

This paper substantiates prolonging the service life of a covered railroad freight car that has exhausted its rated life of operation. In this case, the actual amount of wear of the bearing structure elements, registered during operation, have been taken into consideration. The reported calculations showed that the design service life of the bearing structure of a covered freight car when taking into consideration the extension of operation, is not less than 2 years.

Mathematical modeling was carried out to determine the dynamic load on the bearing structure of a covered freight car. It has been established that the maximum accelerations that act on the bearing structure of a covered freight car are 42 m/s^2 . In order to determine the acceleration distribution fields relative to the bearing structure of a covered freight car, a computer simulation was performed. The calculations showed that the maximum acceleration amounts to 43.2 m/s^2 and is concentrated in the middle part of the girder beam. F-criterion was used to verify the dynamic load models.

The maximum equivalent stresses of the bearing structure of a covered freight car were determined taking into consideration the wear of its components. It was found that the maximum equivalent stresses occur in the area of interaction between the girder beam and the rod beam and amount to 344 MPa. In other words, the maximum equivalent stresses do not exceed permissible ones.

The basic dynamic indicators for a covered freight car have been calculated. The maximum accelerations that act on the bearing structure of a covered freight car in the center of masses reached about 5 m/s^2 . The acceleration of the body in the regions of resting on bogies was about 6 m/s^2 . The travel of the car is rated «good».

This study would contribute to improving the efficiency of the transportation process, as well as the functioning of railroad transport

Keywords: operating resource, residual resource, resource prolongation, technical condition, structural strength, strained state

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ESTABLISHING PATTERNS IN DETERMINING THE DYNAMICS AND STRENGTH OF A COVERED FREIGHT CAR, WHICH EXHAUSTED ITS RESOURCE

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1. Introduction

The transport industry is an integral part of the development of the national economy. The effectiveness of the transport industry primarily depends on providing it with rolling stock. It is known that at present there is a lack of rolling stock, due to the low level of replenishment of the car fleet. For example, the current Ukrainian fleet of freight cars includes 74.6 % morally and physically obsolete types. And, accordingly, at different intervals, due to unsatisfactory efficiency of operation, only 30 % to 60 % of the freight car fleet on average is in operation, which adversely affects the profitability of railroads. The rest of the fleet is in forced downtime, and, accordingly, generates significant losses. At the same time, reducing the total number of freight cars in the fleet could lead to a shortage of their individual types under peak loads. Consequently, the cost of railroad services and the corresponding losses of transportation customers would increase. The key factors that reduce the efficiency of the operation of freight cars are their unsatisfactory average speed, many failures, reduced stability on the tracks, etc.

It is possible to improve the operational efficiency of the freight car fleet in two main ways: to update the fleet with cars of a new generation or to restore existing obsolete cars providing for better technological and economic indicators at the appropriate competitive level. In this case, the price of a new freight car is about USD 50,000 on average while the repair sufficient to restore it (for example, overhaul) accounts for 20 % to 30 % of the said cost.

The goods that must be protected from precipitation are transported by covered freight cars. At present, the service life of freight cars for a wide gauge, including covered cars, is about 23 years. However, there are devised and implemented technologies that prolong this period two-fold. At the same time, in countries with a developed system of freight transportation, this period amounts to 50 years.

It is important to note that restoring the efficiency of obsolete freight cars is a general engineering issue. This is appropriate for the implementation and design of narrow-gauge freight cars, in particular for implementing on the railroads of Europe whose fleet exceeds 500,000 units.

In this regard, it is a relevant task to investigate the possibility to extend the service life of freight cars. Resolving this issue would ensure timely transportation of goods by rail, as well as maintaining its leadership position in the market of transport services.

2. Literature review and problem statement

Work [1] substantiated the measures to extend the service life of freight cars. The scheme of technical diagnosing of freight cars was devised. The authors drew a conclusion about the possibility to extend the service life of freight cars only by performing high-quality technical diagnosing. However, there is no applied use of the proposed measures. This may be due to the lack of relevant information regarding the technical condition of the operated cars.

Paper [2] highlights the substantiation of extending the service life of cars for the transportation of pellets. To assess the possibility of a car’s longer operation, the authors experimentally determined the level of loading and the stressed state of the bearing structures of car bodies based on static tests, tests on the impact strength, as well as resource tests.

However, the issue of extending the service life of covered freight cars, which have exhausted their rated resource, was not investigated in the cited paper.

Work [3] reports the comparative analysis of results from the theoretical and experimental studies of freight cars such as Sdggmrss-twin. The experimental research into the car structure was conducted at the University of Belgrade. The reported studies could contribute to designing lightweight structures of freight cars.

The research in [4] defines the main strength indicators of the load-bearing structure of the Zans-type freight car. The strength simulation involved the MSC. Marc software package (USA). The calculation results confirmed the feasibility of decisions taken in the design. However, the research does not specify the service life of these models of cars and the methodology for determining it. This may be due to that the studies [3, 4] focused on the bearing structures of cars with nominal sizes.

