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Analysis of the Load of a Passenger Car Frame equipped with a New Concept for an Automatic Coupler

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Abstract

The presented article includes the study of the dynamics and strength of the frame of a passenger car equipped with a new concept of an automatic coupler. The peculiarity of this device is that it reduces the dynamic loads acting on the frame due to the elastic-dissipative forces. These forces arise in it when it receives the longitudinal load from the body from an automatic coupler. A mathematical model of the longitudinal load of the passenger car body was carried out to substantiate the proposed decision. It was established that the use of the designed automatic coupler will contribute to the reduction of the longitudinal acceleration by 3.8% compared with a standard automatic coupler. Together with this, the strength of the load-bearing structure of the passenger car was studied. The results of the calculations showed that the maximum stresses in the frame are lower by 3.2% in comparison with a standard automatic coupler. Further, a modal analysis of the passenger car frame was carried out as part of the study. It is established that the operational safety of the passenger car from the modal properties point of view is met, as the first eigenfrequency of oscillation exceeds 8 Hz. Moreover, the biaxiality indicator of the passenger car frame equipped with a new concept of an automatic coupler was determined. The conducted research will contribute to develop the improvement of the efficiency of the passenger cars operation, as well as maintaining the competitiveness of the railway industry.

Keywords: passenger car, automatic coupler, dynamic load, frame strength, modal analysis, biaxiality indicator.

1 Introduction

Railway transport is a leading branch of the transport network, which not only ensures a stable transportation process, but also the development of the country's economy. To maintain the competitiveness of the railway industry, it is important to provide it with a serviceable rolling stock. Special attention should be paid not only to freight railway vehicles, but also to passenger railway vehicles.

Currently, the operated fleet of 1520 mm gauge passenger cars are formed by cars with different design features and technical and economic indicators. The supporting structure of such railway vehicles is formed by a frame and a body. At the same time, the frame can contain a spinal beam along the entire length or only in the cantilever parts.

The load-bearing structure of passenger cars is affected by constant alternating vertical and longitudinal loads during their operation. The cyclic nature of loads causes the appearance of various defects and damages of the main frames (Figure 1).

a) b)

Figure 1: Damage of the main supporting structure of a passenger car - a frame: a), b), cracks of sheets; c) break of the welding seam; d) corrosive wear and cracks

The appearance of such damages directly during the wagons' operations can cause a violation of traffic safety. It should be noted that in recent years, the fleet of passenger cars with a gauge of 1520 mm has been slightly replenished. The presence of operational wear of its components under the condition of cyclic loads increases the probability of damage on during operation directly on a track (Figure 2).

a) b)

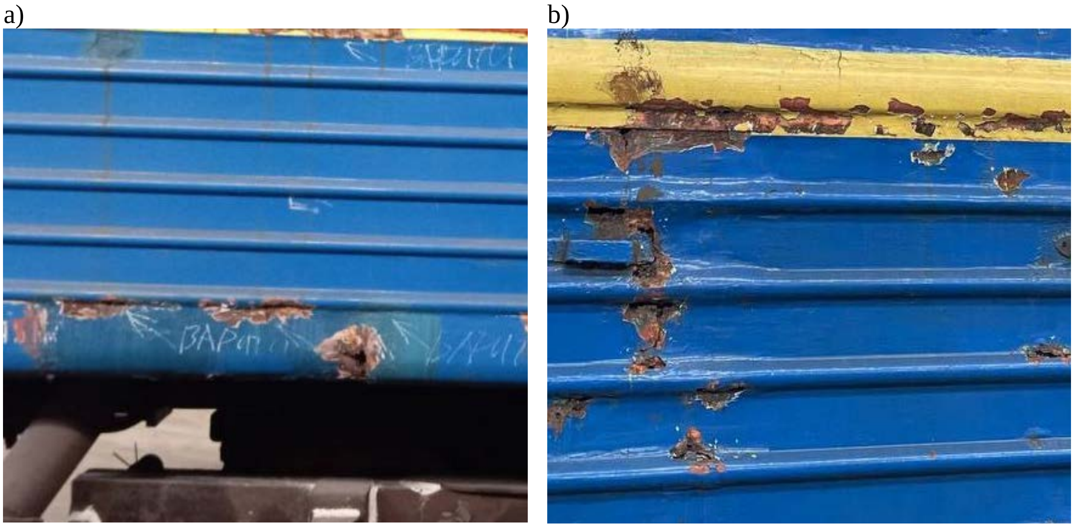

Figure 2: Corrosive wear of the supporting structure of a passenger car: a) lower binding; b) cladding of the side wall

In this regard, there is a need to create solutions aimed at improving the safety of passenger cars. This can be achieved by improving their structural units. Therefore, studies devoted to the issue of the improvement of the passenger cars design are really relevant.

2 Analysis of the recent research and publications

Many scientific publications are dedicated to the researches devoted to the improvement of the load-bearing structures of passenger cars. Thus, for instance, the peculiarities of improving the technique of technical diagnostics of passenger cars are highlighted in the work [1]. At the same time, the authors analysed the sequence of the research of stress distribution in the structural elements of a passenger car frame. The study was conducted on the example of passenger cars built by Kryukovsky Railway Car Building Works Inc. (Ukraine). Typical tests of wagons are considered. The results obtained by the authors made it possible to formulate recommendations for extending the service life of passenger cars beyond that set by a manufacturer. It should be said that the work does not propose solutions to reduce the load on a frame of a passenger car in the operation.

In order to improve the technical and economic indicators of passenger cars, the research [2] highlights the features of topological optimization of its supporting structure. The authors proposed a new procedure for topological constraints. At the same time, the preparation of an equivalent design space is assumed. The results of the conducted research proved the effectiveness of the proposed procedure. At the same time, these solutions do not contribute to reducing the dynamic load of the frame of the passenger car in the operation.

The analysis of existing and promising designs of passenger cars is given in the publication [3]. The authors evaluated the quality of the wagons according to the norms of smoothness of their movement on a railway track. The study was carried out on an example of a passenger car, the 61-920 model. The main attention is paid to the vertical loads acting on the wagon in the operation. Recommendations for the safe operation of passenger cars have been formulated. However, the issue of reducing the longitudinal load of passenger cars in operation was left out of the attention of the author's team.

