DOI: 10.32703/2617-9040-2025-45-12

UDC 629.4.023.14

Igor Martynov[1*], Yuri Kalabukhin[2], Alyona Trufanova[3], Stanislav Martynov[4]

[1]Department of wagons engineering and product quality, Ukrainian State University of Railway Transport, Feuerbach Square, 7, Kharkov, 61050, Ukraine. ORCID: https://orcid.org/0000-0002-0481-3514.
[2]Department of marketing, business activity and economic theory", Ukrainian State University of Railway Transport, Feuerbach Square, 7, Kharkov, 61050, Ukraine. ORCID: https://orcid.org/0000-0003-3693-7607.
[3]Department of wagons engineering and product quality, Ukrainian State University of Railway Transport, Feuerbach Square, 7, Kharkov, 61050, Ukraine. ORCID: https://orcid.org/0000-0003-1702-1054.
[4]Department of operation and repair of rolling stock, Ukrainian State University of Railway Transport, Feuerbach Square, 7, Kharkov, 61050, Ukraine. ORCID: https://orcid.org/0000-0003-1826-6053.

[*]**Corresponding author:** martinov.hiit@gmail.com.

Study of stress-strain state of passenger car body

The article presents the stress-strain state analysis results of the load-bearing elements of the body and frame of the 61-779 model passenger car. To assess the strength of the car body, the finite element method was used with the ANSYS software package. In the model, the body is represented as a system consisting of beam, shell, and solid finite elements. The connections are modeled using rigid links. The model contains a total of 890,436 nodes and 321,874 finite elements. Boundary conditions include restrictions of freedom in support nodes (fixed support) and applied external loads. The main load-bearing element of the car structure is the center sill, made of I-beam profile No. 30. Additional fastening elements are used to increase the stiffness of the connections between the center sill and the crossbeams. The sheathing is made of structural and stainless sheet steel. Corrugated metal 2 mm thick is used as the outer sheathing. A study was conducted on the stress-strain state of the car body with nominal dimensions. The highest stress under load occurs in the bolster beam at the point of contact with the end beam of the frame and amounts to 258 MPa. The stress in the body sheathing between window openings is 65 MPa. The results obtained will further determine the direction of research on the optimization of load-bearing structures of the frame and body.

Key words: *railway transport, passenger car, body, reliability, load, wear, stresses, optimization.*

Introduction. Railway transport remains the leading link in the country's transport system. Despite the decline in passenger traffic for well-known reasons, most passenger transportation is still carried out by railways. Therefore, passenger cars must ensure traffic safety and comfortable conditions for passengers.

Unfortunately, the current state of the passenger industry at the moment does not meet the requirements for the effective implementation of Ukraine's European integration course. The fleet of passenger cars owned by the Passenger Company branch of JSC Ukrzaliznytsia has largely worked out its standard service life and exhausted its resource. The average age of a passenger car exceeds thirty years. The cars are outdated both morally and physically. The major repairs carried out to extend the service life can no longer provide the required reliability and modern level of comfort.

Ukraine needs new innovative passenger cars. But traditional approaches to car design are based on assessing the safety margin. However, the loads acting in operation are of a probabilistic nature. As a result, a deterministic approach to designing primarily bodies turns out to be insufficiently effective in terms of ensuring both the reliability of a passenger car and its efficiency.

Analysis of recent research and problem statement. A number of studies have been devoted to the issues of ensuring the strength and reliability of passenger car bodies both in our country and abroad. Thus, articles [1, 2] present an overview of the technical condition of the passenger car fleet owned by the branch of JSC "Passenger Company" Ukrzaliznytsia. The authors come to the conclusion that a radical renewal of the passenger car fleet is necessary. Moreover, the cars that are designed must meet all the requirements of the European Union regulatory documents [3].

The results of statistical analysis of the amounts of wear and damage to the metal structures of the frame and body of passenger cars from different years of manufacture are presented in the works [4, 5]. Dependencies characterizing the amount of corrosion wear on the operating time are obtained. It is proven that the lower parts of the side and end walls are most often subject to corrosion damage. The results of strength analysis of the passenger car metal structures, taking into account their wear, are presented.

Ways to reduce the tare weight of a passenger car are considered in the articles [6, 7]. The authors performed multi-variant calculations of the strength of a body with different skin thicknesses. It was established that the body of a passenger car has a sufficient margin of safety and, due to the rational use of elements of the car's metal structures, it is possible to achieve a reduction in the tare weight.

The article [88] describes calculation of the main standard dynamic characteristics for a new passenger car. All standardized dynamic indicators of the passenger car model 61-779 do not exceed the permissible values, and the stability indicators have a sufficient reserve, which indicates good running qualities of the car.

In article [9], the authors propose a method for theoretical study of the strength of double-decker car bodies, the selection of rational design schemes and parameters.

Study [10] discusses a method for assessing the durability of a passenger car body under the influence of random dynamic loads. The main focus is on the use of fatigue analysis using the finite element method. Taking into account dynamic stress and a nonlinear damage accumulation model, the service life and level of fatigue damage to the body are determined. The simulation results were experimentally confirmed.

The authors of the study [11] propose a method for testing the fatigue strength of a passenger car body frame. The tests are carried out under laboratory conditions with accurate simulation of loads corresponding to real operating conditions. The authors examined in detail the process of simulation on a vibration test rig during motion.