Work [5] reports research into the possibility of extending the service life of the bodies of universal semi-wagons, which exhausted their rated resource. The authors determined the dynamic loads that act on the bearing structure of a semi-wagon taking into consideration the actual wear of its elements. However, no possibility of extending the service life of covered freight cars was considered. This is due to the fact that a semi-wagon is one of the most common types of railroad cars in operation.

Study [6] analyzed the safety of railroad car movement under a periodic excitement from the rail track. A mathematical model of a heavy car was built, which travels over a rail track. The modeling results were confirmed by comparing their eigenfrequencies derived theoretically and from a field experiment. However, the cited study did not determine the resource of the bearing structures of cars taking into consideration the established dynamic loads.

Paper [7] reports the dynamic modeling of the movement of a freight car on modified bogies. The calculation is based on the Shimmns type of car. The main indicators of the car dynamics were determined.

Work [8] determined the dynamic characteristics of a freight car on different bogies. In that case, the speed varied in the range of 40–120 km/h. The work made it possible to define the main directions for improving the dynamic indicators of freight cars.

At the same time, the impact of dynamic loads on the resource of the bearing structure of a car taking into consideration the use of different types of bogies was not investigated in the above studies.

3. The aim and objectives of the study

The aim of this work is to identify patterns in determining the dynamic loading and strength of the bearing structure of a covered freight car, which exhausted its rated operating resource, in order to prolong it.

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

- to determine the residual resource of the bearing structure of a covered freight car;
- to determine the dynamic loading on the bearing structure of a covered freight car;
- to perform computer simulation of the dynamic loading and strength of the bearing structure of a covered freight car;
- to define the basic dynamic indicators of the covered freight car.

4. Determining the residual resource of the bearing structure of a covered freight car

Based on the statistical data acquired from the car depot «Osnova», the regional branch of the Southern Railroad, JSC «Ukrzaliznytsya», we determined the amount of wear of covered cars, which were in operation for 32 years. Fig. 1, 2 show the respective wear of the main bearing elements of the body and frame, registered during the field study. Measurements were carried out using a caliper, as well as a thickness meter. In this case, the upper index of the serial number of the structural element characterizes the rated value of thickness, and the lower one – actual, registered during the field study.

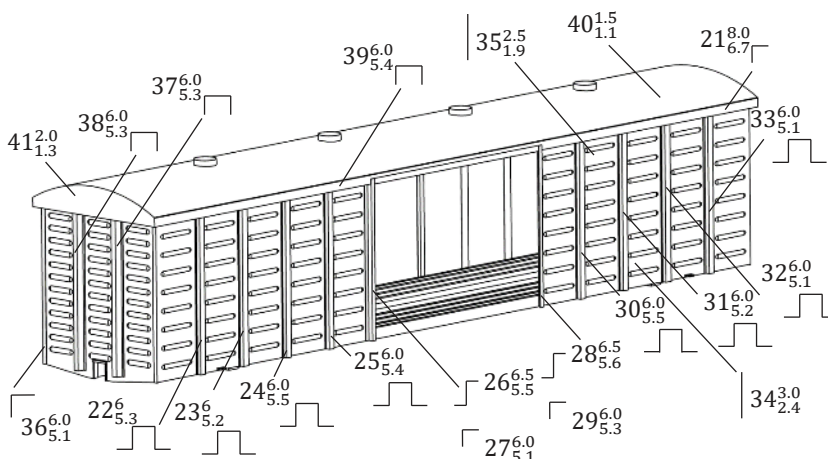


Fig. 1. The wear of a covered car’s body, model 11–217

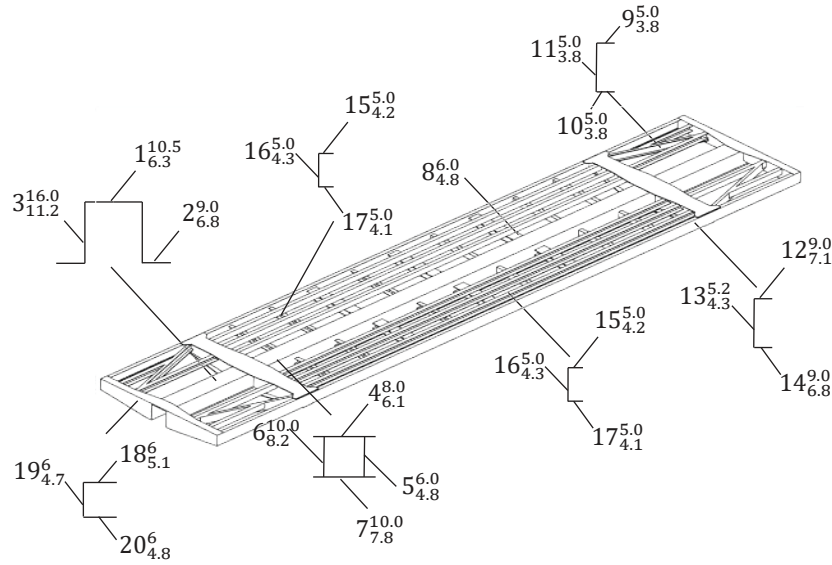


Fig. 2. The wear of a covered car's frame, model 11–217

To determine the possibility of further operation of the car, we calculated the design service life taking into consideration its prolongation. The design service life was determined from the following formula [9]:

