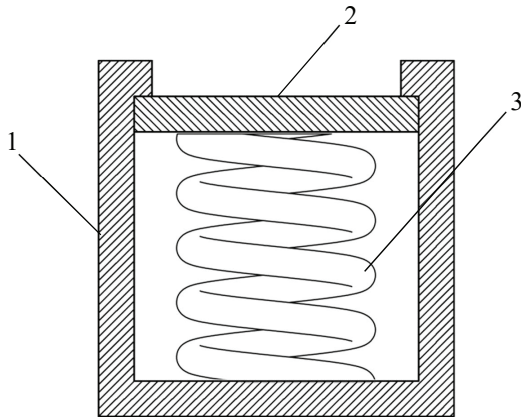


Fig. 1. Cross-section of the main longitudinal beam of the frame: 1 – U-shaped profile; 2 – horizontal bar; 3 – elastic element

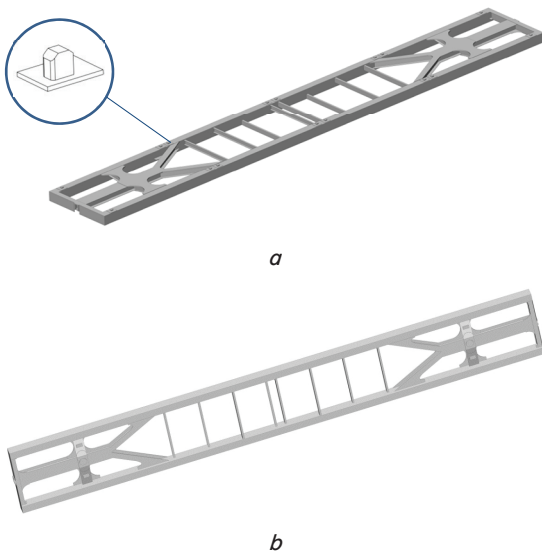


Fig. 2. Spatial model of the improved carrying structure of the platform car: *a* – view from above; *b* – view from below

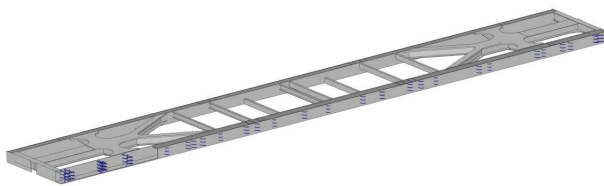


Fig. 3. Carrying structure of the platform car with elastic elements in the longitudinal beam

In this case, fitting stops are placed on the horizontal bar of the longitudinal beam (Fig. 1, *a*). We imply the possibility of using both stationary and folding fitting stops.

### 5. Studying the dynamic loading on the carrying structure of the platform car

Mathematical modeling was carried out to determine the dynamic load on the carrying structure of the platform car taking into consideration the proposed measures. The study

was performed in a flat coordinate system – the *XZ* plane. The estimation scheme of the car is shown in Fig. 4.

In this case, the platform car is considered as a system of three solids: the frame and two bogies, model 18–100, with spring kits that possess rigidity and a coefficient of relative friction.

It is assumed that the following connections are imposed on the system:

- the movement of the frame and bogies of the platform car along the track axis are the same:

$$q_1 = q_2 = q_3;$$

- the wheelsets move without slipping:

$$\psi_{ij} = x_{ij} / R, (i, j = 1, 2),$$

where *R* is the wheel radius, 0,475 m;

- given the lack of elastic elements in the journal suspension, the bouncing of bogies is determined by the bouncing of wheelsets.

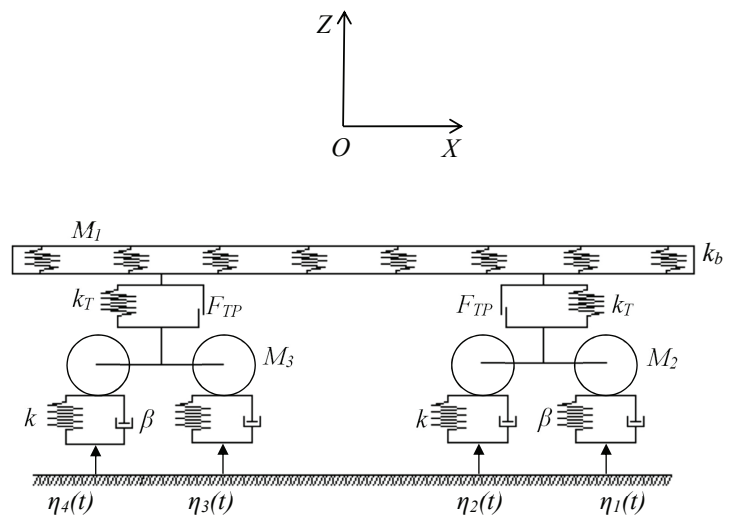


Fig. 4. Estimation scheme of the platform car

It is taken into consideration that the car is moving over a viscoelastic track [10]. In this case, it is assumed that the reactions of the track are proportional to both its deformation and the rate of this deformation. It is taken into consideration that the platform car carries no cargo. When building motion equations, we did not take into consideration the friction forces that occur between the components of the longitudinal beam at vertical fluctuations of the platform car.