Study [12] presents a methodological approach to assessing the reliability of passenger car structures using the finite element method. The authors analyze the impact of various loads on the durability of the car body and propose approaches to optimizing structures to improve their reliability.

The paper [13] considers issues of determining the durability of a passenger car body under dynamic loads. The study is based on the finite element method and allows for high-precision modeling of the real dynamic properties of the body structure.

The article [14] presents the results of the analysis of the metal structures of passenger cars taking into account their operation. It has been proven that the lower parts of the side and end walls are most often subject to corrosion damage. The authors of the article [15] propose a new approach to dynamic optimization of the design of passenger car bodies under static loads.

The study [16] presents a methodology for predicting the safety level of passenger cars using well-known software products. The works [17, 18] assess the accident rate and provide proposals for the modernization of a railway passenger car. Using the finite element method, the authors simulated a collision of a passenger car with a rigid wall. As a result, a car design with better accident properties was obtained.

The purpose and tasks of the study. The purpose of this work is to conduct research for optimization calculations of the structures of bodies of new generation passenger cars. To achieve this, it is necessary to build a computational 3D model of the body and frame of the 61-779 railcar and, using the finite element method, analyze the stress-strain state of the load-bearing elements of the body and frame, taking into account possible wear during operation.

Materials and methods of research. The Finite Element Method (FEM) is currently one of the most essential computational techniques for solving problems in solid mechanics. It operates by discretizing a continuous object into a finite number of smaller regions, known as finite elements, to facilitate the numerical solution of continuum mechanics equations. These equations are assumed to hold true within each individual element.

Finite elements may represent physical segments of the structure or be abstract mathematical representations – such as elements used to model rods, beams, plates, or shells. Each element is assigned specific physical properties (e.g., stiffness, strength, material density) and is used to describe key field variables relevant to solid mechanics, such as displacements, strains, and stresses.

These field variables are typically defined at the nodes of the element. Interpolation functions are then used to estimate their values at any location within the element or along its boundaries. The core of the FEM formulation involves establishing mathematical relationships between the nodal values – primarily displacements and the corresponding forces in the context of continuum mechanics. A rigid compartment passenger car was selected for the calculation. The frame design is based on a spine beam configuration with additional reinforcement along the entire perimeter of the frame. The main load-bearing element of the structure is the spine beam, made from I-beam profile No. 30. Cross beams made of channel sections are welded to it. To increase the rigidity of the joints between the spine and cross beams, additional fastening elements in the form of triangles are used.

Longitudinal beams present in the structure are intended for installation of undercarriage equipment. The end part of the frame consists of two channel sections, and the lower frame binding is also made from channel section No. 20. The load-bearing elements efficiently combine traditional structural and low-alloy steels commonly used in railcar manufacturing. The sheathing consists of sheet-rolled structural and stainless steels. Corrugated metal with a thickness of 2 mm is used as the exterior cladding. The roof frame consists of 26 arch-shaped cross beams with a Z-shaped profile. For reinforcement, each of them is additionally welded to an unequal-angle bar along the base of the arch and two reinforcement elements. The car floor is made of two corrugated trapezoidal sheets.

The calculation scheme is shown in Fig. 1

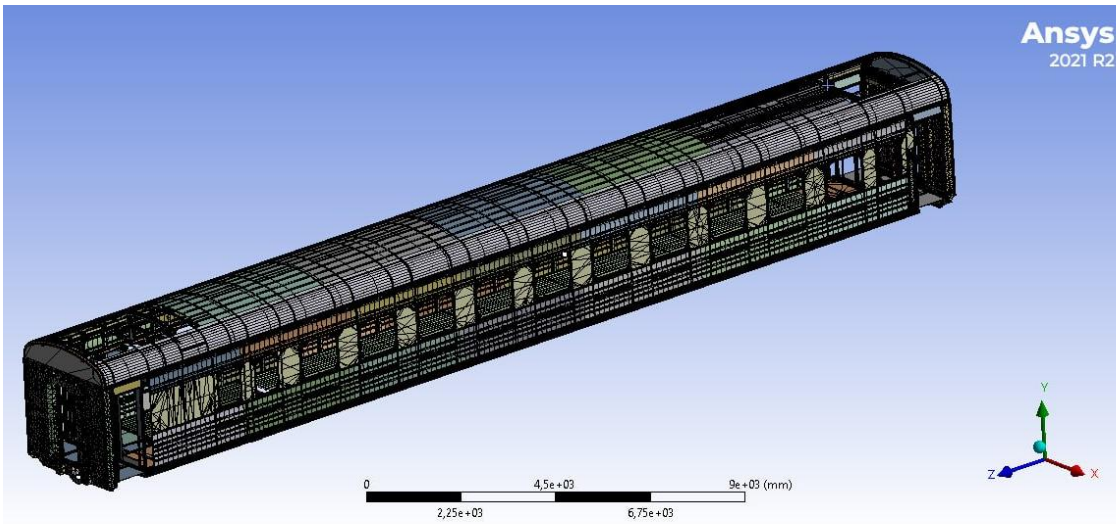

Fig. 1. **Passenger car 61-779 body model**

The load-bearing elements rationally combine traditional structural and low-alloy steels used in railcar construction. Sheathing is a rolled sheet made of structural and stainless steel.

To assess the strength of the body, the method of finite elements was applied using the ANSYS software complex in accordance with the requirements of the DSTU [19]. In the model, the body is presented as a system consisting of beam, shell and solid finite elements.