$$T_k = \frac{\left(\frac{\sigma_{a_i N}}{[n]}\right)^m \cdot N_0}{N_{cl} \cdot \sum_j (\sigma_{a_j}^I)^m \cdot P_j^I + N_{cII} \cdot \sum_k (\sigma_{a_k}^{II})^m \cdot P_k^{II}}, \quad (1)$$

where $\sigma_{a_i N}$ is the average value of the endurance limit; $[n]$ is the permissible strength reserve coefficient; N_0 is the test base; N_{cl} , N_{cII} is the number of similar faulty elements of the car design under study; m is the indicator of the degree of a fatigue curve; $\sigma_{a_j}^I$, $\sigma_{a_k}^{II}$ is the limit of strength of the examined structure's material; P_j^I , P_k^{II} is the probability of failure of a structure's element.

The source data take the following values:

- the average value of endurance limit $\sigma_{a_i N} = 245$ MPa;
- the permissible strength reserve coefficient $[n] = 2$;
- the test base $N_0 = 10^7$;
- the number of similar faulty elements in a cars' structure;
- the indicator of the degree of a fatigue curve $m = 2$;
- the strength limit of the bearing structure's material is equal to 490 MPa for steel 09G2D.

Our procedure significantly increases the accuracy of the chosen decision on the appointment of a new service life of the car in comparison with expert evaluation. In addition, a given procedure makes it possible to determine the residual service life at the same reliability for any number of cars.

In accordance with the procedure, we determine the probability of failures P_i for the elements of the bearing structure of a covered freight car being in operation from the following formula:

$$P_i = \frac{\sum_k \frac{R_F}{R}}{k}, \quad (2)$$

where k is the number of examined cars; R_F is the number of faulty elements of the same type in a car; R is the total number of elements of the same type.

The results allowed us to conclude that the design service life of the bearing structure of a covered freight car, taking into consideration the prolongation of operation, is not less than 2 years.

5. Determining the dynamic loading on the bearing structure of a covered freight car

Mathematical modeling was involved in determining the dynamic loading on the bearing structure of a covered freight car. We have considered the most adverse case of loading the bearing structure of a car during operation – shunting collision. The estimation scheme is shown in Fig. 3.

The study was conducted in a flat coordinate system. We took into consideration the presence of three degrees of car freedom: gradual movements relative to the longitudinal axis, angular movements relative to the longitudinal axis, gradual movements relative to the vertical axis [10, 11].

The motion equations take the following form:

$$\left(M_1 + 2 \cdot m_T + \frac{n \cdot I_{WS}}{r^2}\right) \cdot \ddot{x}_C + M_1 \cdot h \cdot \ddot{\varphi}_C = P_I, \quad (3)$$

$$M_2 \cdot \ddot{\varphi}_C + M' \cdot \ddot{x}_C - g \cdot \varphi_C \cdot M' = l \cdot F_{FR} (\text{sign} \dot{\Delta}_1 - \text{sign} \dot{\Delta}_2) + l (k_1 \cdot \Delta_1 - k_2 \cdot \Delta_2), \quad (4)$$

$$M_1 \cdot \ddot{z}_C = k_1 \cdot \Delta_1 + k_2 \cdot \Delta_2 - F_{FR} (\text{sign} \dot{\Delta}_1 - \text{sign} \dot{\Delta}_2), \quad (5)$$

where M_1 is the weight of the bearing structure of a car; M_2 is the moment of inertia of the car relative to the longitudinal axis; P_I is the magnitude of the longitudinal force of impact against an automated coupling; m_T is the weight of a bogie; I_{WS} is the moment of inertia of a wheelset; r is the radius of an average-worn wheel; n is the number of bogie axles; $2a$ is the half of a car base; F_{FR} is the absolute value of dry friction force in a spring kit; k_1 , k_2 is the rigidity of springs in the spring suspension of a car's bogies; x_C , φ_C , z_C are the coordinates corresponding to, respectively, the longitudinal displacement, angular displacement around the cross axle, and the vertical displacement of a car.

