UDC 621.3
DOI: 10.15587/1729-4061.2025.321962

DETERMINING THE AMPLITUDE-FREQUENCY CHARACTERISTICS OF AN ELECTRIC TRAIN CAR FLOOR

Oleh Holovashchenko
PhD Student*
Yaroslav Shostak
PhD Student*
Viktor Tkachenko
Corresponding author
Doctor of Technical Sciences, Professor*
E-mail: tkachenko_vp@gsuite.duit.edu.ua
*Department of Electromechanics and Rolling Stock of Railways
State University of Infrastructure and Technologies
Kyrylivska str., 9, Kyiv, Ukraine, 04071

The object of this study is the process of vibration propagation through the floor of a passenger railroad car from the base to passenger seats. The problem addressed relates to the influence of the structure of an electric train car floor on its anti-vibration properties, in particular the role of rubber shock absorbers in the process of damping floor vibrations. A mathematical model of the vibration propagation chain from the base of the car to the top layer of the floor was built. As a source of vibrations in the mathematical model, the kinematic disturbance of the base of the car was used. Two variants of the floor structure were studied – with five layers and with seven layers. The parameters of the floor layers obtained experimentally by the method of free oscillations were used in the calculations.

Based on the application of specific parameters of the floor layers, a transition was performed from distributed mass, elastic, and dissipative parameters to concentrated ones. The solution to the system of differential equations of oscillations was obtained in the form of time functions of displacements and velocities of floor elements. Layer-by-layer amplitude-frequency characteristics (AFR) of the floor were constructed. Zones of resonant frequencies of the oscillating system of the car floor have been determined. The dependence of the influence of the degree of passenger loading on the resonance frequency of floor surface oscillations was established. When the load changes from minimum to maximum, the resonant frequency decreases from 15–22 Hz to 8–12 Hz.

It has been confirmed that the influence of parameters of the layer of rubber shock absorbers on the floor AFR is decisive for its vibration protection. When comparing the two floor variants under consideration, there were no advantages of a seven-layer floor over a five-layer floor. AFR for the first and second layers of plywood flooring are identical.

Based on the data on "dangerous" vibration frequencies for humans, the preferred value for the shock absorber installation diagram was derived – 5–6 units/m^2, which could prove useful under the conditions of modernization of suburban electric train cars

Keywords: railroad transport, passenger cars, car floor vibration, amplitude-frequency characteristics, passenger comfort

Received 12.11.2024
Received in revised form 27.12.2024
Accepted 16.01.2025
Published 26.02.2025

***How to Cite:** Holovashchenko, O., Shostak, Y., Tkachenko, V. (2025). Determining the amplitude-frequency characteristics of an electric train car floor. Eastern-European Journal of Enterprise Technologies, 1 (7 (133)), 64–75. https://doi.org/10.15587/1729-4061.2025.321962*

1. Introduction

The suburban railroad provides a significant part of passenger transportation as part of daily work trips along the "suburban area – metropolis" routes. The trips of workers to and from work are part of the enterprise's production process. The passenger's state of well-being and productivity during the working day depend on the passenger's comfort level during trips on electric trains. Suburban electric trains are characterized by a periodic mode of movement with frequent accelerations and braking in a wide range of speeds. This mode of movement is the cause of vibrations over a wide range of frequencies transmitted to passengers from the seats and floor of the car. A feature of suburban diesel and electric trains is that passengers sit on seats or even stand on the floor throughout the trip. This makes them particularly vulnerable to floor vibrations.

