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DETERMINATION OF LOADS ON THE BODY OF A BOXCAR WITH ELASTIC ELEMENTS IN THE CENTER SILL

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Abstract.

The authors suggest elastic elements in the body of a centre sill being the basic carrying element of the frame to decrease the dynamic loads. This solution can transform the dynamic loads on the body into the work of the dry friction forces between the components of the centre sill. The authors substantiated the solution by means of mathematic modelling of the dynamic loads on the body of a boxcar in the vertical plane, including the bouncing oscillations. The differential equations of the motion were solved with the Runge–Kutta method under the zero initial conditions. This solution can decrease the accelerations on the body of a boxcar by about 20 % in comparison to that of the prototype car. The study presents the strength calculations and the design service life for the body of a boxcar. It was calculated that the design service life of a boxcar was longer than that of the prototype car by about 20 %. The research may be used by those who are concerned about higher efficiency of railway transportation.

Keywords: Transport mechanics, boxcar, body, dynamic loading, strength, design service life.

1. Introduction

The prospects of transport infrastructure development require a higher operational efficiency of the railway transport as one of the leading industries. Special attention should be paid to the technical maintenance of the rolling stock.

Boxcars are intended for transportation of goods requiring weather protection. As it is known, the most vulnerable element of the body is the frame, due to high dynamic loads occurring during the operation. These cyclic loads decrease the strength of the body of a boxcar. Therefore, it leads to off-schedule repairs or a complete removal of boxcars from the inventory.

And it is of primary importance for the railway transport to introduce an innovative rolling stock. The design of such an innovative vehicle requires an application of new effective solutions aimed at higher fatigue strength, and, therefore, longer design service life. Thus, the appropriate research and data collection in the field should be conducted.

The special features of a static and modal numerical analysis for a boxcar are discussed in study [1]. The strength characteristics are determined by means of FEM.

The authors studied the structural peculiarities of basic boxcar models manufactured in CIS countries in a previous publication [2]. However, the issue of a boxcar with an improved body for lower dynamic loads is not studied.

Aluminium "sandwich" panels, as improvements for the body, are described in [3]. The authors search for an optimal combination of the maximum stresses and displacements. However, the study does not explain the mechanism for a decrease of the dynamic load in a car improved by these panels.

The research into the structural peculiarities of BC-NHL car is given in publication [4]. The article outlines some possible solutions to improve the technical parameters of carbodies. The structural peculiarities of cars intended for combined transportation are studied in [5]. The article gives the strength study for the body of such a car. The design model was built in accordance with the PN-EN standards.

However, these studies have nothing to do with measures to reduce the dynamic loads on the body of cars.

The usage of elastic and viscous elements in the linkage components and the body of rail cars to decrease the dynamic loads during the combined transportation is presented in studies [6, 7]. It was found that flexible links could reduce the dynamic loading of transport facilities by 30 % in comparison with the standard structures.

However, the authors do not study an impact of these elastic and viscous elements in the body of a car on the loading.

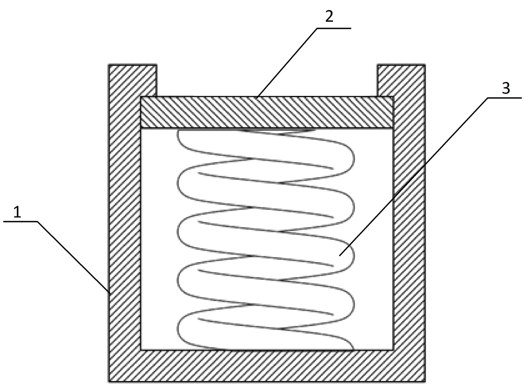

Figure 1. Section of the centre sill of a boxcar, 1 – U-like profile; 2 – horizontal plate; 3 – elastic element.

The literature review made it possible to conclude that at present, the problem of lower loads on the body of a boxcar by means of an application of elastic elements has not been studied. Therefore, there is a need to carry out the appropriate research in the field.