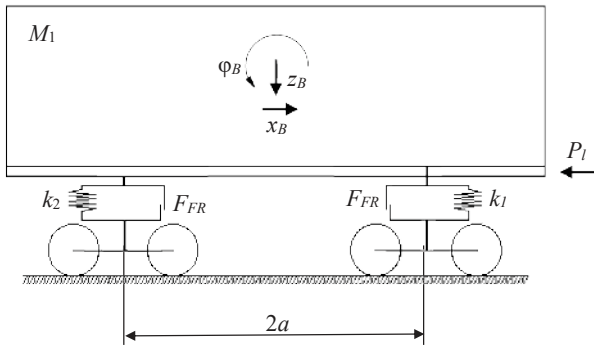


Fig. 3. The estimation scheme of a covered freight car

Equations (3) to (5) were solved in the programming environment Mathcad (USA) [12–14]. The initial movements and speeds are taken equal to zero. The input parameters to the mathematical model are the specifications of the bearing structure of a car, bogies, as well as the longitudinal force of impact against an automated coupling. In this case, it is taken into consideration that the bearing structure of a car rests on two bogies, model 18–100. The longitudinal force of impact against an automated coupling of the car is taken equal to 3.5 MN. It was established that the maximum acceleration that acts on the bearing structure of a car during an impact against an automated coupling is 42 m/s^2 .

6. Computer simulation of the dynamic load and strength of the bearing structure of a covered freight car

To determine the fields of acceleration distribution relative to the bearing structure of a covered freight car, a computer simulation was carried out using the software package SolidWorks Simulation (CosmosWorks) (France). The calculation was carried out by a finite-element method. To this end, we built a spatial model of the bearing structure of a covered freight car, model 11–217, made by VAT «Altayvagon» (Russia) (Fig. 4). When building a spatial model, the design elements were taken into consideration, which rigidly interact through welding or rivets. The model does not take into consideration self-sealed doors because they are connected by hinges to the bearing structure of the car. It was established that the weight of the bearing structure of the covered freight car, taking into consideration the wear of its elements, is 17 % less compared to the bearing structure of a covered freight car of rated dimensions.

The finite-element model (FEM) of the bearing structure of a covered freight car is shown in Fig. 5.

In determining the number of grid elements, the graph-analytic method was applied [15–17]. Ten-node isoparametric tetrahedra were used as finite elements. The number of grid elements was 661,885, with 231,671 nodes. In this case, the maximum size of the element was 100 mm, the minimum was 20 mm, the maximum aspect ratio was 845.3, the percentage of elements with an aspect ratio of less than 3–11.6, more than 10–46.7. The ratio of increase in the size of the element is 1.8. The minimum number of elements in a circle is 22. The model was fixed in the regions where the bearing structure rests on the bogies.

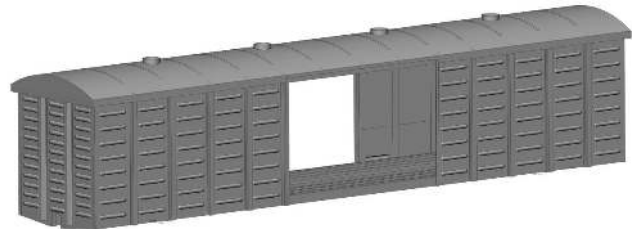


Fig. 4. The bearing structure of a covered freight car showing the wear of its elements

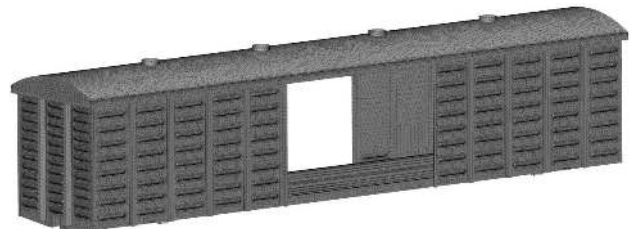


Fig. 5. The FEM of the bearing structure of a covered freight car

When compiling an estimation scheme of the bearing structure of a covered freight car, it is taken into consideration that it is exposed to the vertical static loading P_v , as well as the longitudinal loading P_{long} on the rear supports of an automated coupling, which is equal to 3.5 MN (Fig. 6).

Our calculations showed that the maximum acceleration, which acts on the bearing structure of a covered freight car, is 43.2 m/s^2 , concentrated in the middle part of the girder beam (Fig. 7).

To verify the dynamic load models of the bearing structure of a covered freight car, the calculation was performed according to the F-criterion [18, 19]. In this case, the force of impact against an automated coupling varied in the range of 1,100–35,500 kN (Table 1). The simulation results are shown in Fig. 8.