The differential equations of the movement of the platform car take the following form:

$$M_1 \cdot \ddot{q}_1 + C_{1,1} \cdot \dot{q}_1 + C_{1,2} \cdot \dot{q}_2 + C_{1,3} \cdot \dot{q}_3 = -F_{TP} \cdot (\text{sign}(\dot{\delta}_1) + \text{sign}(\dot{\delta}_2)), \tag{1}$$

$$M_2 \cdot \ddot{q}_2 + C_{2,1} \cdot \dot{q}_1 + C_{2,2} \cdot \dot{q}_2 + B_{2,2} \cdot \dot{q}_2 = F_{TP} \cdot \text{sign}(\dot{\delta}_1) + k(\eta_1 + \eta_2) + \beta(\dot{\eta}_1 + \dot{\eta}_2), \tag{2}$$

$$M_3 \cdot \ddot{q}_3 + C_{3,1} \cdot \dot{q}_1 + C_{3,3} \cdot \dot{q}_3 + B_{3,3} \cdot \dot{q}_3 = F_{TP} \cdot \text{sign}(\dot{\delta}_2) + k(\eta_3 + \eta_4) + \beta(\dot{\eta}_3 + \dot{\eta}_4), \tag{3}$$

where  $M_1$  is the mass of the bearing structure of the platform car;

$M_2, M_3$  is the mass, respectively, of the first and second bogie in the forward direction;

$C_{ij}$  is the characteristics of elasticity of the elements of the oscillatory system, which are determined by the values of the rigidity coefficients of springs  $k_T$ ;

$k_b$  is the rigidity of elastic elements in the side beams of the frame;

$B_{ij}$  is the scattering function;

$a$  is the half of the base of the bogie;

$k$  is the track stiffness;

$\beta$  is the damping coefficient;

$F_{TP}$  is the force of absolute friction in the spring kit;

$\delta_i$  is the deformation of elastic elements of spring suspension;

$\eta(t)$  is the track irregularity.

The body's connection to the running part was described as a sequential elastic link:

$$C' = \frac{k_b \cdot (k_T + k_r)}{k_b + (k_T + k_r)}. \quad (4)$$

The input parameters of the model are the technical characteristics of the carrying structure of the platform car, spring suspension, perturbing action (Table 1).

The mathematical model was solved in the software suite Mathcad (USA) [11, 12]. To this end, the model was reduced to the normal form by Cauchy and then integrated according to the Runge-Kutta method.

$$F(t, y) =$$

$$= \begin{bmatrix} y_2 \\ y_4 \\ y_6 \\ \frac{-F_{TP} \cdot (\text{sign}(\dot{\delta}_1) + \text{sign}(\dot{\delta}_2)) - C_{1,1} \cdot y_1 - C_{1,2} \cdot y_3 - C_{1,3} \cdot y_5}{M_1} \\ \frac{F_{TP} \cdot \text{sign}(\dot{\delta}_1) + k(\eta_1 + \eta_2) + \beta(\dot{\eta}_1 + \dot{\eta}_2) - C_{2,1} \cdot y_1 - C_{2,2} \cdot y_3 - B_{2,2} \cdot y_4}{M_2} \\ \frac{F_{TP} \cdot (\text{sign}(\dot{\delta}_1) + \text{sign}(\dot{\delta}_2)) - C_{3,1} \cdot y_1 - C_{3,3} \cdot y_5 - B_{3,3} \cdot y_6}{M_3} \end{bmatrix}, \quad (8)$$

Table 1

Input parameters to the mathematical model

Parameter name	Value
Carrying structure of the platform car	
mass, t	18.0
base half, m	9.25
Bogies	
mass, t	4.3
base half, m	0.925
spring suspension rigidity, kN/m	8,000
relative friction coefficient	0.1
Track	
damping coefficient, kN·s/m	200
rigidity, kN/m	100,000
irregularity amplitude, m	0.01
irregularity length, m	25

In this case, the matrix of elastic coefficients takes the following form:

$$C = \begin{bmatrix} 2k_T & -k_T & -k_T \\ -k_T & k_T + 2k & 0 \\ -k_T & 0 & k_T + 2k \end{bmatrix}, \quad (5)$$

and the matrix of dissipative coefficients

$$B = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 2\beta_1 & 0 \\ 0 & 0 & 2\beta_1 \end{bmatrix}. \quad (6)$$

