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Special Features of Determining the Loads on the Passenger Car Frame with Closed Center Sill

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Abstract. This paper aims at an improvement in the frame of a passenger car for its higher efficiency. A special feature of this improvement is the closed configuration of the center sill that can be formed with sheets. The mass of the center sill can be minimized through its optimization with linear programming. The results of the calculation demonstrated that the mass of the frame in terms of the solutions proposed was 1.7% lower than that of the standard structure. The paper presents the results of the strength calculation of the frame of a passenger car at main loading modes. It was established that the strength of the frame at the schemes of loading studied was ensured. The motion of the improved frame of a car was estimated with the vertical accelerations on the frame. The motion was estimated as excellent. The research also included the modal analysis of the frame of a passenger car. The research can be used by those who are involved in the design of the bearing structure of a passenger car with better technical and operational characteristics that can increase the operational efficiency of passenger cars.

INTRODUCTION

The development of transport infrastructure is an integral part of the effective operation for a country's economy. Only effective use of all its components can guarantee the coordinated work of the transport industry. For a long time railway transport has been the most important and competitive component of the industry. The leading position of railway transport in the transportation sector can be maintained through the introduction of highly efficient and resource-saving transport facilities. And these decisions must be taken not only for freight but also for passenger transport, which provides considerable part of all services.

It should be noted that the bearing structure of a passenger car is the most highly loaded unit in operation. And it is the frame that suffers mostly from the main defects including those caused by the cyclic nature of the loads. The most common failures of the passenger car frame are cracks and deformations of its components. These defects can pose a threat to safe operation and be dangerous for the life and health of the passengers. Therefore, the improvements to the bearing structure of a passenger car can provide better efficiency of passenger transportation.

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The special structural features of passenger car bodies are presented in [1]. The research was made for a passenger car intended for high-speed transportation. The authors proposed measures for lower material capacity of the bearing structure of a passenger car by decreasing the thickness of the walls and the roof. The results of the strength calculation confirmed the effectiveness of the solutions suggested. However, the issue of improvements to the car frame was not discussed in the publication.

The special features of topological optimization of the bearing structure of a passenger car are described in [2]. The authors proposed the use of composite materials to decrease the mass of the bearing structure of a passenger car. They also substantiated the effective use of this optimization methodology. However, the authors did not address the issue of optimization of the frame as the most highly loaded unit of the bearing structure of a passenger car.

An algorithm of optimization of passenger car bodies through the application of steel vacuum panels is given in [3]. It has been found that these panels can decrease the tare of a car and provide the required operational capacity; besides they can improve the thermal insulation characteristics. The authors also presented scenarios of how to introduce steel panels in the body structure. However, they did not study the issue of improvements in the passenger car frame.

The choice of a new profile for the center sill of a car is given in [4]. The paper gives a number of structural solutions for car frames of various types. The results of the strength calculation for the bearing structure of an improved car are also presented. However, this solution was given only for the frame of a freight car.

The choice of structural solutions for the car elements with low tare mass is given in study [5]. The method included the comparative estimation of the values of strength, endurance and stability of the bearing structures of cars made of various materials. It should be noted that these solutions were realized only for the bearing structures of freight cars and did not include the bodies of passenger cars.

Research into the dynamic loading of the passenger car body transported by a train ferry is described in [6]. The numerical values of the dynamic loads on the body of a passenger car were determined by means of mathematical and computer modelling. The authors presented the results of modal analysis for the body. However, they did not suggest any measures for improvements in the body of a passenger car to improve its strength.

The use of aluminum high-strength alloys in passenger car bodies to decrease their tare weight, provided that the strength characteristics are ensured, is substantiated in [7, 8]. The authors described the advantages and disadvantages of using these alloys and the prospects of using aluminum alloys in machine building, particularly, in rail car production. However, they did not examine the issue of lower tare of cars through optimization of their structure.

The analysis of literature makes it possible to conclude that the issues of optimization and improvements of the bearing structures of a car are rather urgent. And they are to be studied in order to obtain higher operational efficiency.