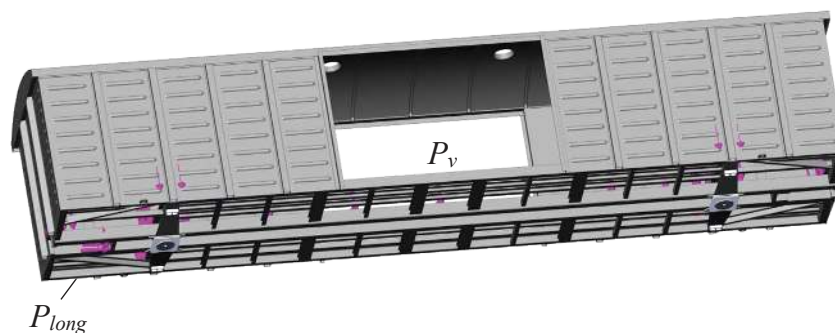


Fig. 6. The estimation scheme of the bearing structure of a covered freight car

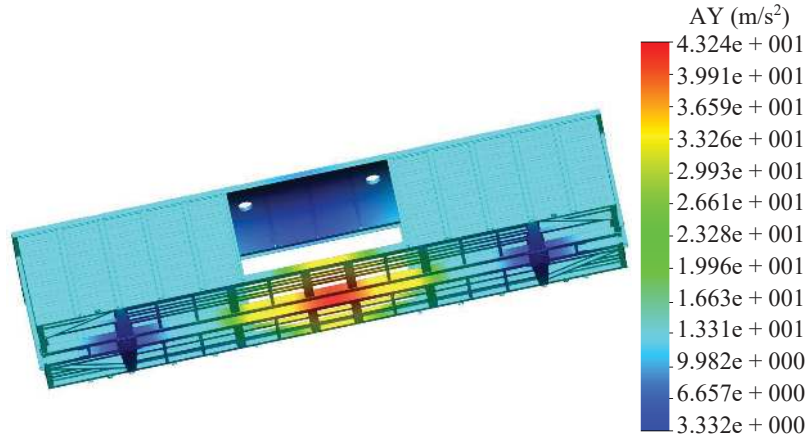


Fig. 7. Acceleration of the bearing structure of a car at an impact

Table 1

Results of simulating the dynamic loading on the bearing structure of a covered freight car

The force of impact against an automated coupling, kN	1,100	1,400	1,700	2,000	2,300	2,600	2,900	3,200	3,500
Computer simulation	10.5	13.3	17.1	20.2	23.2	28.4	33.5	35.9	39.3
Mathematical modeling	14.2	18.1	21.3	25.1	31.1	33.6	35.8	39.5	43.2

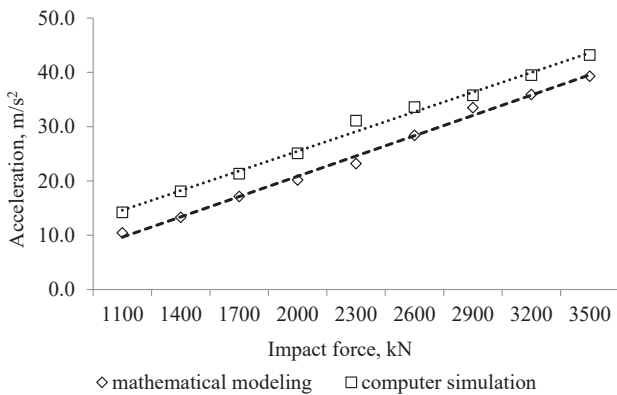


Fig. 8. Acceleration of the bearing structure of a covered freight car

It was determined that the estimated value of F -criterion $F_p=0.94$ while the tabular value is $F_t=3.29$. Thus, the hypothesis of adequacy is not rejected.

To determine the main indicators of strength of the bearing structure of a covered freight car with the wear of its elements, the calculation was performed. In this case, we used the estimation scheme shown in Fig. 6. The calculation results are shown in Fig. 9, 10.

The maximum equivalent stresses occur in the region of interaction between the girder beam and rod beam and amount to 344 MPa.

That is, the maximum equivalent stresses do not exceed allowable ones [20–22]. The maximum displacements occur in the middle part of the girder beam and are equal to 6.4 mm. The maximum equivalent deformations were $3.85 \cdot 10^{-3}$.