The publication [4] is of scientific interest. It discusses the main approaches used to study the influence of passenger car designs on their stiffness and strength in the operation. The results of strength analyses of the load-bearing structure of a passenger car are presented. The obtained results were experimentally confirmed. The methodology proposed by the authors makes it possible to choose an effective structural scheme of the load-bearing structure of the passenger car. However, at the same time, no solutions were proposed to improve the strength of the frame of the passenger car, as the most loaded node of its supporting structure in the operation.

The analysis of methods for determining the stress state of railway wagons, both freight and passenger, is carried out in the research presented in [5]. The authors have mainly taken into account the methods of numerical analysis of the statics and dynamics of wagons. It should be noted that the models proposed in the framework of the study are verified by the results of experimental studies. The work also provides solutions that will contribute to improving the dynamics of railway wagons. However, no attention has been paid to improve the load-bearing structures strength of the wagons.

In order to reduce the longitudinal dynamic load of passenger cars and to improve their strength indicators, the improvement of the frame design is proposed in the paper [6]. It is proposed to manufacture a spinal beam made from profiles with a rectangular cross-section, which are filled with energy-absorbing material. Such a decision helps to reduce the stresses that arise in the spinal beam by 11.7% compared with its typical design. The results of the modal analysis of the improved passenger car frame are presented. The strength of the weld joints in the most loaded areas of the frame was evaluated. It is important to note, that such an implementation is quite difficult in practice from the point of view of maintenance and repairing of the passenger car frame. In addition, this solution can be implemented only at the stage of manufacturing new wagons, not at the stage of their modernization. This fact limits wider practical applications of this technical solution.

The method of creation and features of the design of a modern lightweight body of passenger cars are highlighted in the paper [7]. When, topological optimization was used during creating its design. It was carried out according to the criterion of minimum material capacity. Current Japanese and European standards are taken into account when drawing up the calculation diagram of the body. The obtained results proved the expediency of the design solutions adopted during the design. However, at the same time, the authors have not considered the possibility of reducing the load of the supporting structure of the passenger car during operational modes.

Further, the features of parametric optimization of the passenger car body are highlighted in the research [8]. The authors have used the ANSYS software package. A comparative analysis of the parameters of the typical and optimized design of the passenger car confirmed the effectiveness of the procedure. However, the authors limited themselves to optimizing the body and have not paid an attention to the frame. Apparently, this is because this optimization is mainly aimed at reducing the weight of the passenger car. Along with this, issues of optimization and improvement of the framework also need to be investigated.

The issue of the improvement of the safety of the passenger car movement is solved in [9]. It is considered the possibility of using a spring suspension, which allows increasing the frequency of oscillations of a wagon body. A dynamic analysis of the body of a passenger car was carried out. The expediency of the proposed solution for achieving the set goal within the framework of scientific research has been proven. It is important to note that the authors have not considered the impact of this spring suspension on the load of the passenger car frame.

Improving the strength of the supporting structure of the car is possible by reducing the dynamic loads acting on it during the operation. For example, the concept of creating a body with walls made of sandwich panels is proposed in the publication [10]. The authors highlighted the design of the sheets that make up the sandwich panel and the features of optimizing their parameters. There are presented the results of calculations, which have taken into account the proposed improvement. Analysis of these results has proved their effectiveness.

Further, the feasibility of using sandwich panels as body walls was suggested by the authors of the research presented in [11]. The peculiarities of the maintenance of a car with such walls, as well as the prospects for the development of this implementation, are indicated. It should be noted that an application of sandwich panels as the walls of the car body will really contribute to the reducing its load during operation. However, the authors of publications [10, 11] have not considered the possibility of reducing the load on the wagon frame.

The use of vacuum honeycomb panels on passenger rolling stock is substantiated in the article [12]. The main goal of this implementation was to reduce the tare of a passenger car, as well as to improve its thermal insulation characteristics. The results of the conducted research confirmed the possibility of achieving the goal specified by the authors. Though, any attention has not been paid to determining the impact of such panels on the overall strength of the load-bearing structure of a passenger car.

Improving the strength of the wagon frame is also possible by means of due an introduction of more modern designs of automatic couplers. For example, in the previous work of the author's team [13], the expediency of using a new concept of a automatic coupler on articulated type wagons was considered. The peculiarity of such wagons is, that their frame components are made of round pipes. The reduction of dynamic loads acting on the frame of the wagons is due to the viscous resistance forces arising in the backbone beam of the frame in which the concept of the proposed device is installed. The theoretical justification of such an implementation is confirmed on the main types of articulated wagons. However, this implementation has a number of significant drawbacks, at which, the high cost and complexity of maintenance and repair of wagons and devices in operation are the main of them.

It is important to note that the author team also conducted a patent analysis of the designs of passenger cars. Some of these patents are analysed below. For example, in the patent [14], a compartment-type passenger car is proposed, which contains a welded body. It is composed of a supporting frame made of a backbone beam and two pivot beams. The backbone beam is made of a double-sided fork type of variable cross-section. In the middle part between the pivot beams, the backbone is made in the form of one I-beam, which is rigidly connected to two channels protruding beyond the pivot beams. On the sides, the supporting frame is equipped with supporting channels and pivot beams and beams of the transverse set of the supporting frame rest on it. The invention allows to increase the base of the wagon while increasing the strength of its supporting frame.

The patent [15] proposed a supporting structure of a passenger car, which is all-metal and formed from a frame made in the form of a spineless beam and a power sheathing. This sheathing is formed from longitudinal power elements composed of the walls of a body and a roof. These components are made of steel sheets with longitudinal corrugations and rigidly connected to each other by means of weld joints. The structure is made of carbon steel.