The initial movements and speeds are taken equal to zero

$$Y0 = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \quad (7)$$

The solution to the model in the Mathcad programming suite was determined in the following form

$$Z = \text{rkfixed}(Y0, tn, tk, n, F).$$

In this case,

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

The generalized accelerations were calculated in the  $ddq_{j,i}$  array:

$$ddq_{j,1} = \frac{-F_{TP} \cdot (\text{sign}(\dot{\delta}_1) + \text{sign}(\dot{\delta}_2)) - C_{1,1} \cdot y_1 - C_{1,2} \cdot y_3 - C_{1,3} \cdot y_5}{M_1}, \quad (9)$$

$$ddq_{j,2} = \frac{F_{TP} \cdot \text{sign}(\dot{\delta}_1) + k(\eta_1 + \eta_2) + \beta(\dot{\eta}_1 + \dot{\eta}_2) - C_{2,1} \cdot y_1 - C_{2,2} \cdot y_3 - B_{2,2} \cdot y_4}{M_2}, \quad (10)$$

$$ddq_{j,3} = \frac{F_{TP} \cdot (\text{sign}(\dot{\delta}_1) + \text{sign}(\dot{\delta}_2)) - C_{3,1} \cdot y_1 - C_{3,3} \cdot y_5 - B_{3,3} \cdot y_6}{M_3}. \quad (11)$$

The results are shown in Fig. 5, 6.

The maximum vertical acceleration of the carrying structure of the platform car without cargo is about  $2.0 \text{ m/s}^2$  (0.2 g); that of the bogies – about  $8.29 \text{ m/s}^2$  (0.8 g). Taking into consideration the proposed solution, it becomes possible to reduce the vertical accelerations that act on the carrying

structure of the platform car by 15 % compared to the prototype car. The car movement is rated “excellent” [13, 14].

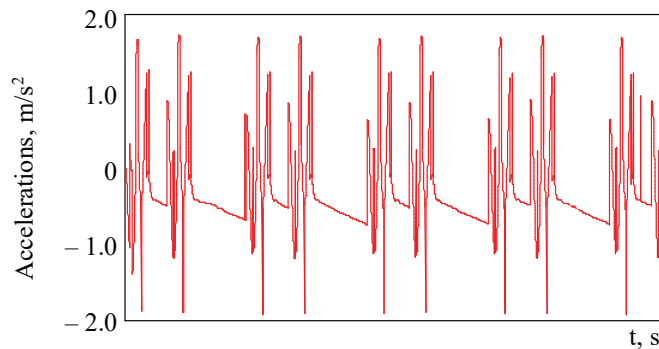


Fig. 5. Acceleration of the carrying structure of the platform car in the center of masses

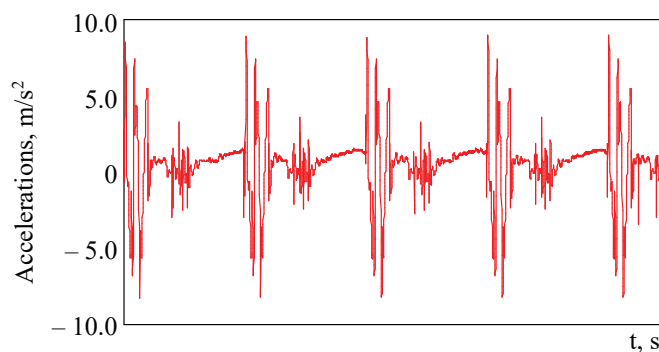


Fig. 6. Acceleration of the carrying structure of the platform car in the regions of resting on the bogies

### 6. Studying the strength of the carrying structure of the platform car

To determine the main indicators of strength of the carrying structure of the platform car with elastic elements in the longitudinal beams, we performed the calculation. In this case, the method of finite elements was applied, implemented in the software package SolidWorks Simulation (CosmosWorks), [15, 16]. The finite-element model of the carrying structure of the platform car is shown in Fig. 7. Spatial tetrahedrons were used as finite elements. To determine the optimal number of elements, the graphoanalytic method was used [17, 18]. The number of nodes in the model was 364,279; elements – 1,165,256. The maximum size of the element was 30 mm; minimum – 6 mm. The percentage of elements with an aspect ratio of less than three is 87.4, exceeding ten – 0.0402. The minimum number of elements in the circle was 10, the ratio of magnifying the size of the elements was 1.8. The carrying structure’s material was steel, grade 09G2S.