The technical condition of suburban trains on Ukrainian railroads is characterized by significant wear and tear. In particular about 85 % of rolling stock needs major repairs [1]. Ukrzaliznytsia commissioned PrAT "Kyiv Electric Car Repair Plant" ("KEVRZ") to modernize electric trains of the ER-2, ER-9, EPL-2, EPL-9, ED-9M series, etc. "KEVRZ" is the only enterprise in Ukraine that carries out deep modernization of rolling stock of electric trains of all series. Designing electric rolling stock that meets modern European standards of comfort, traffic safety, fire safety, information support is the main task of modernization. Calculations of the economic feasibility of modernizing Ukrzaliznytsia's electric trains that have reached the end of their service life have shown the advantage of such a path in comparison with the deployment of the design and production of new trains. This is especially appropriate under the conditions of Ukraine's martial law. The decision to deeply modernize electric trains made it possible to significantly change the structure of car floor. One of the main tasks of modernization is to improve the characteristics of vibration isolation of passengers of the suburban electric train. Suburban electric trains are characterized by a periodic mode of movement with frequent accelerations and braking in a wide range of speeds. This mode of movement is the cause of vibrations over a wide range of frequencies transmitted to passengers from the seats and floor of the car.

In general, the model of a vibration protection system can be represented in the form of a chain of three links: vibration disturber; vibration isolation link; object of vibration protection. The source of vibrations of a passenger car is, as a rule, the base of the car body, which is affected by disturbances from the running gear and attached equipment. Vibration isolation, in the case of a car, is a multi-component floor that includes elastic and dissipative layers, the function of which is to absorb vibrations, and rigid layers, the function of which is to support the load from passengers, seats, partitions, and interior equipment of the car. The object of the car's vibration protection is the passenger resting on the covering – the upper layer of the floor. The level of vibrations perceived by passengers depends on the mechanism of disturbance oscillation generation, the characteristics of the vibration transmission channel to the passenger, and the quality of the vibration isolation of the passenger seats and the car floor.

Therefore, it is a relevant task to carry out studies on the effect of the structure of the floor of an electric train car on the propagation of vibrations from the base to passengers.

2. Literature review and problem statement

Work [2] reports the results of research into the vibration of the floor of passenger cars. The characteristic frequencies of vibrations of the floor in operation were determined. It is noted that undercarriage equipment – transformers, traction and auxiliary converters, air compressors – is a powerful source of vibrations of the floor of an electric train car. The results of the study of the influence of the characteristics of the undercarriage suspension on the vibration level are also given in [3]. It was concluded that it is necessary to take into account the instability of the stiffness of the rubber shock absorbers of the undercarriage equipment. It was determined that in the temperature range from −30 to +40 °C, the dynamic stiffness of suspension shock absorbers of undercarriage equipment changes from −20 % to +50 %, respectively, relative to normal conditions. The data from works [2, 3] are important for determining the characteristics of the vibration source in the mathematical modeling of excitations of the oscillating system of the car floor. However, works [2, 3] are limited only to the list of these sources. The task of analyzing the influence of floor characteristics as a vibration transmission chain on passengers was not set.

There are a number of papers reporting analysis of the impact of vibrations of different frequencies on the feeling of comfort and the condition of passengers. In particular, in [4], the results of the study of vibrations on the example of the Tehran-Andymeshk train (Iran) are given. The statistical relationship between psychological and physiological disorders in passengers was studied depending on age, gender, number of trips, and equivalent acceleration of the car. The significant influence of the equivalent acceleration of the car on the well-being of passengers has been confirmed.

Work [5] can also be considered as an extension of study [4] on the influence of car floor vibrations on passengers. The phenomenon of numbness of passengers' legs due to the vibration of the car floor in the frequency range of 20–50 Hz is analyzed in the work. It has been confirmed that local resonance of the floor occurs at frequencies of vibrations of the car frame close to the natural frequencies of the floor, which is the main cause of such a phenomenon as the numbness of passengers' legs. Optimization of the dynamic stiffness of the body's elastic supports is proposed in [5]. However, no analysis of the characteristics of individual floor layers was carried out because the subject of the study was the vibrations associated with the phenomenon of "numbness of the legs" of passengers.