The objective of the research is to ground the application of elastic elements in the body of a boxcar in order to decrease the dynamic loads and prolong the design service life. Research tasks are:

(1.) to offer suggestions for decreasing the dynamic load of a boxcar;

(2.) to make the mathematic modelling of the dynamic load of a boxcar with consideration of these suggestions;

(3.) to research the strength of the body of a boxcar with these suggestions; and

(4.) to determine the design service life of a boxcar taking into account the suggestions offered.

Thus, in order to decrease the dynamic load on the body of a boxcar and increase the fatigue strength during operational modes, the authors suggest elastic elements in the centre sill along the length between the rear supports of the automatic couplers. It can be fulfilled by using the U-like profile for the centre sill, instead of the standard profile (Figure 1).

It is assumed that the decline in the load of the centre sill is achieved due to resistance from the dry friction forces between the vertical planes of the U-like profile and the vertical parts of the horizontal plate during the bouncing oscillations of a car.

This solution was suggested as a concept for substantiation of the elastic friction links in the bearing structure of a car to decrease its dynamic loading in operation. Therefore, this stage did not include the number of elements in the frame, the distance between them, and other parameters.

This solution is patented as a utility model.

The wear of the contacting surfaces can be controlled during the technical maintenance of a car. Sensors can also be used for the diagnosis of the wear of the structural elements.

The worn elastic elements can be replaced through special windows or turnable parts in the frame.

Corrosion can be prevented by means of anticorrosion coatings widely used in machine engineering, particularly, wagon construction.

When applied, the solution proposed can lead to an increase in the tare weight. However, this can be solved by means of materials with a lower weight and high strength values at operational loading for the body components, such as composites. The tare weight can also be decreased by means of optimization of structural elements with reserve strength.

It should be noted that the standard configuration of the centre sill of freight cars is open and generally consists of two Z-profiles. The authors suggest that the open configuration should be used, however it should be covered with a horizontal sheet with elastic elements beneath. Here, the main vertical loading is taken by the horizontal sheet that transfers it to the elastic elements. This loading transfer diagram is used in machine engineering and has proved effective in the spring suspension of bogies, where the bolster beam functions as the horizontal sheet.

The research was made for an 11-217 boxcar. The computer model of the body of a boxcar was designed in the SolidWorks software (Figure 2). The placement of the elastic elements in the centre sill of a car is given in Figure 3.

2. Mathematical modelling of dynamic loading on the bearing structure of a box car

The inertial load on the body of the improved boxcar was defined by means of mathematic modelling. The calculation scheme of the boxcar is given in Figure 4.

This boxcar was considered as a system of four solid bodies: body, two bogies with suspension groups of a certain rigidity and relative friction coefficient and freight. And the elastic elements are components of the bearing structure of a car because they are located in its centre sill. This assumption is reasonable as the elastic elements are used for reducing its dynamic loading; they do not link separate elements of the car body together.

When conducting the research, it was taken into account that the bouncing of bogies is determined by the bouncing of wheel pairs.

Equations (1)–(3) included that:

- $Z_1 \sim q_1$ – coordinate that describes translational displacements of a bogie relative to the vertical axis;
- $Z_2 \sim q_2$ – coordinate that describes translational displacements of the first bogie facing the engine relative to the vertical axis;
- $Z_3 \sim q_3$ – coordinate that describes translational displacements of the second bogie facing the engine relative to the vertical axis;