The estimation scheme of the carrying structure of the platform car is shown in Fig. 8. When compiling the estimation scheme, it was taken into consideration that the carrying structure was loaded with four containers

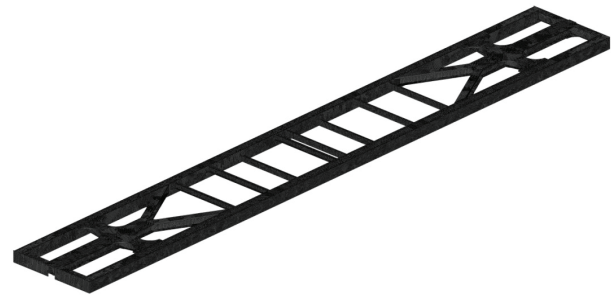


Fig. 7. The finite-element model of the carrying structure of the platform car

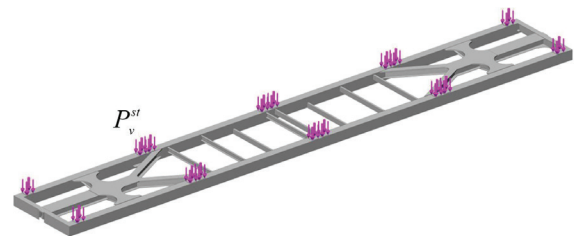


Fig. 8. Estimation scheme of the carrying structure of the platform car

the size of 1SS. Consequently, in the regions where containers rest on the carrying structure of the platform car, a vertical load  $P_v^{st}$  was applied. The model was fixed in the regions of resting on the running parts.

The results of calculating the strength of the carrying structure of the platform car are shown in Fig. 9, 10.

The maximum equivalent stresses occur, in this case, in the region of interaction between a pivot beam and the spreads, and are 254.0 MPa. The maximum movements were registered in the middle part of the longitudinal beams; and are equal to 12.6 mm. That is, the strength of the carrying structure of the platform car is ensured [13, 14, 19].

In addition, within the framework of this study, we have determined the numerical values of accelerations and the fields of their distribution relative to the carrying structure of the platform car. The calculation was performed in the software suite SolidWorks Simulation (CosmosWorks). In this case, the study parameters were identical to those used in determining the strength indicators of the carrying structure of the platform car. The calculation results are shown in Fig. 11.

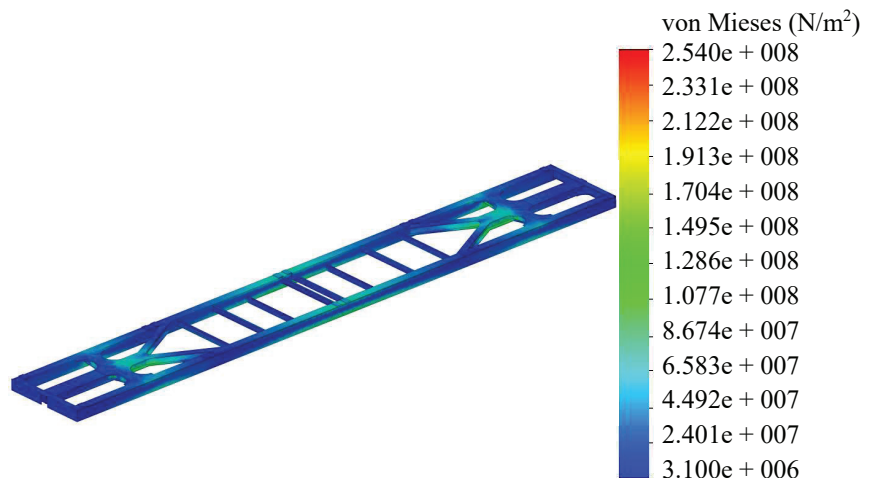


Fig. 9. The stressed state of the carrying structure of the platform car



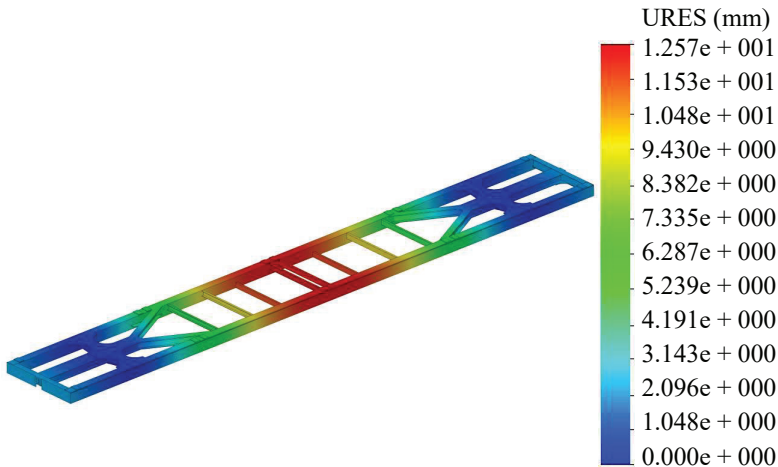


Fig. 10. Displacements in the nodes of the carrying structure of the platform car