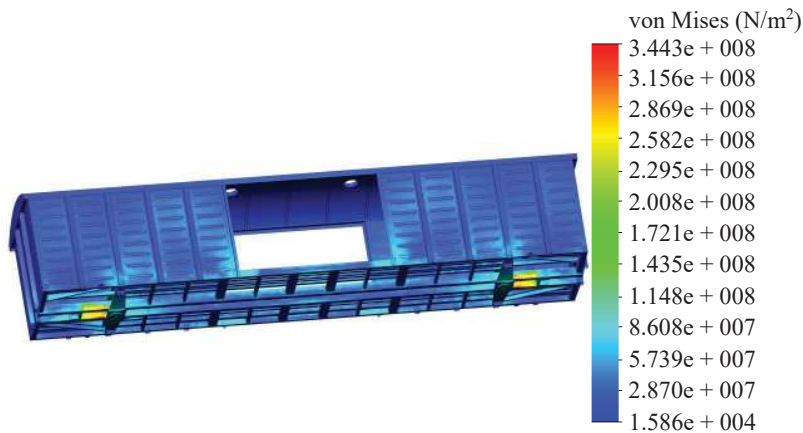


Fig. 9. The stressed state of the bearing structure of a covered freight car

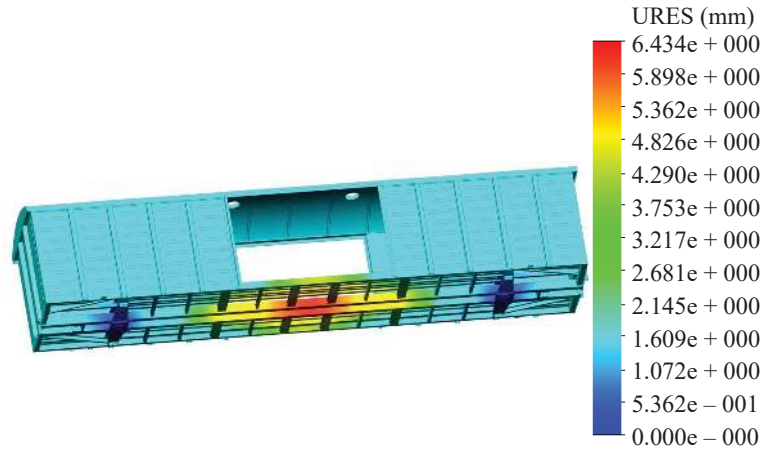


Fig. 10. Displacements in the nodes of the bearing structure of a covered freight car

7. Determining the main dynamic indicators of a covered freight car

Mathematical modeling of the vertical dynamics was carried out to assess the smooth run of a covered freight car. In this case, we use a mathematical model given in [23]. The estimation scheme of the car is shown in Fig. 11. The calculation was performed for the movement of an empty car.

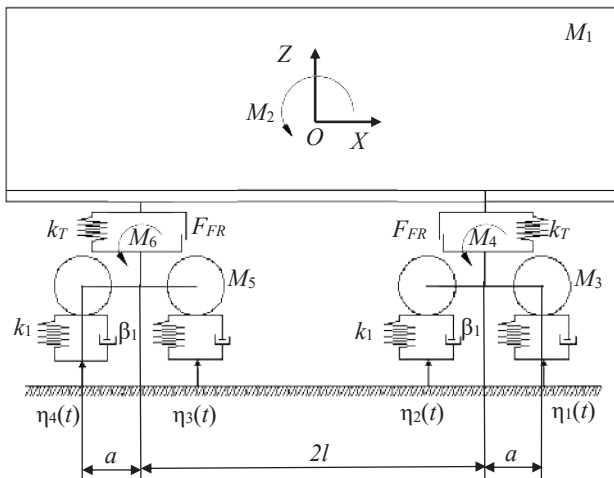


Fig. 11. The estimation scheme of a covered freight car

The motion equations for the estimation model take the following form:

$$M_1 \cdot \frac{d^2}{dt^2} q_1 + M_1 \cdot h \cdot \frac{d^2}{dt^2} q_3 = P_1, \tag{6}$$

$$M_1 \cdot \frac{d^2}{dt^2} q_1 + C_{1,1} \cdot q_1 + C_{1,3} \cdot q_3 + C_{1,5} \cdot q_5 = -F_{FR} \cdot \left(\text{sign} \left(\frac{d}{dt} \delta_1 \right) + \text{sign} \left(\frac{d}{dt} \delta_2 \right) \right), \tag{7}$$

$$M_2 \cdot \frac{d^2}{dt^2} q_2 + C_{2,2} \cdot q_2 + C_{2,3} \cdot q_3 + C_{2,5} \cdot q_5 = F_{FR} \cdot l \cdot \left(\text{sign} \left(\frac{d}{dt} \delta_1 \right) + \text{sign} \left(\frac{d}{dt} \delta_2 \right) \right), \tag{8}$$

$$M_3 \cdot \frac{d^2}{dt^2} q_1 = H_1, \tag{9}$$

$$M_3 \cdot \frac{d^2}{dt^2} q_3 + C_{3,1} \cdot q_1 + C_{3,2} \cdot q_2 + C_{3,3} \cdot q_3 + B_{3,3} \cdot \frac{d}{dt} q_3 = F_{FR} \cdot \text{sign} \left(\frac{d}{dt} \delta_1 \right) + k_1 (\eta_1 + \eta_2) + \beta_1 \left(\frac{d}{dt} \eta_1 + \frac{d}{dt} \eta_2 \right), \tag{10}$$

$$M_4 \cdot \frac{d^2}{dt^2} q_1 = H_2, \tag{11}$$

$$M_4 \cdot \frac{d^2}{dt^2} q_4 + C_{4,4} \cdot q_4 + B_{4,4} \cdot \frac{d}{dt} q_4 = -k_1 (\eta_1 - \eta_2) - \beta_1 \cdot a \cdot \left(\frac{d}{dt} \eta_1 - \frac{d}{dt} \eta_2 \right), \tag{12}$$

$$M_5 \cdot \frac{d^2}{dt^2} q_5 + C_{5,1} \cdot q_1 + C_{5,2} \cdot q_2 + C_{5,5} \cdot q_5 + B_{5,5} \cdot \frac{d}{dt} q_5 = F_{FR} \cdot \text{sign} \left(\frac{d}{dt} \delta_2 \right) + k_1 (\eta_3 + \eta_4) + \beta_1 \left(\frac{d}{dt} \eta_3 + \frac{d}{dt} \eta_4 \right), \tag{13}$$

$$M_6 \cdot \frac{d^2}{dt^2} q_6 + C_{6,6} \cdot q_6 + B_{6,6} \cdot \frac{d}{dt} q_6 = -k_1 \cdot a \cdot (\eta_3 - \eta_4) - \beta_1 \cdot a \cdot \left(\frac{d}{dt} \eta_3 - \frac{d}{dt} \eta_4 \right), \tag{14}$$