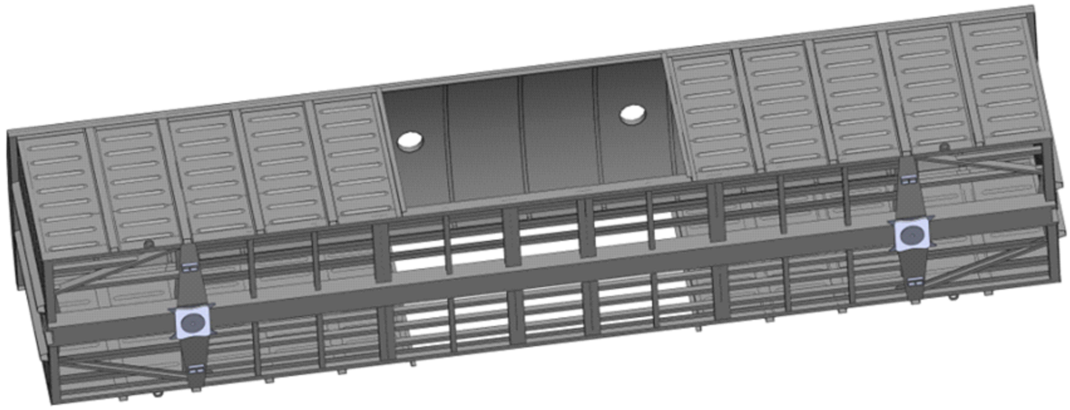

Figure 2. Computer model of the improved body of a boxcar.

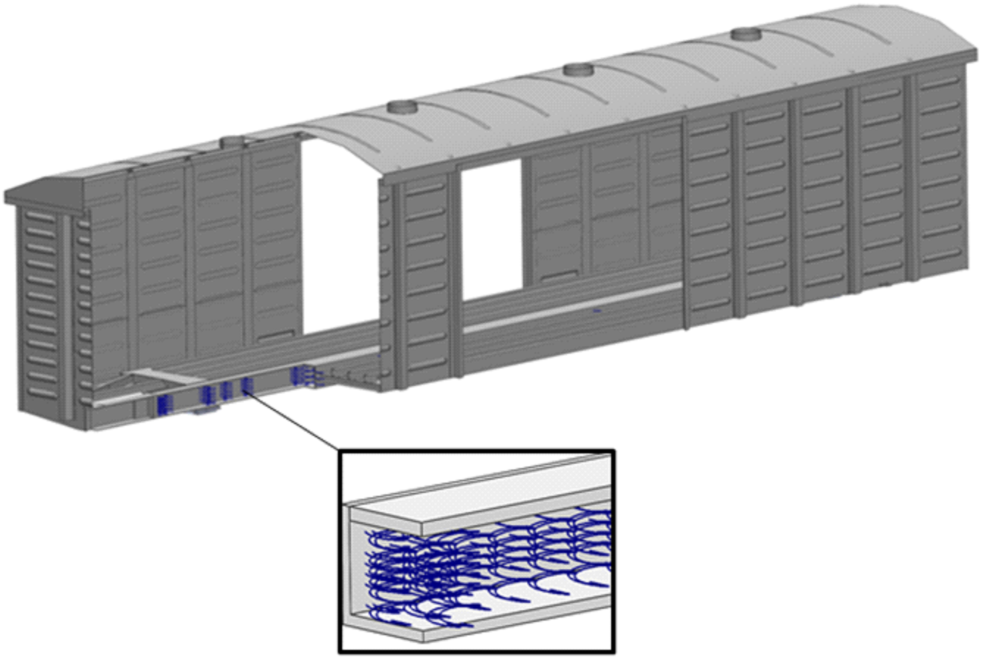

Figure 3. Body of a boxcar with elastic elements in the centre sill.

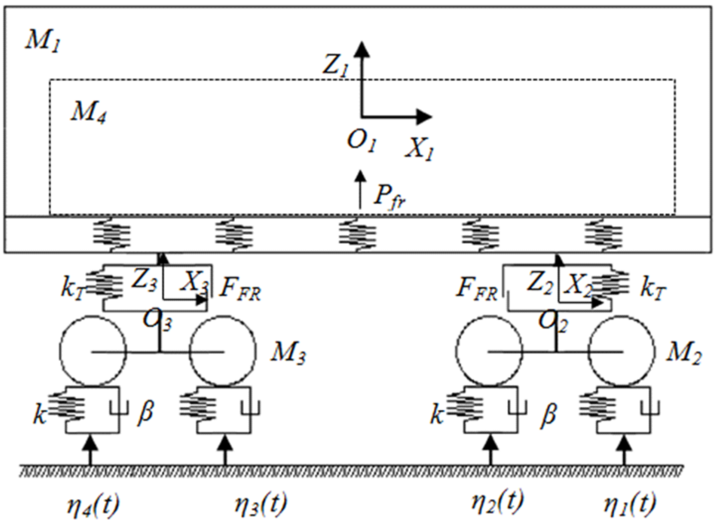

Figure 4. Calculation scheme of a boxcar.