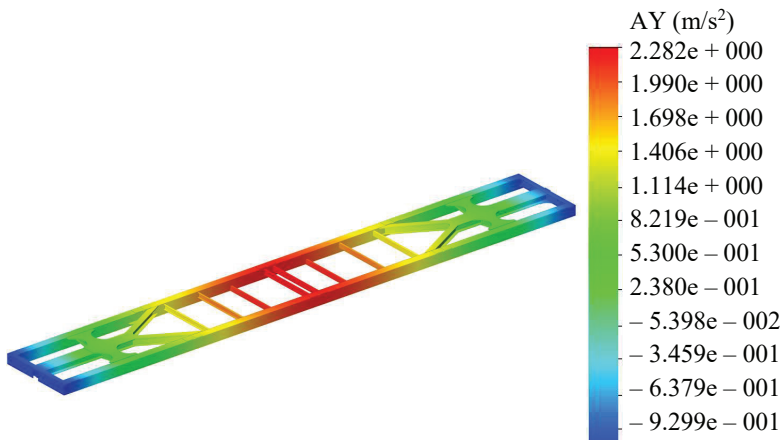


Fig. 11. The fields of acceleration distribution relative to the carrying structure of the platform car

It has been established that the maximum accelerations are concentrated in the middle part of the carrying structure of the platform car, and are 2.28 m/s<sup>2</sup>. In console parts, the acceleration amounted to 1.22 m/s<sup>2</sup>.

### 7. Studying the fatigue strength and the eigenfrequencies of oscillations in the carrying structure of the platform car

Applying the developed estimation scheme (Fig. 8), we determined the fatigue of the carrying structure of the platform car. In this case, the test base was 10<sup>7</sup> cycles. The fatigue curve was derived on the basis of the 09G2S steel elasticity module employing a feature of the SolidWorks Simulation (CosmosWorks) software. The study into fatigue was carried out on the basis of pre-calculations for static strength with the subsequent taking into consideration the cyclicity of the load.

Our study has made it possible to determine the most loaded zones of the bearing structure of the platform car (Fig. 12). These include the regions of interaction between the pivot beams and spreads, as well as the middle parts of the longitudinal beams.

In addition, based on the calculation results, we have determined an indicator of biaxiality of the bearing structure of the platform car (Fig. 13). This indicator characterizes the ratio of minimal stress in the carrying structure of the platform car to the maximum one [20–23].

Our calculations have made it possible to conclude that the fatigue strength of the carrying structure of the platform car is ensured at the predefined test base, because, after applying the cyclic load, no damage in it was detected. At the same time, the fatigue strength of the carrying structure increases by 16 % compared to a regular structure, where, following the base of 9.22·10<sup>7</sup>, in accordance with the devised estimation scheme, there are damages in the middle parts of the main longitude beams.

To determine the projected service life of the platform car, we applied the methodology given in [24]:

$$T_p = \frac{(\sigma_{-1L} / [n])^m \cdot N_0}{B \cdot f_d \cdot \sigma_{ad}^m}, \quad (12)$$

where  $\sigma_{-1L}$  is the average value of endurance limits;  $n$  is the permissible strength reserve coefficient;  $m$  is the indicator of the degree of fatigue curve;  $N_0$  is the test base;  $B$  is the coefficient that characterizes the time of continuous operation of the object in seconds;  $f_d$  is the effective frequency of dynamic stresses;  $\sigma_{ad}$  is the amplitude of equivalent dynamic stresses.

The amplitude of equivalent dynamic stresses was determined from the following formula

$$\sigma_{ad} = \sigma_{sw} (k_{dv} + \psi_\sigma / K_\sigma), \quad (13)$$

where  $\sigma_{sw}$  is the stress due to static weight load;  $k_{dv}$  is the vertical dynamics coefficient;  $\psi_\sigma$  is the sensitivity coefficient;  $K_\sigma$  is the overall coefficient of fatigue strength reduction.

In the calculations, the following input parameters were adopted:  $\sigma_{-1L}=245$  MPa;  $n=2$ ;  $m=8$ ;  $N_0=10^7$ ;  $B=3.07 \cdot 10^6$  sec;  $f_d=2.7$  Hz;  $k_{dv}=0.35$ ;  $\psi_\sigma/K_\sigma=0.2$ .