The calculation scheme of the body (without roof) is shown in Fig. 2 (view from below).

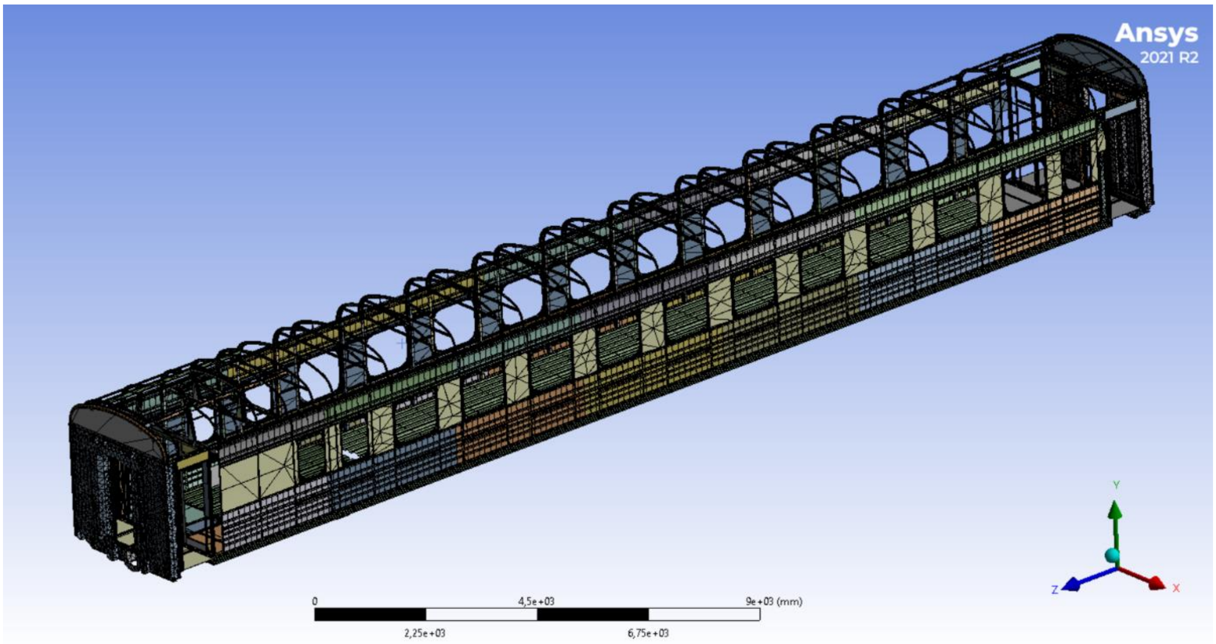

Fig. 2. **Passenger car 61-779 body model** (without roof)

The scheme of the body fastening in the thrust bearing unit is shown in Fig. 3 (view from below).

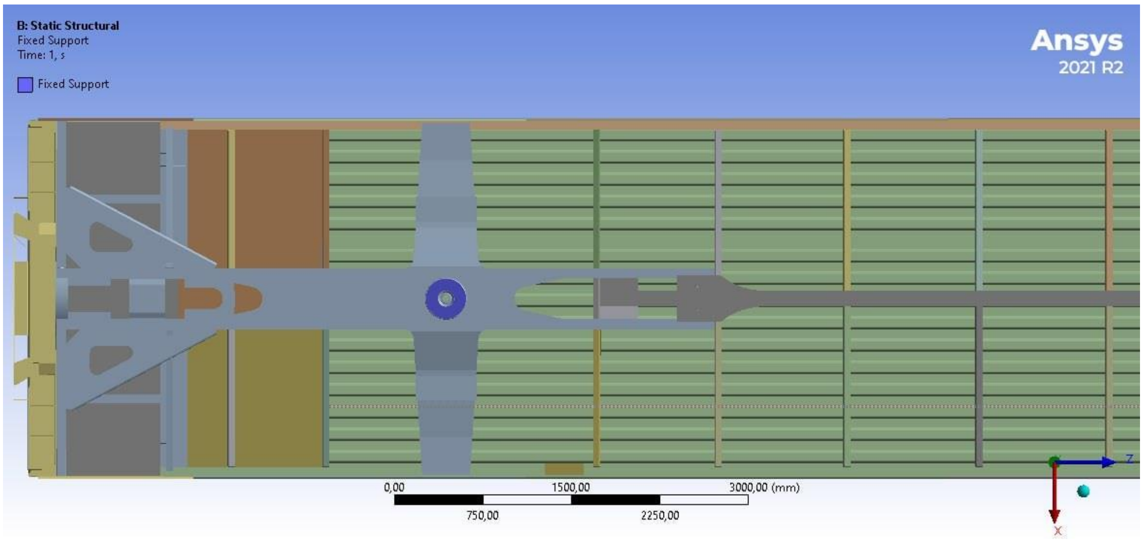

Fig. 3. **Passenger car 61-779 body model (view from below)**

Beam elements include risers, lower sidewall trim, roof arches, etc. Elements such as the upper trim, wall and roof trim, and floor were modeled using shell elements. Solid elements were used to describe structures of complex geometry such as (under the handrails) or end walls. Beam elements perceive all types of loads (tension, bending in two planes and torsion). Each node of such an element has three linear and three angular degrees of freedom. Beam elements - each node of such elements is endowed with six degrees of freedom: three linear displacements along the axes of the local reference system and three angles of rotation around these axes. Since the nodes of solid finite elements have only three degrees of freedom, all applied external moments are reduced to zero. Therefore, each finite element transfers only forces and linear displacements to neighboring elements.