- $Z_4 \sim q_4$ – coordinate that describes translational displacements of the freight relative to the vertical axis.

The study included that the car displacements were described by the equations:

$$M_1 \cdot \frac{\mathrm{d}^2}{\mathrm{d}t^2} q_1 + C_{1,1} \cdot q_1 + C_{1,2} \cdot q_2 + C_{1,3} \cdot q_3 = -F_{FR} \cdot \left(\mathrm{sign}\left(\frac{\mathrm{d}}{\mathrm{d}t}\delta_1\right) + \mathrm{sign}\left(\frac{\mathrm{d}}{\mathrm{d}t}\delta_2\right)\right) - F_z, \quad (1)$$

$$M_2 \cdot \frac{\mathrm{d}^2}{\mathrm{d}t^2} q_2 + C_{2,1} \cdot q_1 + C_{2,2} \cdot q_2 + C_{2,3} \cdot q_3 + B_{2,2} \cdot \frac{\mathrm{d}}{\mathrm{d}t} q_2 = F_{FR} \cdot \mathrm{sign}\left(\frac{\mathrm{d}}{\mathrm{d}t}\delta_1\right) + k(\eta_1 + \eta_2) + \beta\left(\frac{\mathrm{d}}{\mathrm{d}t}\eta_1 + \frac{\mathrm{d}}{\mathrm{d}t}\eta_2\right), \quad (2)$$

$$M_3 \cdot \frac{\mathrm{d}^2}{\mathrm{d}t^2} q_3 + C_{3,1} \cdot q_1 + C_{3,2} \cdot q_2 + C_{3,3} \cdot q_3 + B_{3,3} \cdot \frac{\mathrm{d}}{\mathrm{d}t} q_3 = F_{FR} \cdot \mathrm{sign}\left(\frac{\mathrm{d}}{\mathrm{d}t}\delta_2\right) + k(\eta_3 + \eta_4) + \beta\left(\frac{\mathrm{d}}{\mathrm{d}t}\eta_3 + \frac{\mathrm{d}}{\mathrm{d}t}\eta_4\right), \quad (3)$$

$$M_4 \cdot \ddot{q}_4 = F_z - M_4 \cdot g, \quad (4)$$

$$F_z = -P_{fr} \cdot \mathrm{sign}(\dot{q}_1 - \dot{q}_4) + k_b \cdot (q_1 - q_4), \quad (5)$$

where $M_{i,j}$ – mass and moment of inertia of the oscillation system elements; $C_{i,j}$ – elasticity of the oscillation system components; $B_{i,j}$ – damping coefficients; q_i – coordinates corresponding to translational displacement relative to the vertical axis of the car body, and first and second bogies, respectively; k_T – spring suspension rigidity; k – track rigidity; β – damping coefficient; F_{FR} – absolute friction force in a spring group; δ_i – deformations of elastic elements in a spring suspension; η – track irregularity; k_b – rigidity of the elastic elements in the centre sill; P_{fr} – friction force arising in the centre girder.

The matrix of elastic coefficients has the form:

$$C = \begin{vmatrix} 2k_r & -k_T & -k_T & k_b \\ -k_T & k_T + 2k & 0 & 0 \\ -k_T & 0 & k_T + 2k & 0 \\ k_b & 0 & 0 & 0, \end{vmatrix} \quad (6)$$

It was taken that an empty car passed over a joint irregularity described by periodic function [8]:

$$\eta(t) = \frac{h}{2}(1 - \cos\omega t), \quad (7)$$

where h – irregularity depth; ω – oscillation frequency defined by the formula $\omega = 2\pi V/L$ (V is car speed and L is irregularity length).

The differential equations (1)–(4) were solved in MathCad [9, 10]. The initial conditions are set equal to zero. The calculation results obtained are shown in Figures 5 and 6.