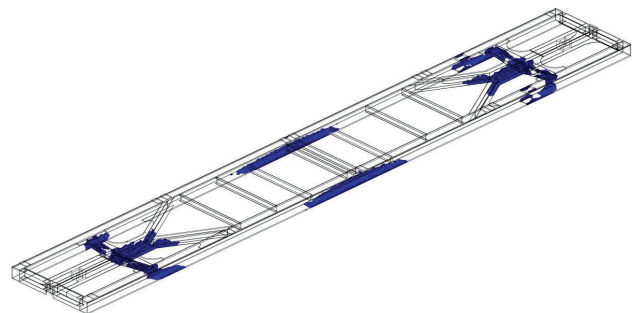


Fig. 12. The most loaded regions of the carrying structure of the platform car

Our calculations showed that the projected service life of the proposed carrying structure of the platform car is 41.9 years. That is, the resulting value of the projected service life is higher by 20 % than the service life of the prototype car. It is important to note that the resulting value of the projected service life should be refined by taking into consideration additional research into the longitudinal loading of the carrying structure of the platform car.

In addition, within the framework of this study, we determined the value of oscillation eigenfrequencies in the

carrying structure of the platform car. The calculation was carried out according to the estimation scheme shown in Fig. 8. The results of the calculation are shown in Fig. 14.

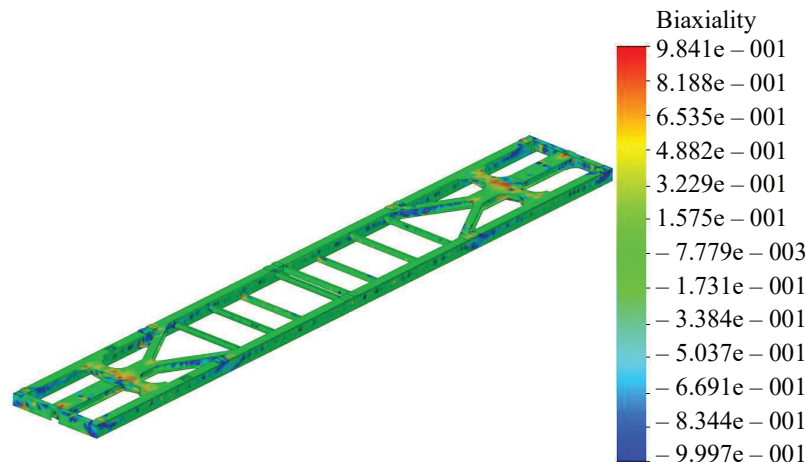


Fig. 13. Indicator of biaxiality of the bearing structure of the platform car

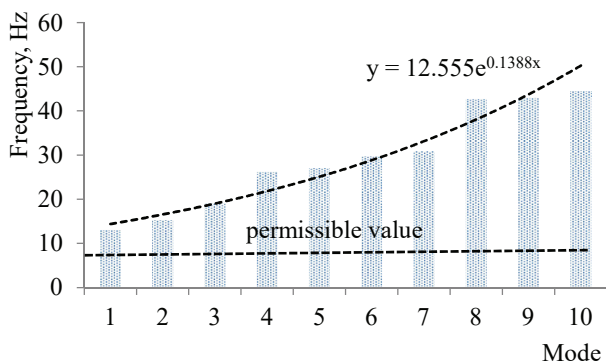


Fig. 14. Values of oscillation eigenfrequencies in the carrying structure of the platform car

According to the data illustrated in Fig. 14, we can conclude that the values of oscillation eigenfrequencies are within acceptable limits since the first oscillation eigenfrequency exceeds 8 Hz [13, 14].

### 8. Discussion of results of determining the loading on the long-base structure of the platform car with elastic elements in longitudinal beams

To reduce the dynamic load on the bearing structure of the long-base platform car, it is proposed to install elastic elements in it. The elastic elements are planned to be arranged in the longitudinal beams of the frame. To this end, it is proposed to replace them with beams of the U-shaped profile (Fig. 1).

To justify the proposed solution, mathematical modeling of the dynamic load on the bearing structure of the car was carried out. The bounce oscillations were taken into consideration. Our study was conducted in a flat coordinate system, in the XZ plane (Fig. 4). It was established that the maximum vertical acceleration of the carrying structure of the platform car was about 2.0 m/s<sup>2</sup>. That is, taking into consideration the proposed solution, it becomes possible to reduce the vertical accelerations that act on the carrying structure of the platform car by 15 %.

When building the model, the wheelset slipping relative to the rails was not considered. In addition, when constructing the model, it was assumed that the movement of the frame and

bogies of the platform car along the track axis were the same. The model takes into consideration that the bouncing of bogies is determined by the bouncing of wheelsets.

The main indicators of strength of the bearing structure of the platform car have been determined. The maximum equivalent stresses occur in the region of interaction between the pivot beam and spreads, and are 254.0 MPa (Fig. 9). The maximum movements occur in the middle part of the longitudinal beams and are equal to 12.6 mm. Consequently, the strength of the carrying structure of the platform car is ensured.