The modeling of connections is implemented through a rigid connection. In total, the model contains

890,436 nodes and 321,874 finite elements.

The boundary conditions include restrictions of freedom in the supporting nodes (fixed support) and applied external loads.

According to the requirements of [19] for passenger cars, external loads include: the self-gravity of the structure (container), in particular objects and equipment, the gravity of passengers and personnel and luggage; inertial forces caused by oscillatory accelerations of masses during the movement of the wagon on a track with irregularities; forces arising during the movement of the wagon as a result of the interaction of the wagon and the track in curved sections of the track and switches, quasi-static wind pressure forces; static and dynamic forces of interaction between cars, between a car and a locomotive, traction and braking forces arising during transient modes, longitudinal forces of inertia and others.

Regarding longitudinal forces, the following load modes are established: I mode (conditional safety mode) and III mode – operational mode. Each of these modes includes appropriate load combinations that allow you to simulate different operating options.

The vertical loads acting on the body of a passenger car include the body's own weight, the mass of installed internal equipment, the stock of operating materials, and the mass of passengers with their luggage. These loads are supplemented by dynamic components that arise during the movement of the car due to vertical accelerations caused by track irregularities, vibrations, and other dynamic effects.

The vertical dynamic load is determined using the vertical dynamic coefficient. The coefficient of vertical dynamics k_{dv} is considered in [19] as a random function with a probability distribution of the form

$$P(k_{dv}) = 1 - exp\left(-\frac{\pi}{4} \cdot \frac{k_{d.v.}^2}{\overline{k}_{d.v.}^2} \cdot \beta^2\right). \quad (1)$$

The coefficient k_{dv} is defined as the quantile of this function with the calculated one-sided probability $P(k_{dv})$ according to the formula

$$k_{dv} = \overline{k}_{dv}\sqrt{\frac{4}{\pi} ln\frac{1}{1 - P(k_{dv})}}, \quad (2)$$

where $\overline{k}_{dv}$ is the average value of the vertical dynamic coefficient (mathematical expectation of the random process of changing the vertical dynamics coefficient $k_{dv}(t)$;
$P(k_{dv})$ – confidence probability. It is equal to $P(k_{dv})$=0.97.

The average value of the vertical dynamic coefficient $\overline{k}_{dv}$ визначається за наступною формулою

$$\overline{k}_{dv} = a + 3{,}6 \cdot 10^{-4} \cdot b\frac{V - 15}{f_{ст}}, \quad (3)$$

where a is an empirical coefficient, which for car bodies is equal to a=0.05;
b – coefficient depending on the number of axles in the cart (for biaxial carts b is equal to 1);
f_{st} – static deflection of spring suspension;

V – speed of movement in m/s.

$$f_{\text{ст}} = \frac{Q_{b.p.} + 2(Q_b + 0.333Q_1)}{2C_{b1}} + \frac{Q_{b.p.} + 2\left(Q_k + Q_1 + Q_{b.f.} + 0.333Q_2\right)}{2C_{b2}}, \tag{4}$$

where $Q_{b.p.}$ - body weight with passengers ($Q_{b.p.} = 518\ kN$);
Q_b - bolster weight ($Q_b = 6\ kN$);
Q_1 - weight of central spring suspension springs of one bogie ($Q_1 = 0.307\ kN$);
Q_2 - weight of axle box springs suspension of one bogie ($Q_2 = 0.312\ kN$);
C_{b1} - vertical rigidity of the central spring suspension of one bogie ($C_{b1} = 2659.22\ kN$);
C_{b2} - vertical rigidity of axle box spring suspension of one bogie ($C_{b2} = 6567.064\ kN$);
Q_k - empty body weight ($Q_k = 476\ kN$);
$Q_{b.f.}$ - weight of the bogie frame ($Q_{b.f.} = 14.68\ kN$);

After entering the corresponding values into formula (4), for a fully loaded passenger carwe obtain $f_{\text{ст}}$ 0.13 m.

During the calculations, it was assumed that the car moves at a maximum speed of 160 km/h and the third calculation mode is used.

Thus, for movement at a speed of 44.4 m/s, the average value of the coefficient of vertical dynamics $\overline{k}_{dv}$ was

$$\overline{k}_{dv} = 0{,}05 + 3{,}6 \cdot 10^{-4} \cdot 1 \frac{44{,}4 - 15}{0{,}13} = 0{,}124.$$

$$k_{\text{дв}} = 0{,}124 \sqrt{\frac{4}{3{,}14} ln \frac{1}{1 - 0{,}97}} = 0{,}263.$$

The gravitational force Q acting on the body is defined as the difference between the total (gross) mass of the car and the total mass of its bogies. During the calculations, the weight of the metal structure of the body itself and the massive units that make up the main equipment was initially taken into account.

The residual part of the force Q – that is, the difference between the total gravitational force and the already taken into account masses of structures and equipment — was evenly distributed over the floor area in the form of a surface load. This approach allows you to correctly model the influence of passengers and small equipment on the overall load distribution.