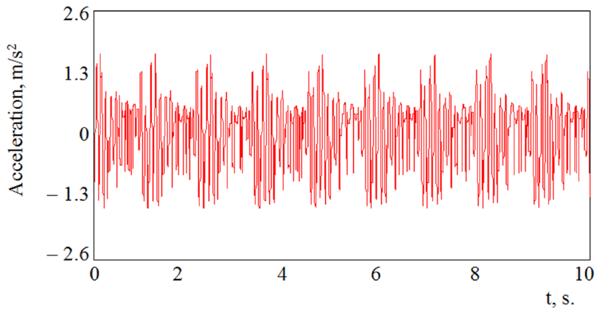

FIGURE 5. Accelerations on the body of a boxcar in the centre of mass.

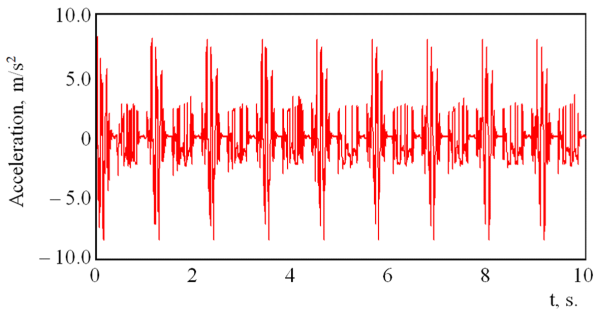

FIGURE 6. Accelerations of bogies.

The initial parameters presented in Table 1 were included in the calculation.

Parameter	Value
M_1, t	15.3
M_2, M_3, t	4.3
M_4, t	68
k_T, kN/m	8000
k_b, kN/m	8000
P_{fr}, kN	73.6
k, kN/m	100 000
β, kN · s/m	200
h, m	0.01
L, m	3.0
V, km/h	100

TABLE 1. The initial parameters for determination of the dynamic loading of a boxcar.

The maximum acceleration of the body of an empty boxcar was about 1.43 m/s^2 (0.15 g), and that of bogies – about 8.2 m/s^2 (0.8 g). The solution offered can decrease the vertical accelerations on the body of a boxcar by about 20 % in comparison with the standard structure. For a standard boxcar, the acceleration of the bearing structure in the mass centre was 1.9 m/s^2, and in the support areas on the bogies – 9.7 m/s^2.

The motion of the car can be estimated as excellent [11–13], and the normative value of the acceleration of the bearing structure in the mass centre is 4.4 m/s^2.

3. Results of strength calculations of the bearing structure of a box car

The strength of the body of a boxcar with elastic elements in the centre sill were defined with FEM in the SolidWorks Simulation software. The continual model of the body of a boxcar is given in Figure 7. Tetrahedrons were taken as the FE. The FEM was built on the basis of these elements as the bogie of the box car was taken as a solid body. We would like to note that the application of such elements gives a good convergence of the computer model and the physical experiment; it was proved by means of the research conducted by the authors regarding similar tasks. The optimal number of FEM elements was determined with the graphical-analytical method.

The number of FE was defined with the graphical-analytical method (Table 2).

Number of elements	234 833
Number of nodes	667 471
Maximal size of an element, mm	100
Minimal size of an element, mm	20
Percentage of elements with a ratio of sides less than three	6.81
Percentage of elements with a ratio of sides more than ten	51.1
Minimal number of elements in a circle	22
The ratio of an increase in the size of an element	1.8

Table 2. Characteristics of the continual model of the body of a boxcar.

Steel 09G2S was applied as the material for the body.

The calculation scheme of the body of a boxcar is given in Figure 8. The calculation scheme included that the body was under vertical load P_v and it was fully loaded with a conditional freight. The vertical loading includes two components: static – 667.08 kN and dynamic – 97.24 kN, which includes the acceleration obtained through the mathematical modelling. The vertical loading was applied to the horizontal surface of the centre sill. The model also included friction forces P_{fr}. The friction force was calculated by means of mathematical modelling in model (1) – (4) and amounted to 73.6 kN. The calculation of the friction force included the condition of the maximum efficiency for the solution proposed regarding the issue of lower dynamic loading.

The model was secured in the areas of support on the running gear parts.