Based on the estimation scheme shown in Fig. 8, the numerical values of the acceleration of the carrying structure of the platform car and the fields of their location were determined.

The fatigue strength of the bearing structure of the platform car was investigated. The results of our calculation showed that the fatigue strength of the improved carrying structure of the platform car increases by 16 % compared to a typical structure.

The limitations of this study relate to the fact that we determined the dynamic load and strength of the carrying structure of the platform car only in the vertical plane. In further studies in this area, it is important to determine the dynamic load of the platform car in the spatial coordinate system. In addition, it is necessary to take into consideration the pliability of the cargo transported.

At the same time, our study could contribute to building modern structures of platform cars with improved technical and economic, and operational indicators.

### 9. Conclusions

1. Measures are proposed to improve the carrying structure of the platform car to increase fatigue strength during operation. A feature of the proposed solution is that the main longitudinal beams in the frame consist of U-profiles covered with horizontal sheets. Elastic elements are placed between the horizontal parts of the profiles and sheets. This technical solution would reduce the dynamic load on the carrying structure of the platform car during operation.

2. Our study of the dynamic load on the carrying structure of the platform car has shown that the maximum vertical acceleration of the carrying structure of the platform car was about 2.0 m/s<sup>2</sup> (0.2 g); that of the bogies – about 8.29 m/s<sup>2</sup> (0.8 g). Taking into consideration the proposed solution, it becomes possible to reduce the vertical accelerations that act on the carrying structure of the platform car by 15 % compared to a typical structure.

3. Studying the strength of the carrying structure of the platform car has revealed that the maximum equivalent stresses occur in the interaction zone of the pivot beam with the spreads, and are 254.0 MPa. The maximum movements occur in the middle part of the longitudinal beams and are equal to 12.6 mm. That is, the strength of the carrying structure of the platform car is ensured.

Based on the computer simulation of the dynamic load on the carrying structure of the platform car, the numerical

values of accelerations and the fields of their location relative to the carrying structure of the platform car have been determined. The maximum accelerations were detected in the middle part of the carrying structure of the platform car, and are equal to  $2.28 \text{ m/s}^2$ .

4. Our study into the fatigue strength of the bearing structure of the platform car has shown that the fatigue strength is ensured at the test base of  $10^7$ . In this case, the fatigue strength of the load-bearing structure increases by 16 % compared to a standard structure.

The results of calculating the oscillation eigenfrequencies in the carrying structure of the platform car have

demonstrated that their values are within the permissible limits.