Paper [6] analyzes the resonance frequencies of floor vibrations of high-speed trains and their impact on passenger comfort. As a result of experimental studies, the peak amplitudes of vibrations in the frequency range of 30–35 Hz were revealed. A method for reducing vibration based on a group of dynamic vibration dampers with different characteristics was proposed. However, the work does not report the results of studying the amplitude-frequency characteristics of the structural components of the multilayer floor, due to the fact that it was considered as a single element.

Based on the results of research [7], it is claimed that the vibrations of the car floor have mainly low frequencies in the range of up to 50 Hz. According to the researchers, the main source of vibrations are geometric defects of wheels and rails. The vibrations are transmitted to the passengers through the suspension stages and finally reach the body, floor, and seats. The dynamics of two subsystems of the car, namely the floating floor and the passenger seat, were studied. The test results demonstrate a strong influence of the connection between the floor and the seat on the frequency response of the oscillatory process. The paper proposes a methodology based on both laboratory tests and numerical models. The results are important for improving the vibration protection of passengers. However, the parameters of the floor layers were not investigated. The reason for this is the tasks set in the paper, which were limited to the analysis of the sources of vibrations and the mechanism for their transmission to the passenger seats.

Thus, works [4–7] draw important conclusions about the typical resonance frequencies of floor vibrations of passenger cars, which affect the state of comfort and feeling of passengers. Conclusions about the possibility of reducing floor vibration levels due to modernization of its structure are especially valuable. However, papers [4–7] do not contain specific data on the resonance characteristics of elements of a multilayer floor with improved vibration protection characteristics, which was not part of the research task.

Paper [8] investigates the effect of spring suspension parameters of an electric train car on passenger comfort. Ride comfort was assessed based on the NMV index (ride smoothness index). The theoretical part of the research is based on the use of a virtual vehicle model. The simulation results were verified experimentally during the trips of cars with three variants of stiffness and damping parameters of the primary and secondary suspension stages. The NMV index was calculated at 15 selected points of the body floor. It was established that lower stiffness values of the secondary suspension system lead to a decrease in ride comfort. However, the lower values of stiffness parameters and damping of the primary suspension system do not significantly affect the level of comfort. Paper [8], in accordance with the objectives of the study, is limited to the analysis of the influence of spring suspension parameters on passenger comfort indicators. At the same time, the research results do not contain solutions regarding the structure of the floor, in particular in the multi-layer version.

Study [9] considers a complex model of the "track-train-seat-passenger" oscillating system for lateral, vertical, and roll oscillations. The comfort characteristics of a high-speed train were studied. Based on the results of the study, it was found that the flexibility of the rail web does not have a significant effect on the vibrations of the car floor. However, the flexibility of the body caused an increase in floor vibration amplitudes at frequencies above 7.5 Hz. The worst comfort indicators were noted on the end parts of the body. Thus, the conclusions of work [9] confirm the conclusions of [8], in which the significant dependence of the amplitudes of oscillations on the location of passenger seats is noted. At the same time, the authors of [9] do not solve the problem of dampening vibrations transmitted by the floor due to the fact that the paper does not consider the option of a multilayer structure of the floor.

Papers [10, 11] consider the search for the best vibration-absorbing materials for the floor of railroad cars. In particular, study [10] discusses the possibilities of using bamboo materials for flooring. Its important properties are emphasized: vibration protection, noise protection, fire resistance, strength. The engineered composite material with bamboo inclusions combines the unique advantages of laminated construction and the bionics of the composite interface. The low density (730 kg/m^3) of the layered composite provides a specific modulus of elasticity of 0.013 GPa·m^3/kg, a vibration damping coefficient of 7 % and an impact toughness of 0.14 MJ/m^2. This is significantly higher than other wood-based composite materials used for high-speed train car floors. The proposed bamboo-wood composite material has great potential for use in the flooring of environmentally friendly railroad cars. Paper [11] compares the visco-elastic characteristics regarding the reduction of vibrations of various materials in passenger train carriages. In particular, three variants of materials were compared: damping material based on bitumen; water-based damping coating; butyl rubber damping material. Measurements of vibration and noise levels were carried out on experimental cars decorated with new materials in various modifications. It was established that the effect of reducing vibrations significantly depends on the frequency of disturbance. One result suggests that the first two materials reduce vibration over a wider frequency range of 63–1000 Hz than the last damping material. However, papers [10, 11] do not provide data on the AFR of the materials under investigation. This does not make it possible to draw a conclusion about the expediency of their use in the multi-layer floor of the car.