The patent [16] describes the design of a passenger car, which the body frame is placed between transverse partitions. The length of the frame equals to the distance between these partitions. The transverse connections of the frame at their ends are made in the form of transverse support plates, connected from above at the floor level into a single unit by a horizontal transverse plate and power brackets. The supporting force plates placed in the planes of the transverse dividing partitions are located at the distance from the longitudinal plane of symmetry of the wagon. This approach ensures the passage of the moving part of the coupling device between them, and they are fastened with brace beams made solid. Power jacks are placed in the planes of the vertical walls of the brace beams, and they are fastened to the support plates and brace beams of the cantilever parts of the frame.

The disadvantage of these structures of passenger cars is the insufficient fatigue strength of the frame elements under the action of cyclic loads, which contributes to the appearance of cracks in them.

The literature review shows that the issue of improving passenger cars is currently quite relevant. However, to improve the efficiency of their operation, these issues require further research.

3 The purpose and the main goal of the research

The main goal of the research is to highlight the results of determining the dynamic load and strength of the passenger car frame with a new design of an automatic coupler. The following tasks are set to achieve the described goal:

- to propose the design a new concept of an automatic coupler;
- to investigate the longitudinal dynamic load of a passenger car with a new design of an automatic coupler;
- to determine the strength of the passenger car frame with a new design of an automatic coupler;
- to conduct the modal analysis and to determine the biaxiality indicator of the passenger car frame with a new design of an automatic coupler.

4 Features of the design and principle of operation of the new design of an automatic coupler

In the process of operation, transmission and damping of longitudinal traction forces in the train is carried out with the help of an auto-coupling device [17, 18]. Currently, rolling stocks (freight and passenger wagons) of railways with a gauge of 1520 mm mostly uses an automatic coupler, the SA-3 type (Figure 3) [19]. The type of an absorption device is the main distinguishing feature depending on the purpose of the wagon.

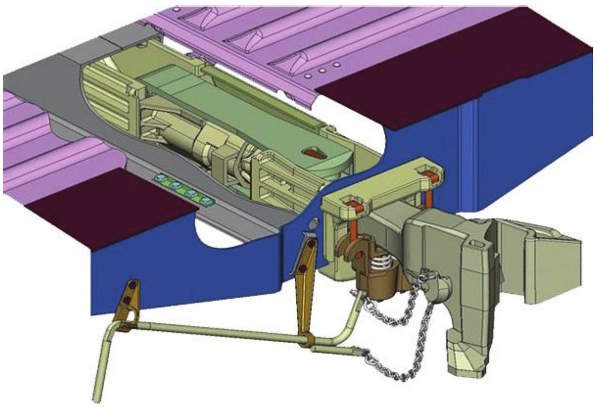

Figure 3: An automatic coupler, the SA-3 type.

It is important to note, that this device has a number of structural and operational shortcomings. The complexity of its design and the maintenance is one of the most significant of them.

In order to ensure the strength of the passenger car frame and to reduce the costs of its maintenance, a new automatic coupler (Figure 4) is designed.

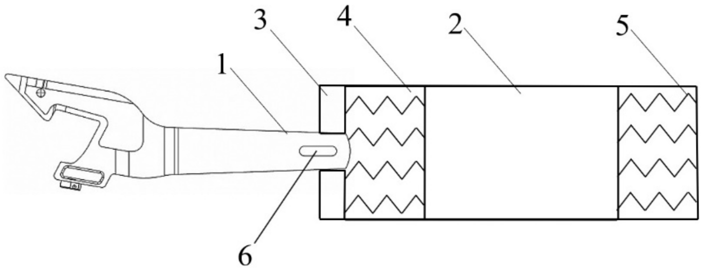

Figure 4: A new design of an automatic coupler: 1 – a body of an automatic coupler; 2 – a pusher; 3 – a device body; 4, 5 – springs; 6 – a wedge.

This device will be removable, which will allow its use not only on new wagons, but also on the operated fleet of wagons. The dynamic loads will be reduced due to elastic-frictional forces arising in it when it receives a longitudinal load from the body of the auto coupling. At the same time, it is possible to use a typical SA-3 wagon bearing structure. The operational principle of the device works is as following (Figure 4): the longitudinal load from the body 1 is transferred to the pusher 2, which moves relative to the device body 3. At the same time, springs 4 are stretched and springs 5 are compressed. Between the walls of the housing 3 of the device and the pusher 2, frictional forces arise. Due to this, the kinetic energy of the impact/jerk (compression/stretching) is extinguished during the movement of the wagon in a trainset. The body of the automatic coupler 1 interacts with the body 3 of the device using a wedge 6.

The location of the coupler on a wagon is identical to a standard coupler, i.e. in the console parts of the frame (Figure 5).

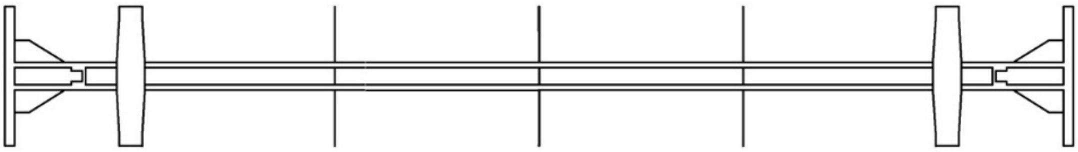

Figure 5: A placement of the device on a passenger car frame.

5 Study of the longitudinal dynamic load of a passenger car with a new concept of an automatic coupler

The longitudinal dynamic load of the proposed coupler in the passenger car frame was determined to substantiate its implementation. A calculation diagram of a passenger wagon is shown in Figure 6.

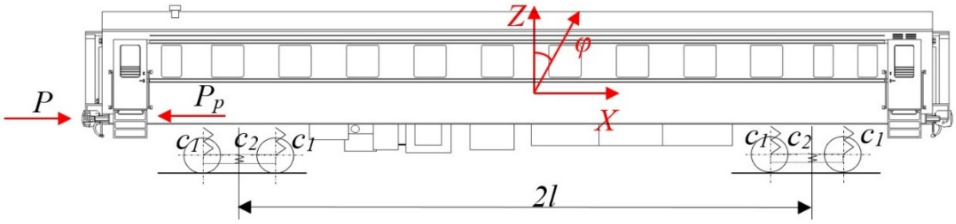

Figure 6: A calculation scheme of a passenger car.