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### References

1. Das, A., Agarwal, G. (2020). Investigation of Torsional Stability and Camber Test on a Meter Gauge Flat Wagon. *Advances in Fluid Mechanics and Solid Mechanics*, 271–280. doi: [https://doi.org/10.1007/978-981-15-0772-4\\_24](https://doi.org/10.1007/978-981-15-0772-4_24)
2. Šťastniak, P., Kurčík, P., Pavlík, A. (2018). Design of a new railway wagon for intermodal transport with the adaptable loading platform. *MATEC Web of Conferences*, 235, 00030. doi: <https://doi.org/10.1051/mateconf/201823500030>
3. Fabian, P., Gerlici, J., Masek, J., Marton, P. (2013). Versatile, Efficient and Long Wagon for Intermodal Transport in Europe. *Communications - Scientific Letters of the University of Zilina*, 15 (2), 118–123. Available at: <http://communications.uniza.sk/index.php/communications/article/view/628>
4. Krason, W., Niezgodna, T. (2014). FE numerical tests of railway wagon for intermodal transport according to PN-EU standards. *Bulletin of the Polish Academy of Sciences Technical Sciences*, 62 (4), 843–851. doi: <https://doi.org/10.2478/bpasts-2014-0093>
5. Nandan, S., Trivedi, R., Kant, S., Ahmad, J., Maniraj, M. (2020). Design, Analysis and Prototype Development of Railway Wagons on Different Loading Conditions. *SSRN Electronic Journal*. doi: <https://doi.org/10.2139/ssrn.3539937>
6. Myamlin, S., Lingaitis, L. P., Dailydka, S., Vaičiūnas, G., Bogdevičius, M., Bureika, G. (2015). Determination of the dynamic characteristics of freight wagons with various bogie. *Transport*, 30 (1), 88–92. doi: <https://doi.org/10.3846/16484142.2015.1020565>
7. Stoilov, V., Simić, G., Purgić, S., Milković, D., Slavchev, S., Radulović, S., Maznichki, V. (2019). Comparative analysis of the results of theoretical and experimental studies of freight wagon Sdggmrrs-twin. *IOP Conference Series: Materials Science and Engineering*, 664, 012026. doi: <https://doi.org/10.1088/1757-899x/664/1/012026>
8. Fomin, O., Lovska, A., Pistek, V., Kucera, P. (2020). Research of stability of containers in the combined trains during transportation by railroad ferry. *MM Science Journal*, 2020 (1), 3728–3733. doi: [https://doi.org/10.17973/mmsj.2020\\_03\\_2019043](https://doi.org/10.17973/mmsj.2020_03_2019043)
9. Fomin, O., Lovska, A., Pištěk, V., Kučera, P. (2019). Dynamic load effect on the transportation safety of tank containers as part of combined trains on railway ferries. *Vibroengineering PROCEDIA*, 29, 124–129. doi: <https://doi.org/10.21595/vp.2019.21138>
10. Domin, Yu. V., Cherniak, H. Yu. (2003). *Osnovy dynamiky vagoniv*. Kyiv: KUETT, 269.
11. Kir'yanov, D. V. (2006). *Mathcad 13*. Sankt-Peterburg: BHV. Peterburg, 608.
12. D'yakonov, V. (2000). *MATHCAD 8/2000: spetsial'nyy spravochnik*. Sankt-Peterburg: Piter, 592.
13. DSTU 7598:2014. Freight wagons. General requirements to calculation and designing of the new and modernized 1520 mm gauge wagons (non-self-propelled) (2015). Kyiv, 162.
14. GOST 33211-2014. Freight wagons. Requirements to structural strength and dynamic qualities. Moscow, 54.
15. Alyamovskiy, A. A. (2007). *SolidWorks/COSMOSWorks 2006–2007. Inzhenerniy analiz metodom konechnykh elementov*. Moscow, 784.
16. Fomin, O., Lovska, A., Radkevych, V., Horban, A., Skliarenko, I., Gurenkova, O. (2019). The dynamic loading analysis of containers placed on a flat wagon during shunting collisions. *ARNP Journal of Engineering and Applied Sciences*, 14 (21), 3747–3752. Available at: [https://www.researchgate.net/publication/337316337\\_THE\\_DYNAMIC\\_LOADING\\_ANALYSIS\\_OF\\_CONTAINERS\\_PLACED\\_ON\\_A\\_FLAT\\_WAGON\\_DURING\\_SHUNTING\\_COLLISIONS](https://www.researchgate.net/publication/337316337_THE_DYNAMIC_LOADING_ANALYSIS_OF_CONTAINERS_PLACED_ON_A_FLAT_WAGON_DURING_SHUNTING_COLLISIONS)
17. Vatulia, G. L., Lobiak, O. V., Deryzemlia, S. V., Verevicheva, M. A., Orel, Y. F. (2019). Rationalization of cross-sections of the composite reinforced concrete span structure of bridges with a monolithic reinforced concrete roadway slab. *IOP Conference Series: Materials Science and Engineering*, 664, 012014. doi: <https://doi.org/10.1088/1757-899x/664/1/012014>
18. Vatulia, G., Komagorova, S., Pavliuchenkov, M. (2018). Optimization of the truss beam. Verification of the calculation results. *MATEC Web of Conferences*, 230, 02037. doi: <https://doi.org/10.1051/mateconf/201823002037>
19. EN 12663-2. Railway applications - structural requirements of railway vehicle bodies - Part 2: Freight wagons (2010).
20. Kondratiev, A. V., Gaidachuk, V. E., Kharchenko, M. E. (2019). Relationships Between the Ultimate Strengths of Polymer Composites in Static Bending, Compression, and Tension. *Mechanics of Composite Materials*, 55 (2), 259–266. doi: <https://doi.org/10.1007/s11029-019-09808-x>
21. Dižo, J., Steišūnas, S., Blatnický, M. (2016). Simulation Analysis of the Effects of a Rail Vehicle Running with Wheel Flat. *Manufacturing Technology*, 16 (5), 889–896. doi: <https://doi.org/10.21062/ujep/x.2016/a/1213-2489/mt/16/5/889>
22. Dižo, J., Harušinec, J., Blatnický, M. (2015). Multibody System of a Rail Vehicle Bogie with a Flexible Body. *Manufacturing Technology*, 15 (5), 781–788. doi: <https://doi.org/10.21062/ujep/x.2015/a/1213-2489/mt/15/5/781>
23. Fomin, O., Lovska, A. (2020). Improvements in passenger car body for higher stability of train ferry. *Engineering Science and Technology, an International Journal*, 23 (6), 1455–1465. doi: <https://doi.org/10.1016/j.jestch.2020.08.010>
24. Ustich, P. A., Karpych, V. A., Ovechnikov, M. N. (1999). *Nadezhnost' rel'sovogo netyagovogo podvizhnogo sostava*. Moscow, 415.