The action of the lateral load should be taken into account only when calculating according to mode III. The force, which is equal to the difference of the centrifugal force and the horizontal component of the gravity force, which arises as a result of the elevation of the outer rail, for passenger cars is 10% of the gross force of gravity. Then for a body with passengers

$$F_b = 0.1Q_{b.p.}. \tag{5}$$

$$F_b = 0.1 \cdot 518 = 51.8\ kN.$$

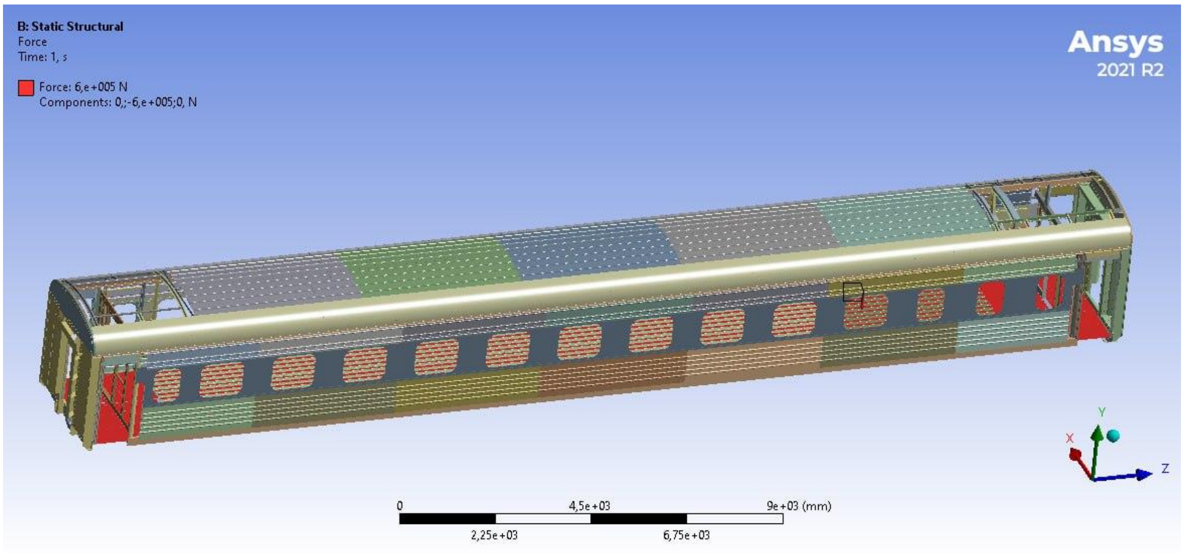

Fig. 4. **Scheme of application of a vertical load on the body of a passenger car**

Also taken into account is the force of wind pressure, which divides the area of the side projection of the body by the specific wind pressure (500 N/m^2).

For the body of a passenger car

$$F_b = p_w \cdot F, \tag{6}$$

where p_w - specific wind pressure ($p_w = 0.5\ kN/m^2$);
F - lateral projection area of the car body ($F = 98\ m^2$).

Then for the body of a passenger car force of wind pressure will be equal

$$F_b = 0.5 \cdot 98 = 49\ kN$$

Thus, the total lateral load will be 100.8 kN. It is applied to the upper and lower lining of the side walls.

Two load application options were considered during the calculations. The first corresponds to Calculation Mode I, while the second corresponds to Calculation Mode III at a speed of 160 km/h.

The distribution of stresses over the car body is shown in Fig. 5.

The highest stresses under loading occur in the pivot beam at the point of support on the end beam of the frame and amount to 258 MPa. Stresses in the roof elements do not exceed 57 MPa. In the side walls, the pillars are the most heavily loaded element. In them, the maximum stress does not exceed 69 MPa. Stresses in the body sheathing in the openings between the windows amount to 65 MPa. The maximum stress in the body occurs in the end walls – 160 MPa.

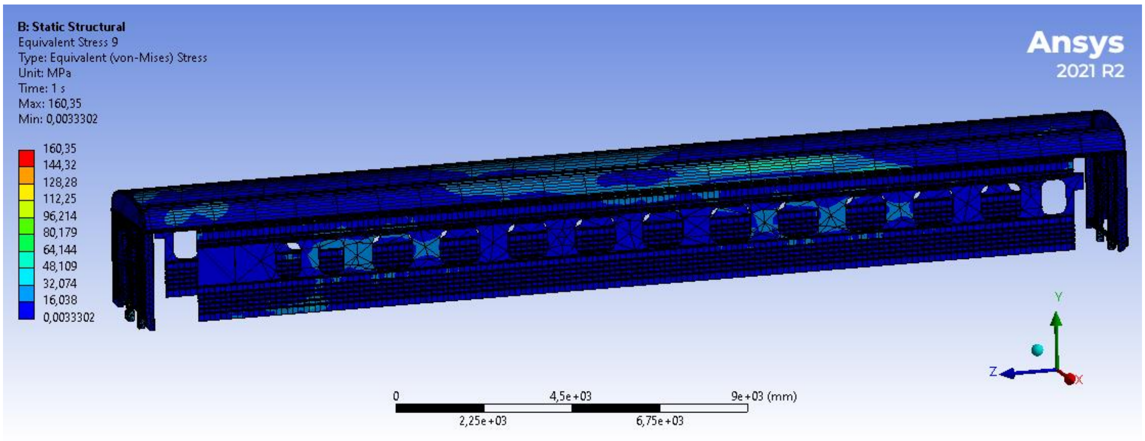

Fig. 5. **Stress distribution graph for the car body**

The distribution of stresses over the pivot beam and the bolster assembly is shown in Fig. 6.