The maximum stresses were found in the contact area between the transverse beam and the centre sill; they accounted for 175.1 MPa (Figure 9), whereas the admissible stress is 295 MPa. The maximum displacements occurred in the middle part of the centre sill; the value measured was 3.3 mm (Figure 10). Thus, the strength of the body of a boxcar was provided [11, 12].

The calculation scheme built (Figure 8) was used for the fatigue research of the body of a boxcar. The authors also found the most loaded zones in the body of a boxcar (Figure 11). These zones included the contact areas between the transverse beam and the center sills. The stress concentration areas for the most loaded zones of the open car frame were determined by means of the maximum equivalent stresses obtained through the Mises criterion with consideration of their cyclic occurrence. The limit stress value is equal to the yield strength of the material – 345 MPa.

The results of the research were used for determination of the biaxiality ratio of the body of a boxcar (Figure 12). This ratio characterizes a relationship between the minimum and maximum stresses occurring in the body of a boxcar. The design service life of a boxcar was defined according to the method presented in [14]:

$$T_p = \frac{(\sigma_{-1L}/[n])^m \cdot N_0}{B \cdot f_e \cdot \sigma_{ad}^m}, \tag{8}$$

where σ_{-1L} – endurance limit; n – allowable strength factor; m – fatigue curve exponent; N_0 – test base; B – coefficient that determines permanent work in seconds; f_e – efficient frequency of dynamic stresses; σ_{ad} – amplitude of dynamic stresses.

The amplitude of dynamic stresses is

$$\sigma_{ad} = \sigma_{sw}(k_{vd} + \psi_\sigma / K_\sigma), \tag{9}$$

where σ_{sw} – stresses caused by the static weight load; k_{vd} – vertical dynamics coefficient; ψ_σ – sensitivity coefficient; K_σ – overall fatigue strength reduction coefficient.

The calculation included the following input parameters: $\sigma_{-1L} = 245$ MPa; $n = 2$; $m = 8$; $N_0 = 10^7$; $B = 3.07 \times 10^6$ s; $f_e = 2.7$ Hz; $\sigma_{sw} = 142.4$ MPa; $k_v d = 0.35$; $\psi_\sigma / K_\sigma = 0.2$.

The calculation demonstrates that the design service life of the proposed body is about 40 years. Therefore, the value of the design service life obtained is about 20 % higher than the service life of a standard boxcar frame with a centre sill consisting of two Z-profiles. It should be noted that the service life obtained should be adjusted through both further research into the longitudinal loading on the body of a boxcar and experiments.

It should be noted that the fatigue of elastic elements was not included in the calculation of the service life of a boxcar, as this element can be replaced by a new one, if needed.

4. Conclusions

(1.) The authors suggested elastic elements in the centre sill being the basic carrying element of the

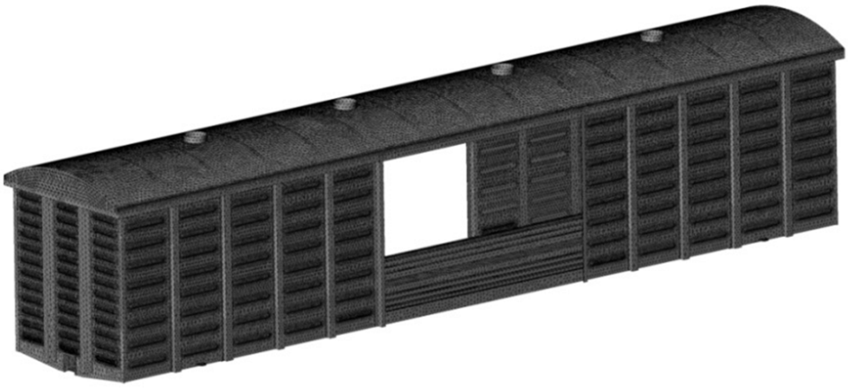

Figure 7. Continual model of the body of a boxcar.