At the same time, the model described in [20] was used. It characterizes the longitudinal load of a platform wagon loaded by the tank containers. As a part of this study, a specified model was refined to determine the dynamic load of a passenger car equipped with new concepts of an automatic coupler. It is taken into account that the passenger car has three degrees of freedom: in the longitudinal, lateral and vertical planes. The case of the longitudinal load by an automatic coupler was investigated - a shock of 2.5 MN. It is taken into account that typical bogies of the KVZ - TsNII type, 1 model are used for the wagon. These bogies have over-axle and cradle spring suspension. Then, the equations of motion of the wagon will have the following form:

$$\left(M_V + 2 \cdot m_B + \frac{n \cdot I_W}{r^2}\right) \cdot x'' + \left(M_V \cdot h\right) \cdot \varphi'' = P - P_P, \tag{1}$$

$$\begin{aligned} & I_V \cdot \varphi'' + M_V \cdot h \cdot x'' - g \cdot \varphi \cdot M_V \cdot h = \\ & = l \cdot F_{FR} \cdot \left(sign\left(z - l \cdot \varphi\right)' - sign\left(z + l \cdot \varphi\right)' \right) + \\ & + l \cdot \left(\left(c \cdot \left(z - l \cdot \varphi\right)\right) - \left(c \cdot \left(z + l \cdot \varphi\right)\right)\right), \end{aligned} \tag{2}$$

$$\begin{aligned} & M_V \cdot \cancel{z}'' = \left(c \cdot \left(z - l \cdot \varphi\right)\right) + \left(c \cdot \left(z + l \cdot \varphi\right)\right) - \\ & - F_{FR} \cdot \left(sign\left(z - l \cdot \varphi\right)' - sign\left(z + l \cdot \varphi\right)' \right), \end{aligned} \tag{3}$$

where M_V is the weight of the load-bearing structure of the passenger car with a coupler; I_V – the moment of inertia of the passenger car relative to its longitudinal axis; P is the compressive force acting on the automatic coupler; P_P - the force of the reaction of the device to the action of force P; m_B is the mass of the wagon bogie; I_W is the moment of inertia of a wheelset; r is a wheel radius; n is the number of the bogies; l – a half of wheel-base of the passenger car; F_{FR} – the force of dry friction in the spring suspension of the bogie; c – total stiffness of spring suspension; x, φ, z are coordinates characterizing a longitudinal movement, an angular movement around the transverse axis and a vertical movement of the wagon, respectively.

The equivalent stiffness of the springs of the bogie suspension system was determined by the known method [21, 22]:

$$c = \frac{c_1 \cdot c_2}{c_1 + c_2}, \tag{4}$$

where c_1, c_2 are the stiffness of the springs of primary and secondary suspension, respectively.

Equations of motion (1) – (3) were solved in the MathCad software [23, 24], which implements the Runge–Kutta method [25, 26]. When we enter such notations, then we can write [27]:

$$\frac{d}{dt}y_2 = y_5, \tag{5}$$

$$\frac{d}{dt}y_3 = y_6, \tag{6}$$

$$\frac{d}{dt}y_4 = \frac{P - P_p - (M_V \cdot h)\cdot y_5'}{\left(M_V + 2\cdot m_B + \frac{n\cdot I_W}{r^2}\right)}, \tag{7}$$

$$\frac{d}{dt}y_5 = \frac{\begin{array}{c} l\cdot F_{FR}\cdot\left(sign(y_3 - l\cdot y_2)' - sign(y_3 + l\cdot y_2)'\right) + \\ +l\cdot\left(\left(c\cdot(y_3 - l\cdot y_2)\right) - \left(c\cdot(y_3 + l\cdot y_2)\right)\right) - M_V\cdot h\cdot y_4' - g\cdot y_2\cdot M_V\cdot h \end{array}}{I_V}, \tag{8}$$

$$\frac{d}{dt}y_6 = \frac{\begin{array}{c} \left(c\cdot(y_3 - l\cdot y_2)\right) + \left(c\cdot(y_3 + l\cdot y_2)\right) - \\ -F_{FR}\cdot\left(sign(y_3 - l\cdot y_2)' - sign(y_3 + l\cdot y_2)'\right) \end{array}}{M_V}. \tag{9}$$

With this in mind:

$$D(t,y) = \begin{bmatrix} y_4 \\ y_5 \\ y_6 \\ \dfrac{P - P_p - (M_V\cdot h)\cdot y_5'}{\left(M_V + 2\cdot m_B + \frac{n\cdot I_W}{r^2}\right)} \\ \dfrac{\begin{array}{c} l\cdot F_{FR}\cdot\left(sign(y_3 - l\cdot y_2)' - sign(y_3 + l\cdot y_2)'\right) + \\ +l\cdot\left(\left(c\cdot(y_3 - l\cdot y_2)\right) - \left(c\cdot(y_3 + l\cdot y_2)\right)\right) - M_V\cdot h\cdot y_4' - g\cdot y_2\cdot M_V\cdot h \end{array}}{I_V} \\ \dfrac{\left(c\cdot(y_3 - l\cdot y_2)\right) + \left(c\cdot(y_3 + l\cdot y_2)\right) - F_{FR}\cdot\left(sign(y_3 - l\cdot y_2)' - sign(y_3 + l\cdot y_2)'\right)}{M_V} \end{bmatrix}, \tag{10}$$

$$T = rkfixed(Z_0, tn, tk, n, D),$$

where Z_0 is a vector with initial conditions, *tn* and *tk* are initial and final integration variables, respectively, *n*' is the number of fixed steps and *D* is a symbol vector:

$$ddq_{j,1} = \frac{P - P_p - (M_V \cdot h) \cdot y_5'}{\left(M_V + 2 \cdot m_B + \frac{n \cdot I_W}{r^2}\right)}, \tag{10}$$

$$ddq_{j,2} = \frac{\begin{array}{c} l \cdot F_{FR} \cdot \left(sign(y_3 - l \cdot y_2)' - sign(y_3 + l \cdot y_2)' \right) + \\ + l \cdot \left((c \cdot (y_3 - l \cdot y_2)) - (c \cdot (y_3 + l \cdot y_2)) \right) - M_V \cdot h \cdot y_4' - g \cdot y_2 \cdot M_V \cdot h \end{array}}{I_V}, \tag{10}$$

$$ddq_{j,3} = \frac{\begin{array}{c} (c \cdot (y_3 - l \cdot y_2)) + (c \cdot (y_3 + l \cdot y_2)) - \\ - F_{FR} \cdot \left(sign(y_3 - l \cdot y_2)' - sign(y_3 + l \cdot y_2)' \right) \end{array}}{M_V}. \tag{10}$$