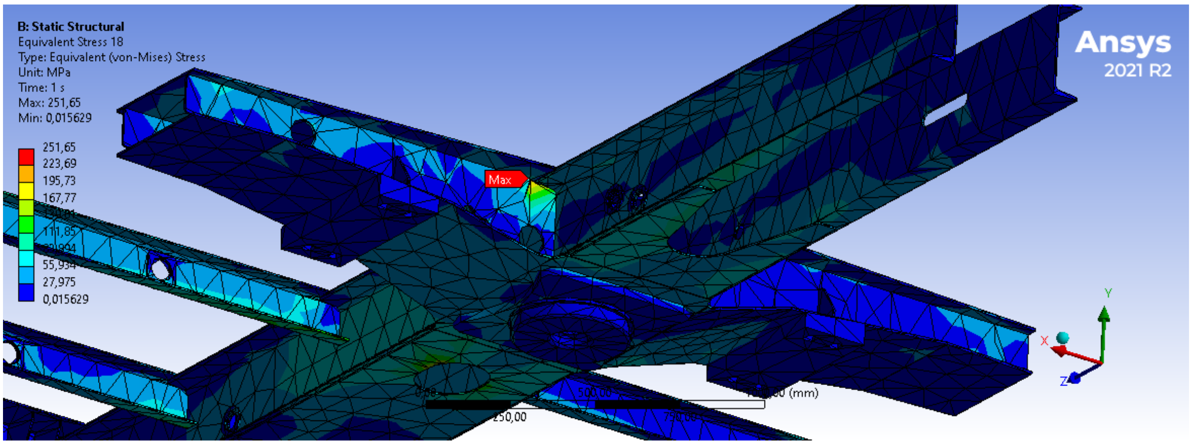

Fig. 6. **Diagram of stress distribution in the frame pivot assembly passenger car 61-779**

Stresses occurring in the center sill do not exceed 98 MPa, and they are localized at the junction with the end part of the frame. The average value of stresses arising in the center sill is 9.6 MPa. The most heavily loaded is the pivot beam. The stresses in the cross beams do not exceed 90 MPa. Except for the first row of cross beams supported by the end frame. In it, the maximum stresses reach 200 MPa in the contact zone with the center sill.

The developed model was verified. In the initial stage, calculations were performed for the structure with standard sheathing thicknesses. The results were then compared with experimental data from strength tests. The close agreement between the results confirmed the accuracy of the model.

For comparison, on Fig. 7 is shown stress distribution in the frame pivot assembly passenger car 47D [**Ошибка! Источник ссылки не найден.**]. The maximum stresses in the kingpin beam are localized near the bolster assembly and do not exceed 191 MPa.

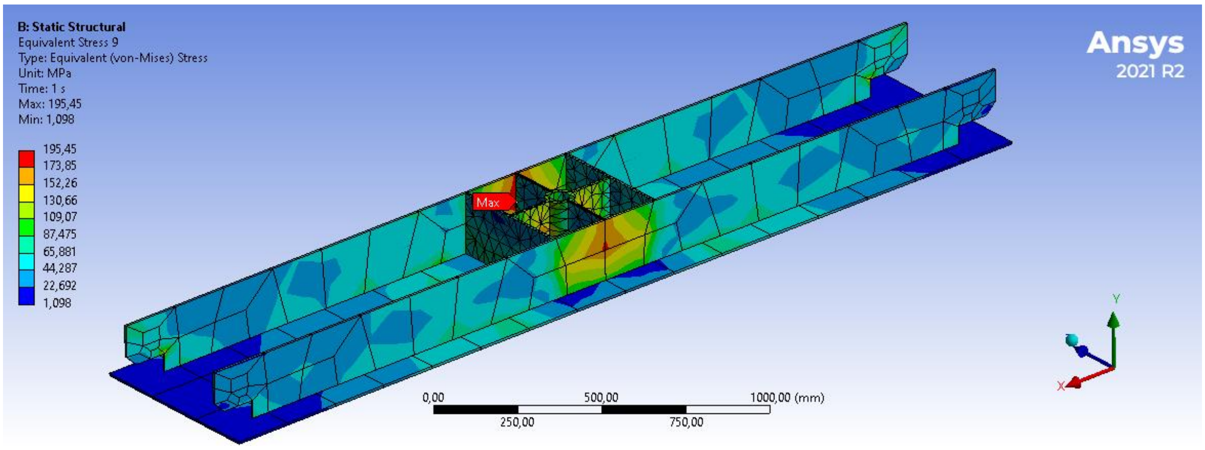

Fig. 7. **Diagram of stress distribution in the frame pivot assembly passenger car 47D**

The distribution of stresses in the center sill is shown in Fig. 8.