The objective of the research is substantiation of improvements in the passenger car frame with the closed configuration of its center sill. The following tasks were set to achieve this objective:

- to optimize the center sill of the passenger car frame in accordance with the criterion of minimum material capacity;
- to calculate the strength of the passenger car frame; and
- to determine the accelerations to the passenger car frame and make its modal analysis.

PRESENTATION OF THE MAIN CONTENT OF THE PAPER

A 61-821 passenger car was chosen as the prototype for improvements in the frame. The metal framework of the car body consisted of a frame, side and end walls, and a roof. The sidewalls had openings for windows and doors. The end walls had openings for the gangway for passengers. Those openings were framed with metal to mount sealing strips (bellows). Deflectors were located on the roof.

The car frame consisted of the center sill, two bolster beams, two end and three bearers (Fig. 1). The center sill was formed of three parts – two end components of U-profile №30B and the middle component – of U-profile №30A [9, 10].

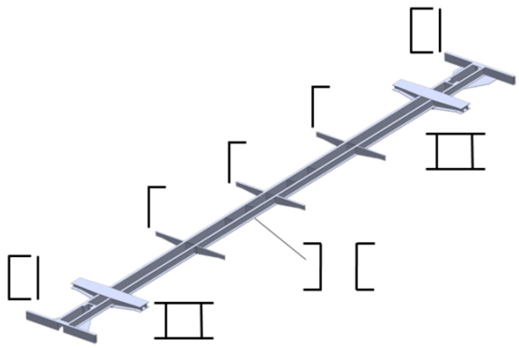


FIGURE 1. Passenger car frame with the center sill.

The material capacity of the center sill in the frame can be decreased through its optimization by the criterion of minimum material capacity. For this purpose a box-section profile formed of sheets (Fig. 2) can be used instead of the U-profile. The optimal thickness of the sheet for this center sill was determined through linear programming [11-14].

(a) (b)

FIGURE 2. Cross-section of the center sill of the frame: (a) standard structure; (b) improved structure.

It was assumed that thickness t of the vertical sheets, which form the center sill, is identical to the thickness of the profile of the standard structure. A variational calculation was made for the horizontal sheets in terms of their strength with consideration of their various thickness t.

Thus, the target function $m = m(t)$ was minimized under the constraints

$$\sigma \le [\sigma] = 345 \text{ MPa}. \tag{1}$$

Obviously, function $m(t)$ is linear:

$$m(t) = t \cdot a^2 \cdot \rho \Rightarrow \min, \tag{2}$$

where a – the side of the beam; ρ – the material density (for steel ρ = 7800 kg/m^3).

The mass of the center sill was determined with the options of SolidWorks, in which the graphic work was made. The strength of the passenger car was determined with the finite elements method realized in SolidWorks Simulation [15-18]. The design diagram represents the frame loaded with vertical load P_v conditioned by the weight of the passengers and the freight, and longitudinal load P_l on the frame from the coupler (Fig. 3). The calculation was made for an impact (Design mode I) as the most loaded condition. A load of 2.5 MN was applied to the rear draft lug of the coupler.

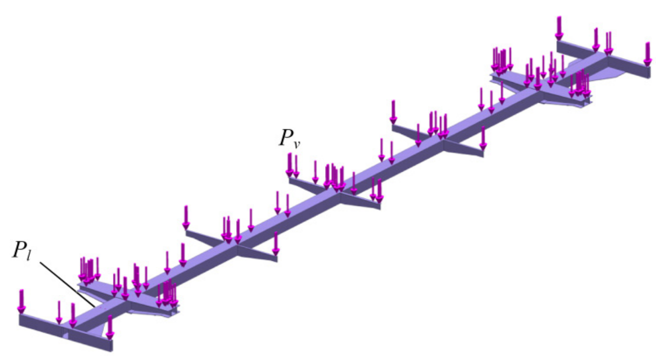


FIGURE 3. Design diagram of the frame.