The initial conditions are set up close to zero. Based on the calculations, the magnitude of the longitudinal acceleration acting on the load-bearing structure of the passenger car body was about 23 m/s² (Figure 7). The obtained value of longitudinal acceleration is lower by 3.8% than that acting on a passenger car with a typical scheme of perception of the longitudinal loads.

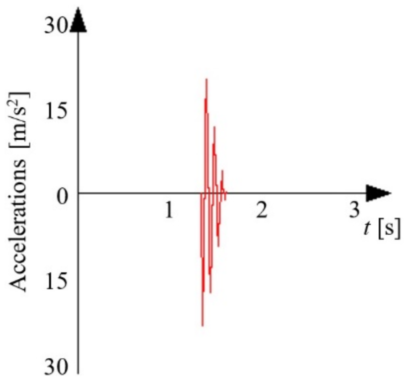

Figure 7: Acceleration of the passenger car in a centre of gravity.

This acceleration is considered for the calculation of the strength of the passenger car frame.

6 Determining the strength of the passenger car frame with the new concept of an automatic coupler

To determine the strength of a spatial model of the passenger car frame equipped with a new concept of an automatic coupler was built to determine its strength. The model 61-821 passenger car frame was chosen as a prototype. The passenger frame scheme is shown in Figure 8. It consists of a spinal beam 1, two pivot beams 2, two end beams 3, as well as three transverse beams 4 [17, 18]. The spinal beam is formed from three component parts. Two end parts of the spinal beam are made of U-profile (No. 30B) and the middle part is made of U-profile (No. 30a). These profiles differ to each other by their wall thickness, namely, U-profile No. 30B has a wall thickness of 9.5 mm and U-profile No. 30a has a wall thickness of 6.5 mm. The channels of the spinal beam along its length are connected by diaphragms 5.

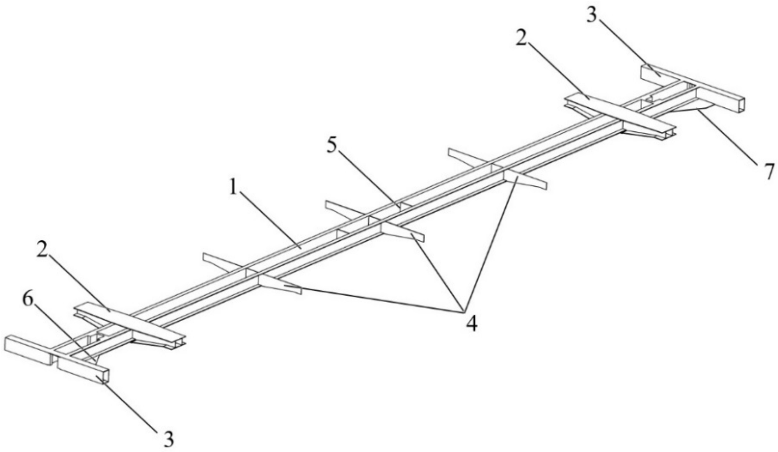

Figure 8: The passenger car frame.

The pivot beam has a box section and it is formed by sheets of different thicknesses: the vertical sheets are 8 mm thick, the bottom sheet is 10 mm thick. The end beams consist of two U-profiles (No. 30). Transverse beams are made of L-profiles, which have a variable height shape and the thickness of 6 mm. Horizontal diaphragms 6 are installed in the cantilever parts of the frame. The length of the frame behind the end beams is 23.457 m. The frame is made of low-alloy steel 09G2D [28]. A three-dimensional model of the frame was created in the SolidWorks software [29, 30] and it is shown in Figure 9.

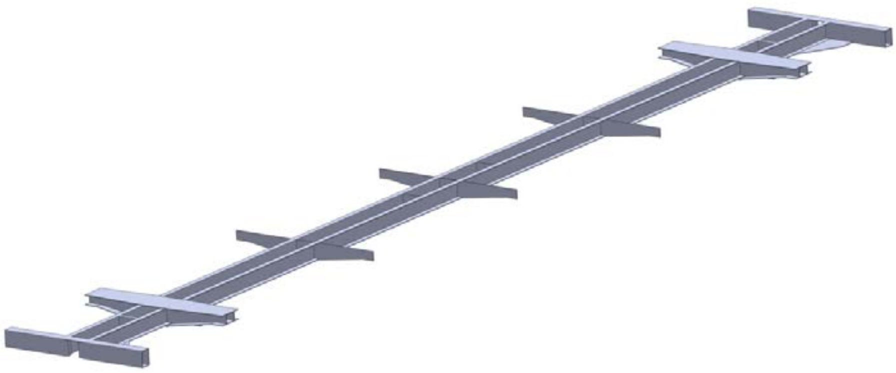

Figure 9: A three-dimensional model of the frame.

The strength calculation was also carried out in SolidWorks Simulation software using the finite element method. The finite element model is formed by tetrahedral modelling elements. Their total number was 14,192 and the number of nodes was 5,203. The optimal number of elements was calculated by graphical-analytical method. In the areas where the frame rests on the bogies, i.e. body slides, rigid connections were placed. When drawing up the calculation scheme, it was taken into account that the frame is exposed to the load P_b. It includes the vertical static and dynamic load, as well as the longitudinal forces P_p (Figure 10).