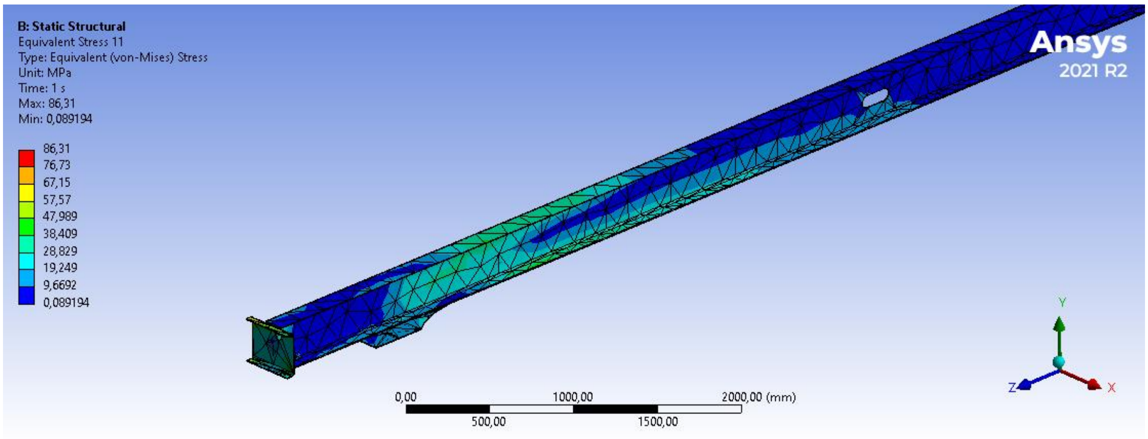

Fig.8. **Diagram of stress distribution in the center sill passenger car 61-779**

In the center sill of the 61-779 model railcar, the maximum stresses are also localized in the area of the pivot beam node and amount to 98 MPa.

The results obtained made it possible to compare the stress-strain state of the 61-779 model car with the 47D model car (Table 1).

Table 1. **Comparison of stresses in passenger car units**

	47D	61-779
Pivot beam	195,4	251,6
Longitudinal beam	258,7	-
Center sill	-	97,8
Cross beam	74,8	89,6
Body sheathing	125,1	65
End wall	143,3	160,3
Upper binding	46,3	57

Due to the fundamental differences in frame design, different elements are subjected to the highest loads. In model 47D, it is the longitudinal beam with 258.7 MPa, whereas in the 61-779 model, the most heavily loaded element is the pivot beam. In both designs, the maximum stresses occur near the bolster assembly. The overall stress values in the car body are similar, differing only in the side wall sheathing. The obtained stress values do not exceed the allowable limits.

Conclusion.

1. A finite element model was constructed based on the 3D model of the body of the rigid compartment car 61-779. The strength calculations of the body were performed using the finite element method. Beam, shell and solid finite elements were used to model the body elements.
2. A study was conducted of the stress-strain state of the body at nominal dimensions with standard skin thicknesses. The highest stresses under loading occur in the pivot beam at the point of support on the end beam of the frame and amount to 258 MPa. The stresses in the body sheathing in the openings between the windows are 65 MPa.
3. The results obtained will further determine the direction of research on the optimization of the supporting structures of the frame and body.

REFERENCES

1. Bozhok, N. O., Bulgakova, Y. V., & Pularia, A. L. (2014). Research on the current state of the passenger car fleet. *Review of transport economics and management*, (8), 78-87. [in Ukrainian].
2. Loboyko, L. N., & Barash, Yu. WITH. (2007). State of the wagon park and wagon repair base in Ukraine. Science and progress of transport. *Bulletin of the Dnipropetrovsk National University of Railway Transport*, (19), 176-182. [in Ukrainian].
3. Kirpa, G. M. (2004). *Integration of Ukrainian railway transport into the European transport system*. D.: Publishing house of Dnipropetrovsk National University of Railway Transport named after Acad. V. Lazaryan. [in Ukrainian].
4. Martynov, I., Trufanova, A., Petukhov, V., & Serhiienko, M. (2021). Research of the dependence of operation of carrying elements of passenger cars. *Transport Systems and Technologies*, (36), 72–81. [in Ukrainian]. https://doi.org/10.32703/2617-9040-2020-36-8.
5. Martynov, I. E., Trufanova, A. V., Pavlenko, Yu. S., & Sergienko, M. O. (2018). Analysis of the technical condition of passenger car bodies. *Bulletin of the National Technical University of KhPI. Series: New solutions in modern technologies*, (45), 41-46. https://doi.org/10.20998/2413-4295.2018.45.06. [in Ukrainian].
6. Myamlin, S. V., Yagoda, P. A., Dedaeva, T. A., & Shkabrov, O. A. (2006). Reducing the weight of metal structures of passenger cars for high-speed transportation. *Bulletin of the Dnipropetrovsk National University of Health Transport named after Academician V. Lazaryan*, (13), 118-120. [in Ukrainian].
7. Prikhodko, V. I., Shkabrov, O. A., Myamlin, S. V., & Yagoda, P. A. (2007). Improving the design of passenger car bodies for high-speed transportation. *Bulletin of the Dnipropetrovsk National University of Health Transport named after Academician V. Lazaryan*, (14), 152-156. [in Ukrainian].
8. Prikhodko, V. I. (2006). Calculation of dynamic indicators of a passenger car. Science and progress of transport. *Bulletin of the Dnipropetrovsk National University of Health Transport named after Academician V. Lazaryan*, (12), 146-152. [in Ukrainian].
9. Sun, W., Zhou, J., Gong, D., & You, T. (2016). Analysis of modal frequency optimization of railway vehicle car body. *Advances in Mechanical Engineering*, *8*(4), 1687814016643640. https://doi.org/10.1177/1687814016643640.
10. Sharma, S. K., Sharma, R. C., & Lee, J. (2022). In situ and experimental analysis of longitudinal load on carbody fatigue life using nonlinear damage accumulation. *International Journal of Damage Mechanics*, *31*(4), 605-622. https://doi.org/10.1177/10567895211046043/.
11. Song, Y., Wu, P., & Jia, L. (2016). Study of the fatigue testing of a car body underframe for a high-speed train. *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, *230*(6), 1614-1625. https://doi.org/10.1177/0954409715618425.
12. de Cisneros Fonfría, J. J. J., Olmeda, E., Sanz, S., Garrosa, M., & Díaz, V. (2024). Failure analysis of a train coach underframe. *Engineering Failure Analysis*, *156*, 107756. https://doi.org/10.1016/j.engfailanal.2023.107756.
13. Bingrong Miao, & Dingchang Jin. (2009). Evaluation of Railway Vehicle Car Body Fatigue Life and Durability using a Multi-disciplinary Analysis Method. *International Journal of Vehicle Structures and Systems,* 1(4). https://doi.org/10.4273/ijvss.1.4.05.
14. Martynov, I., Kalabukhin, Y., Trufanova, A., & Martynov, S. (2024). Analysis of stress state of passenger car bodies. *Transport systems and technologies*, (43), 111-120. [in Ukrainian]. https://doi.org/10.32703/2617-9059-2024-43-9.