The finite element model of the passenger car frame was formed with spatial isoparametric tetrahedrons [19-22]. The number of units in the mesh was 9100, and the number of the elements – 27356. The maximum size of an element was 120 mm, the minimum size – 24 mm. The model was fastened to the slides. Low-alloy Steel 09C2Cu with linear isotropic properties was used for the frame. The results of the calculation are given in Table 1.

TABLE 1. The primary parameters for determining the optimal thickness of the sheet for the center sill.

Measurement number	Thickness *t*, mm	Stress *σ*, MPa	Mass *m*, kg
1	13.5	298.5	2567.92
2	12.5	302.4	2439.06
3	11.5	315.6	2310.2
4	10.5	322.1	2181.35
5	9.5	328.6	2052.49
6	8.5	333.5	1923.64
7	7.5	338.4	1794.78
8	6.5	342.0	1665.92
9	5.5	348.6	1537.07
10	4.5	359.4	1408.21

The optimal number of calculations was found using the Student criterion [23]:

$$n = \frac{f^2 \cdot \sigma'^2}{\delta^2}, \tag{3}$$

where f is determined from the ratio $\Phi(f) = \gamma/2$; $\Phi(f)$ – Laplace's function, tabular value; σ' – the root mean square deviation of a random value under study, known a priori before the experimental measurements; δ^2 – the absolute deviation of the measurements.

The results of the calculation showed that the number of measurements was sufficient. The optimal number of measurements was ten, which was equal to the design value.

It was found that mass m was an increasing function. The minimal value of the mass was reached at thickness t = 4.5 mm. However, at this depth the stresses exceeded the allowable values. Therefore, there is a need to determine the minimal value t at which restriction (1) is fulfilled.

Let us build the diagram of dependencies of the stresses σ on the thickness of the side t according to the values from Table 1 (Fig. 4).

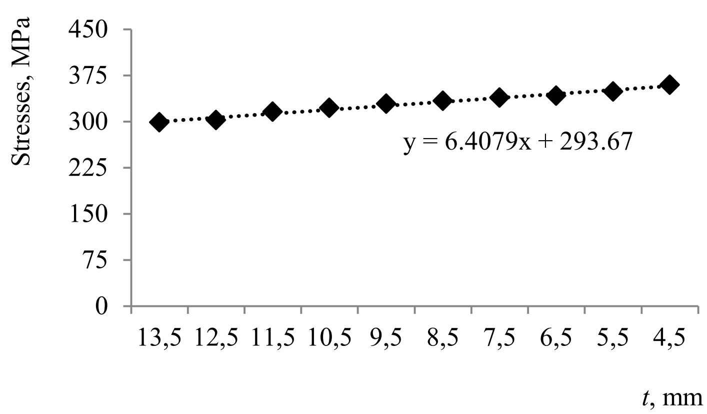


FIGURE 4. Dependency of the stresses in the center sill on the thickness of the sheets.

As function $\sigma(t)$ is close to a piece linear function, it can be approximated to the linear function at each interval. The allowable value of the stresses is reached at the interval $t_8 = 6.5\text{ mm} \le t \le 5.5\text{ mm} = t_9$.

At this interval function $\sigma(t)$ is approximated as linear:

$$\sigma(t) = \sigma_8 + \frac{\sigma_9 - \sigma_8}{t_9 - t_8}(t - t_8), \tag{4}$$

and the value of the stresses with a step of 0.1 mm is calculated.

According to the values given in Table 1, we can obtain

$$m(t) = m_1 + \frac{m_{10} - m_1}{t_{10} - t_1}(t - t_1). \tag{5}$$

In view of this calculation, it has been established that the stresses do not exceed the allowable values at $t = t_0 \geq$ 5.9 mm. And the value of the mass is m_0 = 1594.2 kg, which is 1.7% less than that of the standard structure.

The next stage of the research included the strength calculation of the bearing structure of a passenger car at Design modes I and III. The values of the longitudinal forces included are given in Table 2.

TABLE 2. The values of the longitudinal forces to the passenger car frame in operation.