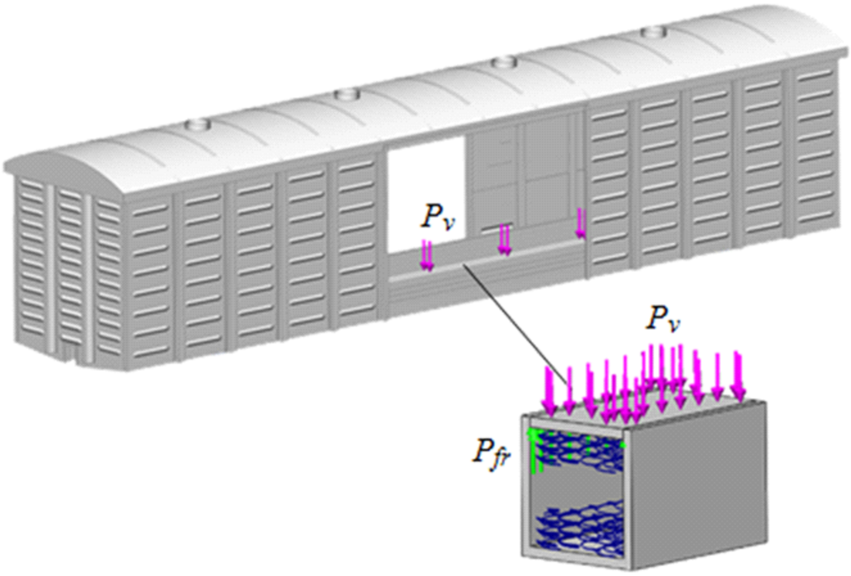

Figure 8. Calculation scheme of the body of a boxcar.

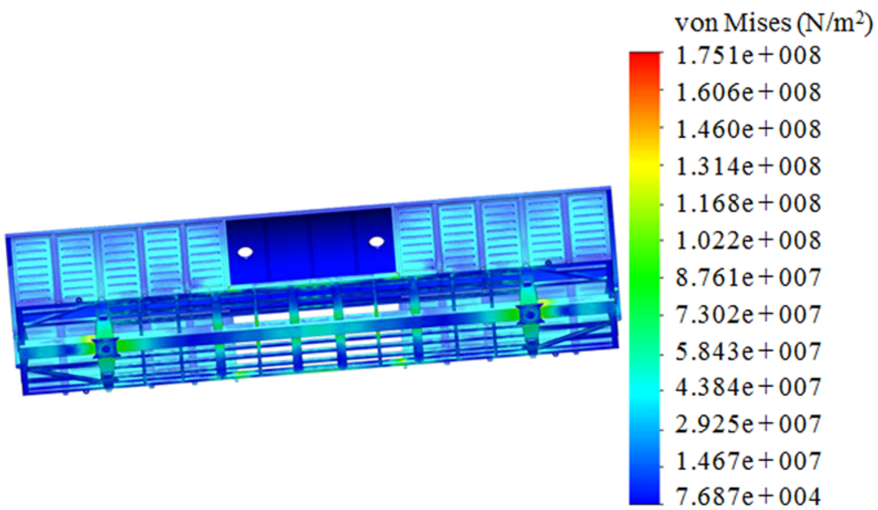

Figure 9. Stress of the body of a boxcar.

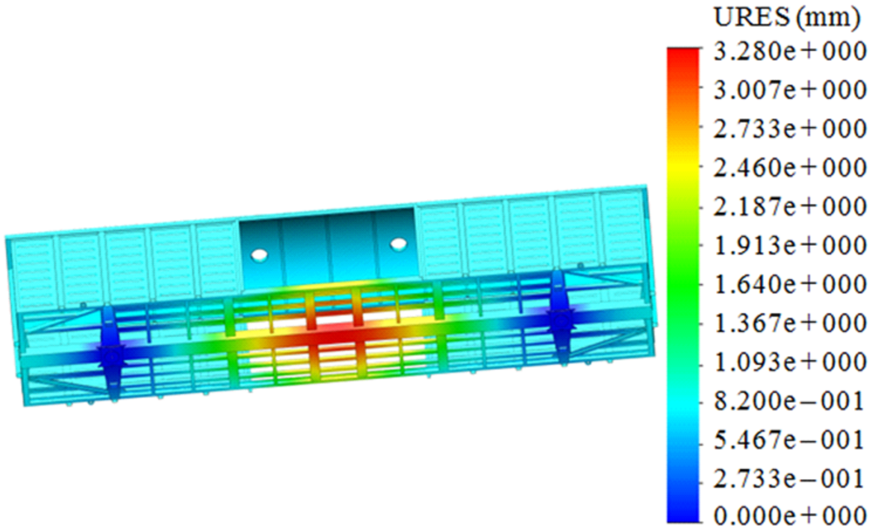

Figure 10. Displacements in the body of a boxcar.