15. Cascino, A., Meli, E., & Rindi, A. (2023). Dynamic size optimization approach to support railway carbody lightweight design process. *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, *237*(7), 871-881. https://doi.org/10.1177/09544097221140933.

16. Kobishanov, V.V., Lozbinev, V.P., Sakalo, V.I., Antipin, D.Y., Shorohov, S.G., & Vysocky, A.M. (2013). Passenger Car Safety Prediction. *World Applied Sciences Journal*, *24*, 208–212. https://www.idosi.org/wasj/wasj24(1)/ 2013.htm.

17. Baykasoglu, C., Sunbuloglu, E., Bozdag, S. E., Aruk, F., Toprak, T., & Mugan, A. (2012, April). Numerical static and dynamic stress analysis on railway passenger and freight car models. In *International Iron & Steel Symposium* (pp. 02-04)..

18. Baykasoğlu, C., Sünbüloğlu, E., Bozdağ, S. E., Aruk, F., Toprak, T., & Mugan, A. (2011). Railroad passenger car collision analysis and modifications for improved crashworthiness. *International Journal of Crashworthiness*, *16*(3), 319-329.https://doi.org/10.1080/13588265.2011.566475.

19. DSTU 7774:2015. (2017). *Mainline passenger locomotive-drawn wagons. General technical standards for the calculation and design of mechanical parts of wagons*. Ministry of Economic Development of Ukraine. [in Ukrainian].

20. Martynov, I. E., Trufanova, A. V., Shovkun, V. O., Martynov, S. I., & Ostapenko, Ya. V. (2023). Modeling the stress-strain state of a rigid-compartment passenger car body. Collection of scientific papers "Rail rolling stock", (27), 59-69. https://doi.org/10.47675/2304-6309-2023-27-59-69.

Ігор Мартинов[1], Юрій Калабухін[2], Альона Труфанова[3], Станіслав Мартинов[4]

[1]Кафедра інженерії вагонів та якості продукції, Український державний університет залізничного транспорту, пл. Фейєрбаха, 7, Харків, 61050, Україна. ORCID: https://orcid.org/0000-0002-0481-3514.

[2]Кафедра маркетингу, комерційної діяльності та економічної теорії, Український державний університет залізничного транспорту, пл. Фейєрбаха, 7, Харків, 61050, Україна. ORCID: https://orcid.org/0000-0003-3693-7607.

[3]Кафедра інженерії вагонів та якості продукції, Український державний університет залізничного транспорту, пл. Фейєрбаха, 7, Харків, 61050, Україна. ORCID: https://orcid.org/0000-0003-1702-1054.

[4]Кафедра експлуатації та ремонту рухомого складу, Український державний університет залізничного транспорту, пл. Фейєрбаха, 7, Харків, 61050, Україна. ORCID: https://orcid.org/0000-0003-1826-6053.

Дослідження напружено-деформованого стану кузова пасажирського вагона

У статті викладено результати аналізу напружено-деформованого стану несучих елементів кузова та рами пасажирського вагона моделі 61-779. Для оцінки міцності кузова застосовано метод скінчених елементів з використанням програмного комплексу ANSYS. У моделі кузов представлений у вигляді системи, що складається з балкових, оболонкових та об'ємних кінцевих елементів. Всього модель містить 890436 вузлів і 321874 кінцевих елемента. Граничні умови включають обмеження свободи в опорних вузлах (нерухома опора) та зовнішні навантаження. Основним несучим елементом конструкції вагона є хребтова балка, виконана з двотаврового профілю № 30. До неї приварені поперечні балки зі швелерів. Для підвищення жорсткості з'єднань хребтової та поперечних балок використовуються додаткові елементи кріплення. Зовнішня обшивка кузова виконана з листових конструкційних та нержавіючих сталей товщиною 2 мм. Проведено дослідження напружено деформованого стану кузова при номінальних розмірах. Найбільші напруження при навантаженні виникають у шворневій балці у місці обпирання на кінцеву балку рами і становлять 258 МПа. Напруження в обшивці кузова в отворах між вікнами становлять 65 МПа. Отримані результати надалі визначать напрямок досліджень з оптимізації несучих конструкцій рами та кузова.

Ключові слова: *залізничний транспорт, пасажирський вагон, кузов, надійність, спрацювання, напруження, оптимізація.*