Study [12] examines the biodynamics of train passengers. The work uses an approach that involves obtaining acceleration in the passenger-train system and then processing acceleration signals to calculate passenger comfort. It is noted that a biodynamic model of a person with dynamic characteristics of a seat is necessary for an objective assessment of the level of comfort. It is proposed to calculate the Sperling comfort index based on the root-mean-square acceleration of different parts of the passenger's body. The approach to modeling the "seat-passenger" system described in [12], however, does not contain recommendations on how to combine the characteristics of the passenger's body with the general model of the "car base-floor-passenger" oscillating system.

Work [13] reports the results of a study of the influence of seat vibrations on the feeling of discomfort of a train passenger. The study was conducted on 3D models of the "seat-passenger" system using the finite element method. The relationship between the seat vibration, the pressure on the passenger's buttocks, and the heat released in the passenger's body under the influence of vibrations was determined. Based on the simulation test, it was established that the greater the vibration amplitude, the greater the heat release, and the lower the buttock pressure concentration. It was concluded that there is a correlation between the vibration parameters, the passenger's body weight, the type of seat covering material, and the passenger's comfort (feeling of heat while sitting). However, in the cited work, only the upper structure of the vibration protection chain, namely the seat, is considered. Other links of the oscillating floor system are not taken into account. Because of this, the problem of the influence of the characteristics of the multilayer structure of the floor on the vibration protection of the passenger was not solved.

In the process of study [14], it became necessary to determine the elastic-dissipative characteristics of the structural components of the passenger train car floor. To measure the stiffness and damping coefficient of the sandwich floor structure, a dynamic measurement was used on a thick plate resting on the system under investigation. At the same time, the hysteresis model of each component is generally considered as independent with its mechanical parameters. A similar method was used in [15], in which a procedure was proposed for determining the elasticity and damping constants of orthotropic sheet materials. The procedure is based on measuring resonant frequencies of low-frequency modes of thin rectangular plates with free edges. Research results [14, 15] are important at the stage preceding the mathematical modeling of the oscillating system. This stage is important for determining the elastic-dissipative parameters of each individual link of the vibration protection chain. However, works [14, 15] did not set the task of comprehensive analysis of sandwich floor components. This did not allow the authors to determine the influence of its structure on the general vibration isolation characteristics.

In [16], the experimental determination of the viscoelastic characteristics of composite materials was performed using a dynamic method. The principle of measurement involved recording the parameters of the damped vibrations of a rectangular plate of material, which was disturbed by removing it from its rest position. Oscillation amplitudes were measured by accelerometers and calculated using a digital method. The option of using the method of free oscillations described in [16] makes it possible to determine only partial characteristics of a separate material of the floor structure and does not allow the authors to obtain the parameters of the floor as a whole.

Analysis of the influence of track irregularities on the vibrations of the train car body was performed in [17]. The mechanism for perturbation of vibrations of the body from the track is considered. The problem was studied on the example of two variants of the mathematical model: a simplified model with three degrees of freedom and a more detailed multi-mass model. Research was conducted in the range of oscillation frequencies of 0–25 Hz. The theoretical results were compared with the experimental data from train tests. The data presented in the paper confirm that taking body flexibility into account when modeling vehicle vibrations gives more accurate results. In work [17], a force perturbation is used as a source of vibrations in the mathematical modeling of the vibration process. However, the use of kinematic disturbance of the base of the car body, which is the source of vibrations, would be closer to real conditions. In addition, the cited paper did not set the task of analyzing the influence of the characteristics of individual floor layers on its anti-vibration properties.