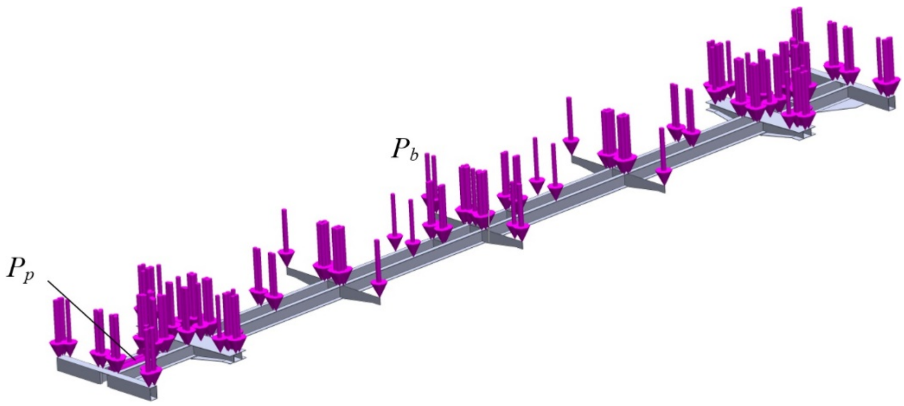

Figure 10: A calculation scheme of the frame.

It was established based on the calculations, that the maximum equivalent stresses occur in the spinal beam, namely in its cantilever part (Figure 11).

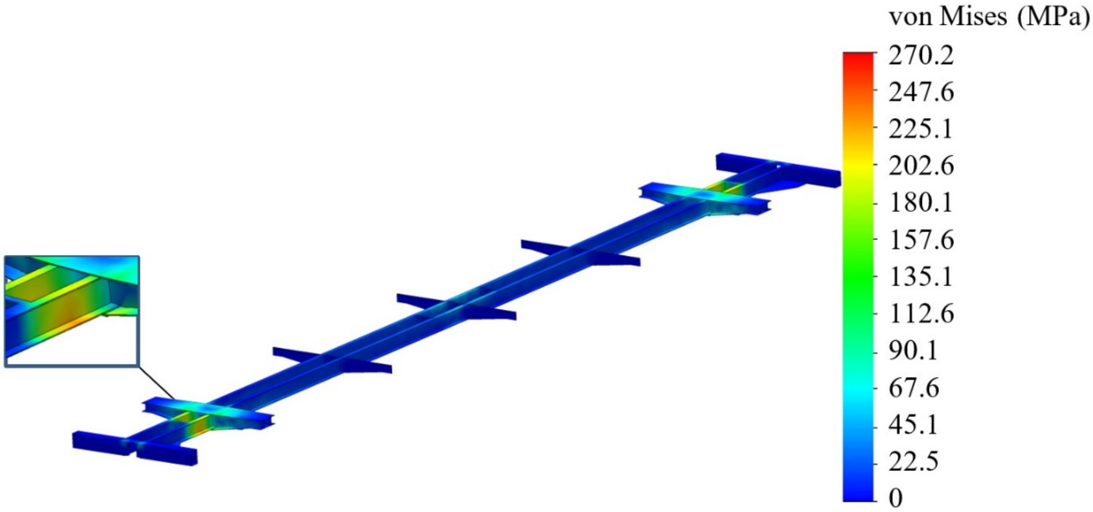

Figure 11: A distribution of stresses in the passenger car frame.

The maximum stresses occur in the lower shelf of the profile of the frame backbone beam and the values amounted to about 270 MPa. It is by 3.2% lower than those occurring in a standard frame design. The most loaded areas of the frame are shown in Figure 12.

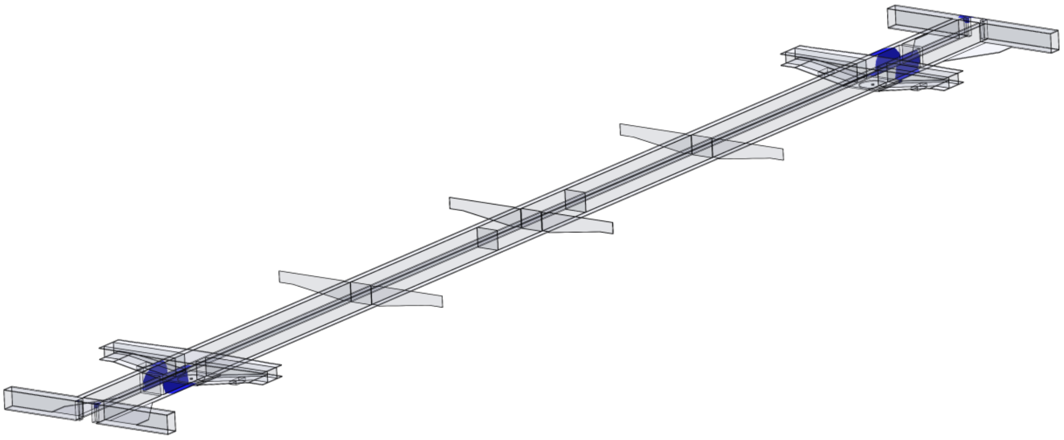

Figure 12: The most loaded areas of the frame.

The stress distribution in the backbone beam of the frame is shown in Figure 13. Therefore, the maximum stresses occur in the cantilever parts of the backbone beam. From the pivot cross-sections of the frame, the stresses decrease slightly and increase again in the middle part of the frame to about 170 MPa. The maximum deflections of the frame take place in its middle part and amount to the value of 11.7 mm (Figure 14).