M_1, M_2 is the mass and moment of inertia of the bearing structure of a car; M_3, M_4 is the mass and moment of inertia of the first bogie in the forward motion; M_5, M_6 is the mass and moment of inertia of the second bogie in the forward motion; C_{ij} are the characteristics of elasticity of the elements within an oscillatory system, which are determined by the values of rigidity coefficients for springs k_b ; B_{ij} is the scattering function; a is the half of a bogie base; k is the track stiffness; β is the damper coefficient; $\eta_i(x)$ is the function that describes a track irregularity; δ_i is the deformation of elastic elements in a spring suspension; F_{FR} is the absolute friction force in a spring kit; H_1, H_2 are the values of the horizontal forces applied to the stops of the first and other bogies; h is the height of the center of masses of the bearing structure of a car.

A track irregularity is described by a periodic function [23]:

$$\eta(t) = \frac{d}{2}(1 - \cos \omega t), \tag{15}$$

where d is the assigned irregularity depth; Ω is the frequency of fluctuations ($\Omega = V/L$, V is the motion speed, L is the irregularity length).

$$M'_1 = M_1 + (M_3 + M_5) + \frac{nI}{r^2}, \tag{16}$$

where n is the bogie axle number; I is the moment of inertia of a wheelset; r is the wheel radius.

The value of the horizontal force applied to the bogie's stop is determined from the following:

$$H_{1,2} = \frac{P_l}{1 - \frac{P_l}{P_k}} \left[\frac{\delta L}{l^2} \left(1 + \frac{L}{a} \right) \pm \alpha \frac{L_c}{R} \right], \tag{17}$$

where P_k is the value of critical force (for a four-axle car, adopted equal to 4.0 MN); δ are the free movements of the rod intersections of a car; L is the semi-length of the frame between the stop plates of automated couplings; l is the half-base of a car; a is the length of an automated coupling's housing; L_c is the half-length of a car based on the clutch axles of automated couplings; R is the radius of the curve.

$$\alpha = \frac{1 - \frac{P_l}{P_k}}{1 - \frac{P_l}{P_b}}, \tag{18}$$

$$P_b = a \cdot c, \tag{19}$$

where c is the stiffness of a spring suspension.

In this case, the first two equations characterize the movement of the body during the fluctuations of bouncing and galloping, the second – fourth – the running part of a car. The mathematical model was solved by the Runge-Kutta method using the Mathcad software [24–27].

The initial displacements and speeds are taken equal to zero. Our calculations took into consideration the parameters of a spring suspension in the bogie of model 18–100.

The calculation results are shown in Fig. 12, 13.

The maximum accelerations that act on the bearing structure of a covered freight car in the center of masses amounted to about 5 m/s² (Fig. 12). The acceleration of the body in the regions of resting on the bogies was about 6 m/s² (Fig. 13).

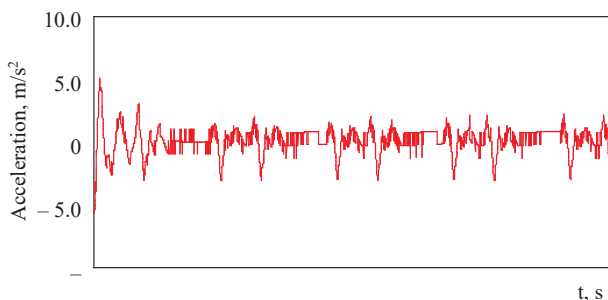


Fig. 12. Acceleration of the body in the center of masses

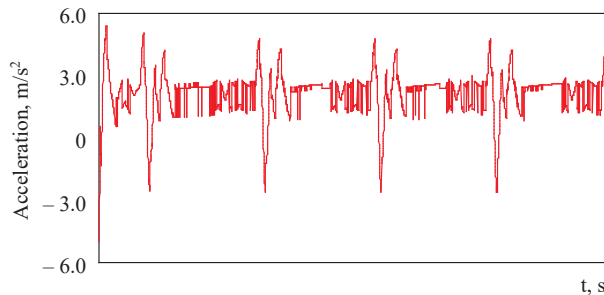


Fig. 13. Acceleration of the body in the regions of resting on bogies

Based on the calculations, we can conclude that the smooth run of a car can be assessed as «good».

The mathematical model (4) to (12) was used to determine other indicators of the dynamics of a covered freight car (Table 2). The calculation was carried out at a car speed of 80 km/h.