Longitudinal force, MN			
Design mode			
I		III	
Quasi-static force	Impact, jerk	Quasi-static force	Impact, jerk
–2.5	–2.5	–1.0	–1.0
+1.5	+2.0	+1.0	+1.0

The results of the strength calculation for the frame at Design mode I (impact) are given in Fig. 5. The maximum equivalent stresses are concentrated in the area between the rear draft lug and the bolster beam (Fig. 6).

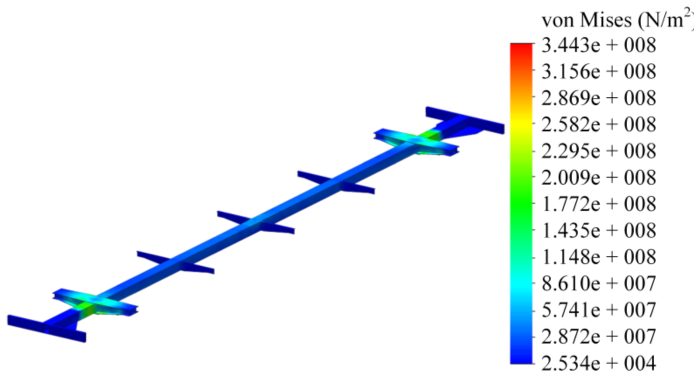


FIGURE 5. Stress state of the frame.

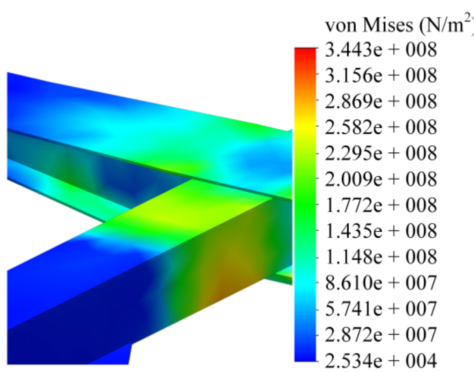


FIGURE 6. Maximum stress concentration zone in the frame.

The maximum equivalent stresses obtained in the calculation for the frame at other modes are given in Fig. 7 in accordance with Table 2.

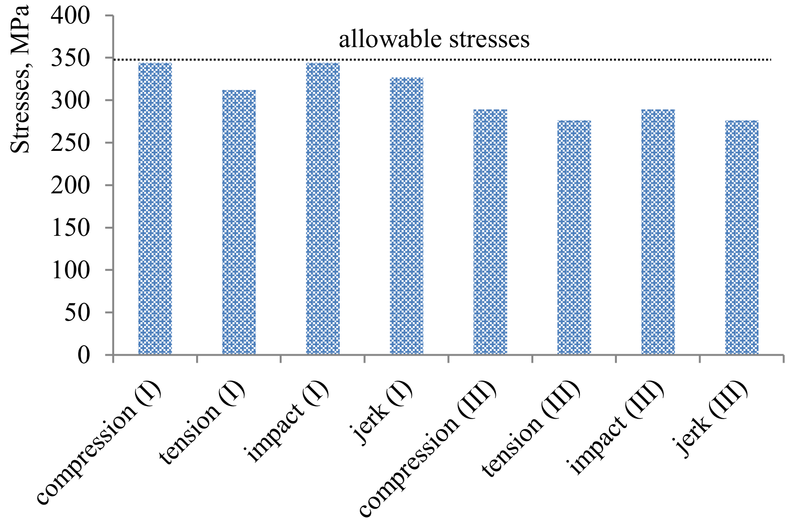


FIGURE 7. Results of the strength calculation for the frame.

Thus, the maximum equivalent stresses occur during Design mode I and correspond to "impact" or "compression". However, these stresses do not exceed the allowable values, which means that the strength of the car frame is provided at the loading modes studied [24-26].

The movement of the car was estimated through the accelerations to its frame in the vertical plane with the options of SolidWorks Simulation. The design diagram built for this estimation is given in Fig. 8.