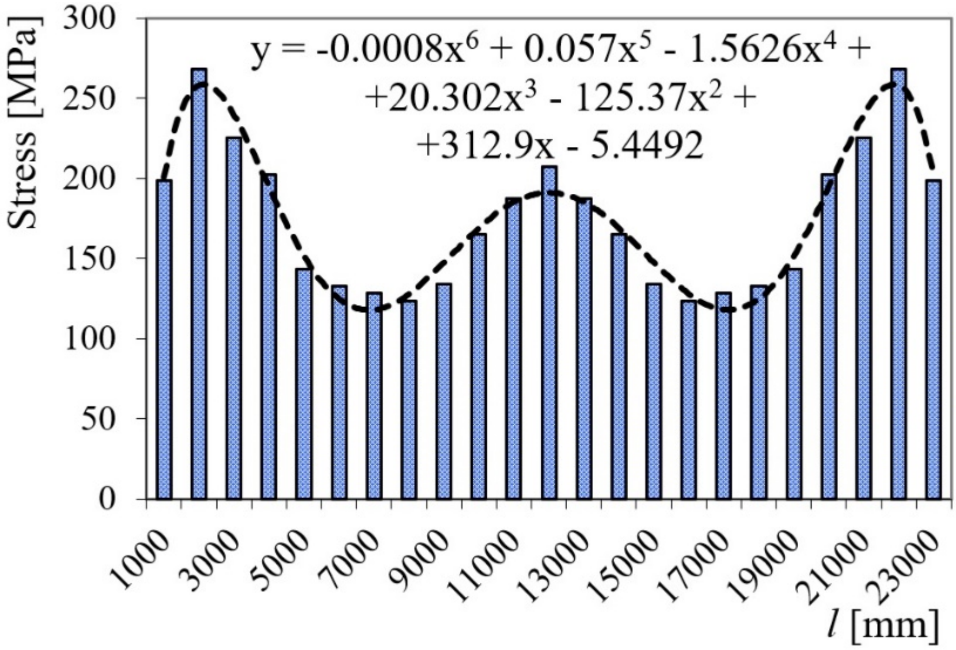

Figure 13: A distribution of stresses along the length of the spinal beam.

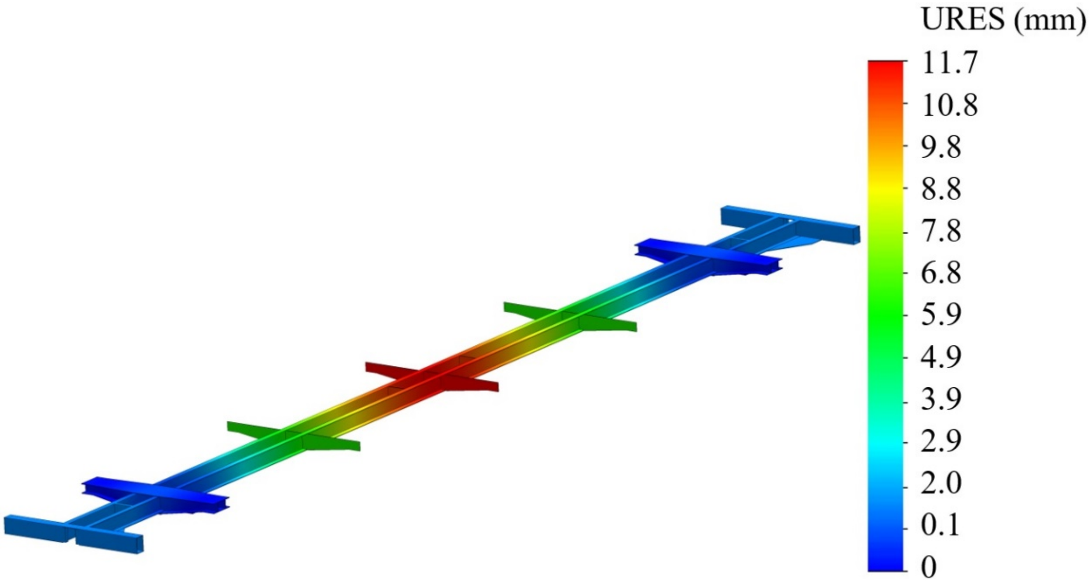

Figure 14: Deflections in frame nodes.

This distribution of deflections fields can be explained by the fact that the frame rests on slides, and its middle part experiences a uniformly distributed load. In this regard, the maximal deflections of the frame appear in its middle part. The smallest values of deflections occur in the pivot beams (Figure 15), as the model was fixed using the sliders.

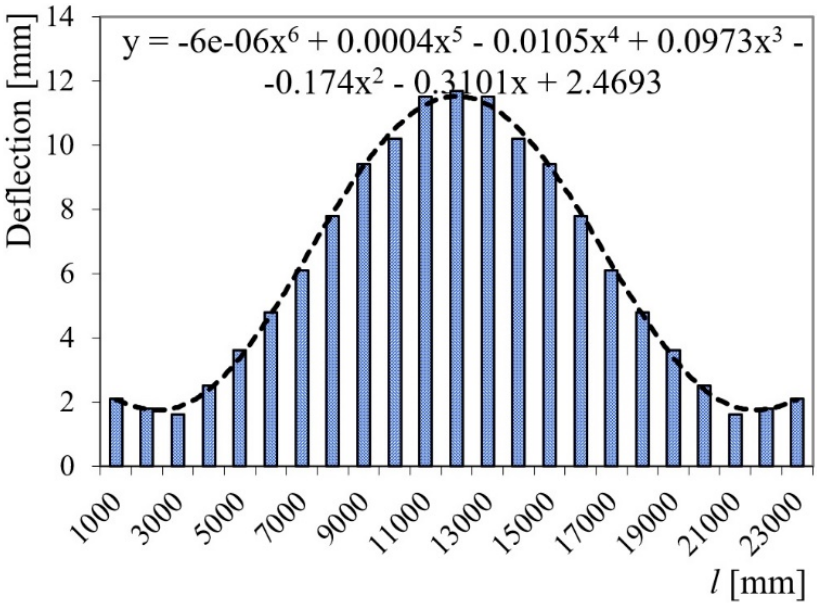

Figure 15: A distribution of movements along the length of the spinal beam.

The calculation was also carried out for other frame load modes in accordance with the regulatory document [28]. The obtained results are summarized in Table 1.

1st mode			3rd mode	
shock/compression	jerk	stretching	shock/compression	jerk/ stretching
Tension [MPa]				
270.2	265.3	257.4	223.6	212.4
Deflection [mm]				
11.7	11.7	11.7	11.6	11.6

Table 1: The results of the calculation of the strength of the passenger car frame.