Table 2

The dynamic indicators of a covered freight car that runs empty

Indicator	Value	
	Estimated	Rated
Body acceleration, m/s ²	5.6	7.5
Body acceleration in the region of resting on the first bogie in the forward motion, m/s ²	6.0	7.5
Body acceleration in the region of resting on the second bogie in the forward motion, m/s ²	6.0	7.5
Force in a spring suspension of the first bogie, kN	41.2	–
Force in a spring suspension of the second bogie, kN	41.2	–
The dynamics factor of the first bogie	0.73	0.9
The dynamics factor of the second bogie	0.73	0.9
Acceleration of the first bogie in the forward motion, m/s ²	8.3	9.8
Acceleration of the second bogie in the forward motion, m/s ²	8.3	9.8
Interaction forces of the first wheelset in the forward motion, kN	35.9	–
Interaction forces of the second wheelset in the forward motion, kN	26.7	–
Interaction forces of the third wheelset in the forward motion, kN	35.8	–
Interaction forces of the fourth wheelset in the forward motion, kN	26.7	–

The results allow us to conclude that the indicators of dynamics are within permissible ones.

8. Discussion of results of studying the possibility of extending the service life of the bearing structure of a covered freight car

Our study has investigated the possibility of extending the service life of the bearing structure of a covered freight car, which exhausted its rated operating resource. In this case, the actual amounts of wear of the elements of the bearing structure of a covered freight car, registered during

the field research (Fig. 1, 2), were taken into consideration. It was established that the design service life of the bearing structure of a covered freight car, taking into consideration the prolongation of operation, is not less than 2 years.

We have determined the dynamic load on a covered freight car with the actual dimensions of bearing elements under operational modes. It was established that the maximum acceleration of the bearing structure of a covered freight car during shunting collision is 42 m/s^2 . Our results were verified by comparing them with the results from a computer simulation (Fig. 8). The impact of the dynamic load on a covered freight car with the actual dimensions of the bearing elements on its strength has been investigated. It was found that the maximum equivalent stresses are 344 MPa, concentrated in the region of interaction between the girder and rod beams (Fig. 10). In this case, the resulting stress values do not exceed permissible ones. The vertical dynamics of a covered freight car have been examined. It is established that the values of the dynamics are within permissible ones (Table 3).

This study's limitation is that the calculations were carried out in a flat coordinate system. In the future, to obtain a more accurate assessment of dynamic loads, it is possible to move to a spatial coordinate system.

In addition, it is important to determine the strength indicators of the bearing structure of a covered freight car under other estimation modes of loading.

At the same time, this study could contribute to improving the efficiency of the railroad industry through the timely provision of it with rolling stock.

9. Conclusions

1. We have determined the residual resource of the bearing structure of a covered freight car. The study involved a covered freight car of model 11–217. At the same time, we took into consideration the actual amounts of wear of the main elements of the bearing structure of a covered freight car. The results allowed us to conclude that the design service life of the bearing structure of a covered freight car, taking into consideration the prolongation of operation, is not less than 2 years.

2. The dynamic loading on the bearing structure of a covered freight car has been determined. The study was conducted in a flat coordinate system. We have taken into consideration the presence of three degrees of car freedom: the translational movements relative to the longitudinal axis,

the angular movements relative to the longitudinal axis, the translational movements relative to the vertical axis. The mathematical model was solved in the software package Mathcad. It was established that the maximum acceleration that acts on the bearing structure of a car at an impact against an automated coupling is 42 m/s^2 .

3. We have performed a computer simulation of the dynamic load and strength of the bearing structure of a covered freight car. The calculation was carried out in the software package SolidWorks Simulation (CosmosWorks) by a finite-element method. The calculations showed that the maximum acceleration that acts on the bearing structure of a covered freight car is 43.2 m/s^2 , concentrated in the middle part of the girder beam.

To verify the models of the dynamic loading on the bearing structure of a covered freight car, the calculation was carried out according to the *F*-criterion. It is determined that the estimated value of the *F*-criterion is $F_p=0.94$ while the tabular value is $F_t=3.29$. Thus, the hypothesis of adequacy is not rejected.

The main indicators of strength of the bearing structure of a covered freight car have been determined. The maximum equivalent stresses occur in the region of interaction between the girder and rod beams, and are 344 MPa; therefore, they do not exceed permissible ones. The maximum movements occur in the middle part of the girder beam and are equal to 6.4 mm. The maximum deformations amounted to $3.85 \cdot 10^{-3}$.

4. The basic dynamic indicators of the covered freight car have been determined. The calculation was performed for the car running empty. In this case, the maximum accelerations that act on the bearing structure of a covered freight car in the center of masses were about 5 m/s^2 . The acceleration of the body in the regions of resting on the bogies was about 6 m/s^2 . Based on our calculations, it has been concluded that the smooth car run can be assessed as «good».

This study could contribute to improving the efficiency of the transportation process, as well as to the functioning of railroad transport.

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