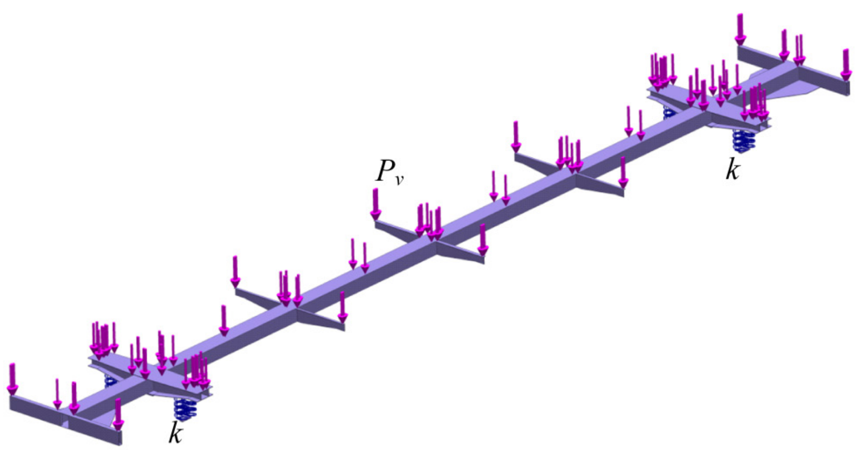


FIGURE 8. Design diagram of the car frame.

The design model of the frame included the vertical loading P_V. The rigid connections with the stiffness k were mounted in the areas of support of the frame on the bogies. The calculation included that the frame rested on a bogie of the model KVZ-TsNII (Type I). As this mode has a two-level spring suspension, the total rigidity of the connection included in the modelling is determined as follows [27, 28]

$$k = \frac{k_a \cdot k_s}{k_a + k_s}, \tag{6}$$

where k_a – the rigidity of the axle-box suspension; k_s – the rigidity of the swing suspension.

The results of the calculation are given in Fig. 9.

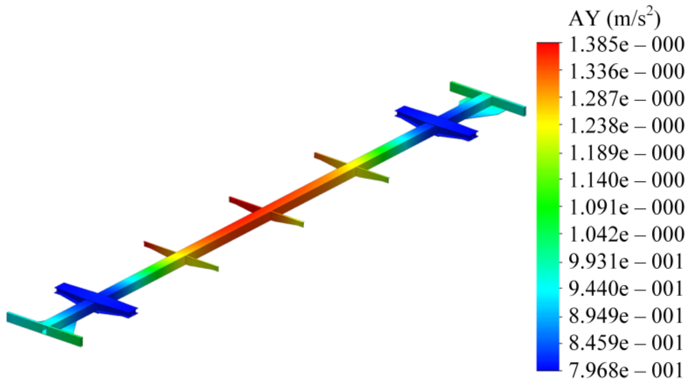


FIGURE 9. Accelerations to the frame.

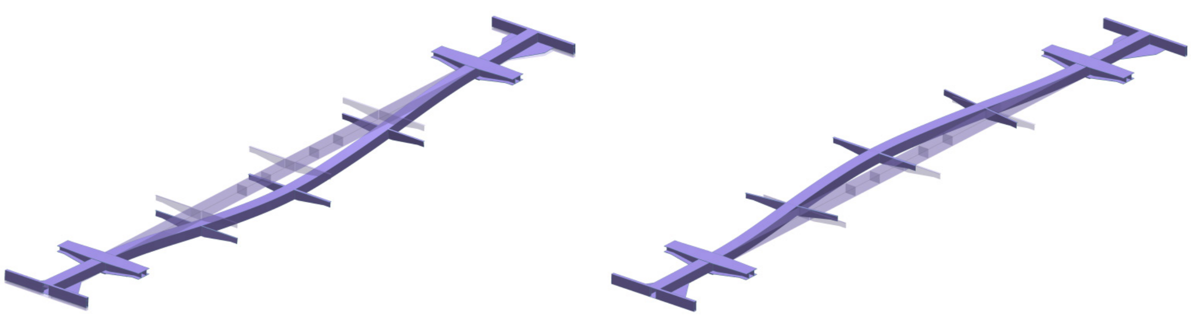
Thus, the maximum accelerations occurred in the middle part of the frame; they were about 1.4 m/s^2 (0.14g). Thus, in accordance with the standards the movement of the car can be estimated as excellent [24-26].