Analysis of the data given in Table. 1 allows us to conclude that the strength of the passenger car frame is ensured during operation modes in accordance with [28].

7 Modal analysis and indicator of biaxiality of the passenger car frame with a new concept of an automatic coupler

Based on the calculation scheme shown in Figure 10, the eigenfrequences and eigen modes of the passenger car frame are determined. The most important ones are shown in Figure 16. The transparent colour (Figure 16) shows the static position of the frame (undeformed shape), and the matte one shows the deformations for individual eigenmodes. Based on the analysis of the obtained data, it can be concluded that the safety operation of the passenger car from the point of view of modal analysis is observed, as the first natural frequency of oscillations exceeds 8 Hz [28].

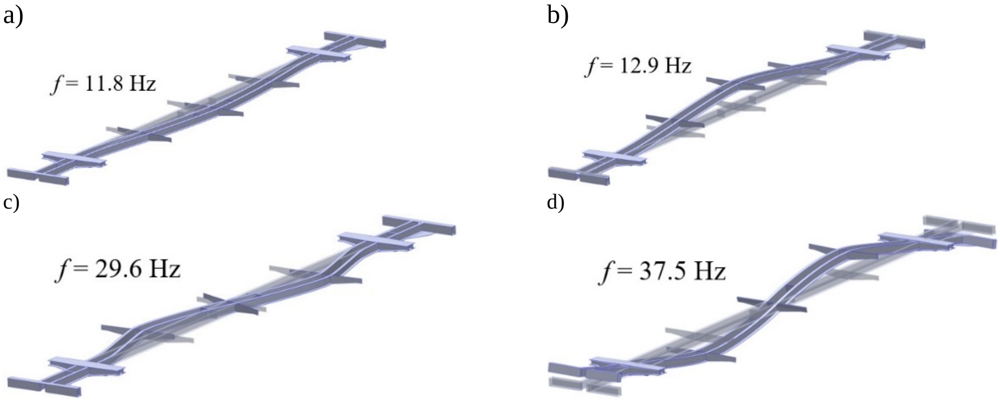

Figure 16: The results of the modal analysis of the passenger car frame (a scale 20:1): a) 1st mode, b) 2nd mode, c) 3rd mode, d) 4th mode.

Also, as part of the study, the biaxiality indicator of the passenger car frame was determined using the options of the SolidWorks Simulation software. This indicator characterizes the ratio of stresses, i.e. the ratio of minimal to maximal values, which occur in the frame. The calculation was carried out based on the results of the performed static analysis (Figure 10). A mandatory component of this calculation is the availability of the fatigue curve of the frame material. The fatigue curve was obtained according to the ASME carbon steel curve [29, 30]. The calculation results are shown in Figure 17.

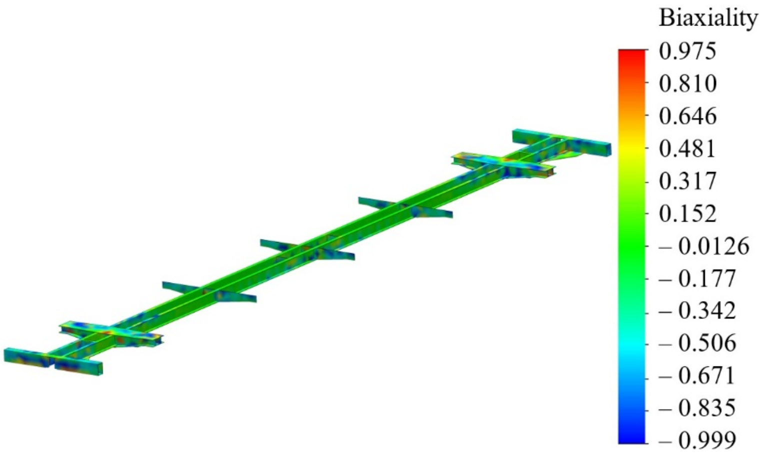

Figure 17: The frame biaxiality indicator.

It is possible to conclude based on analysis of the results shown in Figure 17, that the largest value of the biaxiality indicator by modulus occurs in the transverse beams. This is explained by the fact that the greatest stresses in the frame occur in the centre sill, and in the intermediate ones, these stresses are negligible. Therefore, the largest value of the biaxiality indicator can be traced in them.

8 Conclusions

1. The design of a new concept of an automatic coupler is proposed. This device works based on the principle of a removable module, which allows its use not only on new wagons, but also on the operated fleet of passenger cars. The reduction of dynamic loads acting on the load-bearing structure of the passenger car equipped with the automatic coupler concept occurs due to the elastic-frictional forces. They will arise in the structure, when it receives the longitudinal load from the coupling body.

2. The longitudinal dynamic load of a passenger car with a new concept automatic coupler was studied. It was established that the magnitude of the longitudinal acceleration acting on the load-bearing structure of the passenger car body is about 23 m/s^2. It is important to note that the obtained value of longitudinal acceleration is by 3.8% lower than that acting on a passenger car with a typical scheme of the longitudinal loads.

3. The strength of the passenger car frame with a new concept of an automatic coupler was determined. The maximum stresses were recorded in the lower shelf of the profile of the frame backbone beam and amounted to the value about 270 MPa, which is by 3.2% lower than those occurring in a typical design. The stress distribution along the length of the frame backbone beam was studied. At the same time, the maximum stresses take place in its cantilever parts. From the pivot cross-sections of the frame, the stresses decrease slightly and increase again in the middle part of the frame to the value about 170 MPa. The maximum deflection of the frame take place in its middle part and amount to the value of 11.7 mm.

4. The modal analysis of the passenger car frame with a new concept of an automatic coupler was carried out. The results of the conducted modal analysis showed that the safety of operation of the passenger car from the point of view of the modal analysis is observed, as the first eigenfrequency of oscillations exceeds the value of 8 Hz. The biaxiality indicator of the passenger car frame is determined. It was established that its greatest value in terms of modulus occurs in transverse beams.

The conducted research will contribute to the creation of developments in improving the efficiency of the operation of passenger cars, as well as maintaining the competitiveness of the railway industry.

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