Paper [18] reports the results of dynamic tests of a passenger car with measurement of floor acceleration amplitudes. According to the results, it was found that the vertical vibration of the floor with a frequency of about 8 Hz is a typical result of the winding movement of the car. Moreover, the frequency of these oscillations remains practically unchanged over a wide range of movement speeds. The data given in [18] are useful for determining the characteristics of external excitation in the mathematical modeling of floor vibrations. However, the cited paper is limited only to the collection of experimental data on car vibrations and does not set the task of protecting passengers from vibrations, in particular at these frequencies.

The results of an experimental-numerical study on the influence of the degree of loading of a passenger car on the vertical vibrations of the floor are given in [19]. The test was conducted on three different types of cars with different variants for loading and distributing passengers by wagon. The main attention was paid to the damping effect, which is created by the passengers themselves, as part of the oscillating system. It was found that a significant decrease in acceleration amplitudes was observed when the cars were more densely packed. These results show that the passengers in the vibration system do not behave as rigid bodies, but as elements for damping body vibrations. A simple model of the passenger was proposed as a single-degree-of-freedom oscillator containing a mass, an elastic element, and a damper. However, the important question of the regularity of the distribution of vibrations over the area of the car depending on the degree of its loading remained unresolved in paper [19]. In addition, the authors did not consider the influence of the floor structure on the propagation of vibrations from the base of the car to the upper layer of the floor.

Our review of the literature [1–19] shows that the anti-vibration properties of the electric train floor depend on two groups of factors. The factors of the first group are related to the source of vibrations transmitted to the car body from trolleys and suspended equipment. At the same time, the lower lining (base) of the body becomes a direct disruptor of vibrations for the floor. Amplitude-frequency distribution of disturbances on the floor area remains an important, not fully covered issue. The second group of factors is the vibration-insulating properties of the floor, as a chain of propagation of vibrations from the base of the body to the upper floor covering. In the reviewed sources, AFR of the layers of the multi-layer structured floor of the cars were not investigated. Fragmentary data on the properties of various materials in terms of vibration damping do not make it possible to make the optimal choice of the floor structure. There is also no data on the use of rubber elastic elements – shock absorbers – for the floor. Therefore, it is expedient to study the effect of structure of the floor of an electric train car on the level of vibration protection of passengers. Depending on results, a justified choice of the rational structure for the floor of cars can be made in the process of modernizing electric trains of the suburban railroad.

3. The aim and objectives of the study

The purpose of our study is to determine the influence of the structure and parameters of the floor of the electric train car on the level of vibration protection of passengers. This will make it possible to reduce the level of vibrations transmitted to passengers from the seats and the floor.

To achieve the goal, the following tasks were set:

– to conduct a theoretical study on the process of transmission of vibrations from the floor of the car to the passenger;

– to experimentally determine the elastic-dissipative parameters of the structural components of the car floor;

– to construct the AFR of vibrations of the structural components of the floor of the car.

4. The study materials and methods

The object of our study is the process of vibration propagation through the floor of a railroad passenger car from the base to passenger seats.

The main hypothesis of the study is formulated as follows: improving the vibration-insulating properties of the floor of the passenger car could be achieved through the multilayer structure of the floor with a significant difference in the elastic-dissipative characteristics of the layers. Vibration transmitted to passengers is standardized according to ISO 2631-1-1997. Mechanical vibration and shock: Evaluation of human exposure to whole-body vibration.

One of the projects for modernizing suburban electric trains involves updating the structural scheme of the floor similar to the one used on passenger cars by the Kryukiv Wagon-Building Plant (KWBP). The vibration isolation properties of the floor for a modernized suburban electric train car from two modernization variants based on the design of the KWBP mod 61-776 cars are considered.

Fig. 1 shows variants for upgrading the floor: option No. 1 and option No. 2.