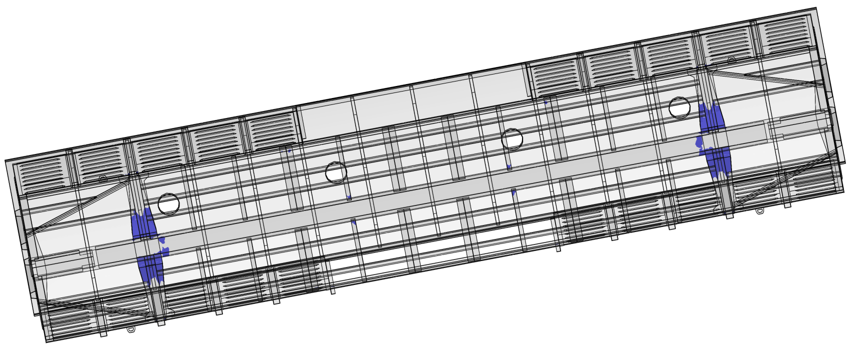

Figure 11. Most loaded zones in the body of a boxcar.

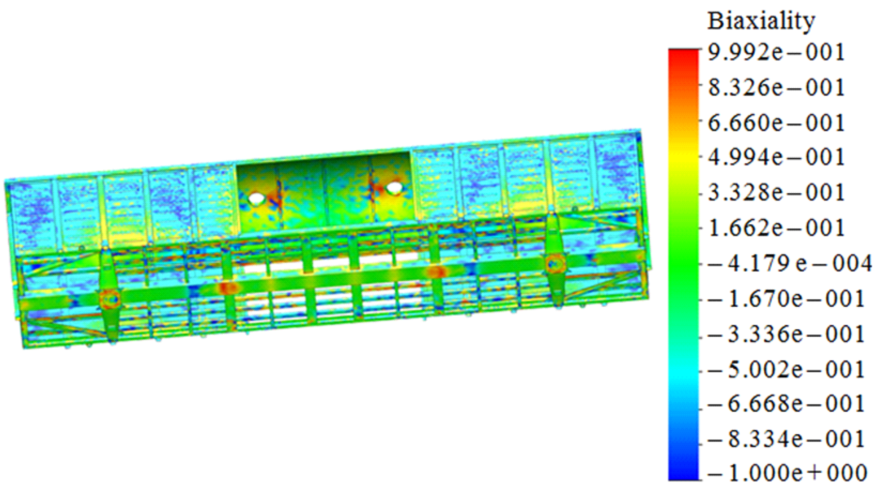

Figure 12. Biaxiality ratio in the body of a boxcar

frame. This solution can transform the loads on the body into the work of the dry friction forces between the components of the centre sill.

(2.) The research deals with the modelling of loads on the improved boxcar. The maximum acceleration of the body of an empty boxcar was 1.43 m/s^2 (0.15 g), and that of bogies was 8.3 m/s^2 (0.8 g). This solution can decrease the accelerations on the body of a boxcar by about 20 % in comparison to those of the prototype car. The motion of the car can be estimated as excellent.

(3.) The authors determined the basic strength characteristics of the body of a boxcar with consideration of the solution proposed. The calculation was made with FEM in the SolidWorks Simulation software. The greatest stresses occurred in the contact area between the bolster beam and the centre sill with a value of 175.1 MPa. The maximum displacements occurred in the middle of the centre sill and were equal to 3.3 mm. Thus, the strength of the body of a boxcar was provided.

(4.) The design service life of the improved boxcar was calculated as 40 years. Thus, the design service life obtained was 20 % longer than that of the prototype car. The research may be of value for those who are concerned about a more efficient operation of railway transport.

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