The research also included the modal analysis of the passenger car frame. The calculation was made in SolidWorks Simulation according to the design diagram given in Fig. 8. The safety of the movement of the car was estimated by the first proper oscillation frequency (Table 3). Since its value exceeded 8 Hz, we can conclude that the safety of movement was provided [24-26].

TABLE 3. The values of proper oscillation frequencies for the passenger car frame.

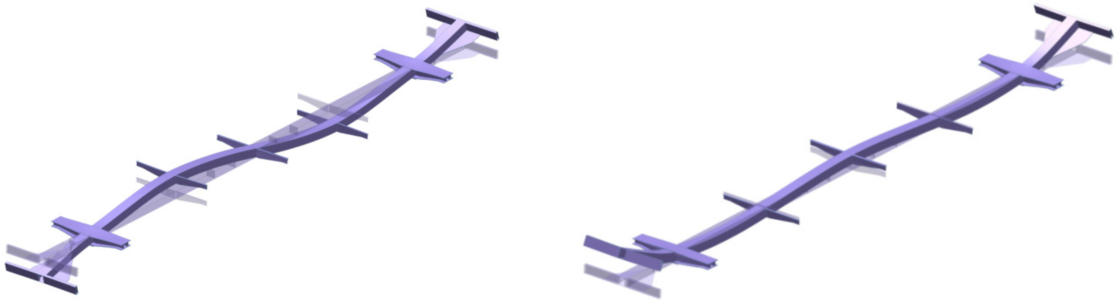
Mode	Frequency, Hz
1	11.9
2	13.9
3	30.5
4	34.7
5	39.8
6	40.1
7	44.3
8	44.8
9	63.1
10	70.9

Some oscillation modes of the frame are given in Fig. 10 and Fig. 11. The deformed model (dull colour) with consideration of the deformation scale 20:1 is placed on the solid model (transparent colour).



(a) (b)

FIGURE 10. Forms of oscillations of the frame: (a) Mode I; (b) Mode II.



(a) (b)

FIGURE 11. Forms of oscillations of the frame: (a) Mode III; (b) Mode IV.

Further research is going to determine the possibility to use elastic frictional connections or filler with the energy-absorbing properties in the frame, which will decrease the dynamic loading on the bearing structure of a car under operational loads and, consequently, increase its operational resource.

It should be noted that the solutions proposed in this research could also be used for passenger cars to be operated in the 1435 track gauge and with the center sill equaling the frame length.

Further research will also determine the loading on the bearing structure of a passenger car through the buffers.

The results of the research can be used by those who deal with designing the bearing structure of a passenger car with better technical and operational characteristics, which will improve the operational efficiency.

CONCLUSIONS

1. This research deals with optimization of the center sill of the passenger car frame through the criterion of minimal material capacity. It can be done by means of replacing the U-profiles with box-section profiles formed with sheets. It has been found that the mass of the frame with consideration of the solutions proposed is by 1.7% lower than that in the standard structure.

2. The research included the strength calculation of the passenger car frame. The maximum equivalent stresses occurred during Design mode I (impact) and were concentrated in the area between the rear draft lug of the coupler and the bolster beam; they were 344 MPa, which is lower than the allowable values.

3. The accelerations to the passenger car frame have been found and the modal analysis of the frame has also been made. The maximum accelerations occurred in the middle part of the frame; they were equal to about 1.4 m/s^2 (0.14g). Thus, in accordance with the standards the movement of the car can be considered as excellent.

The results of the modal analysis have showed that the value of the first proper oscillation frequency of the frame exceeds 8 Hz, thus it can be concluded that safe operation is ensured.

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