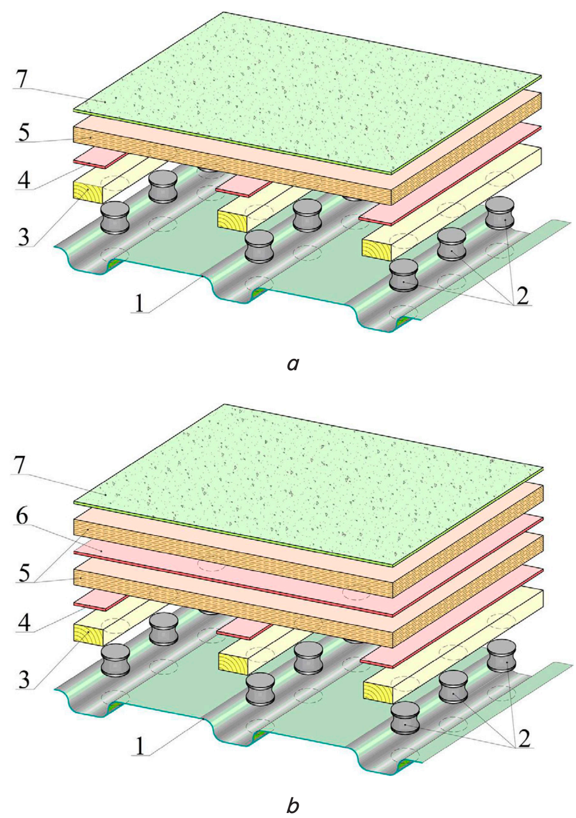


Fig. 1. Variants of the modernized floor of a car: *a* — option No. 1 — with the use of one layer of anti-vibration membrane; *b* — option No. 2 — using two layers of plywood floor and two layers of anti-vibration membrane; 1 — base of the car; 2 — rubber shock absorber; 3 — floor beam; 4 — anti-vibration membrane; 5 — floor sheet; 6 — anti-vibration membrane; 7 — upper floor covering

Floor variant No. 1 has the following structure:

– the base of the car (1) – the lower body cladding (profiled sheet B-PN-0-2.5-3.0. St3);

– rubber shock absorber (2) – B-14 rubber (vulcanization); floor beam (3) – pine board (edged) – 50 mm;

– anti-vibration membrane (4) (Vibrofix ML-2.6 mm – 100 mm strips);

– floor covering (5) (PFA-T fire-resistant plywood 20 mm);

– upper floor covering (7) (PVC, 2TEC (Belgium)).

Floor option No. 2 differs from option No. 1 by the presence of an additional layer of plywood and a second layer of anti-vibration membrane (6) – a continuous layer between two sheets of plywood (5).

The study of the anti-vibration properties of the car floor was performed on a mathematical model with the following simplifications and idealizations:

– in the model, the estimated floor section with an area S is considered;

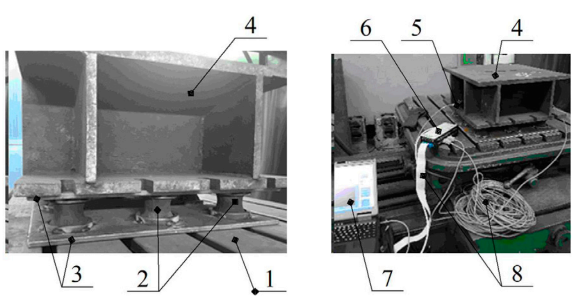
– the distributed parameters of the floor elements are assumed to be concentrated within the estimated area;

– the oscillating system has an input in the form of a harmonic kinematic disturbance from the side of the base of the car.

For the experimental determination of the elastic-dissipative parameters of the structural components of the mathematical model of the floor, a vibration bench is used, shown in Fig. 2.

Structurally, the bench includes a base plate (1), rubber shock absorbers under investigation (2), mounted on a vibrating plate (3). A group of shock absorbers (four or eight) rests on the second vibrating plate (3), on which the load (4) is installed. Two accelerometers (5) are installed on the lower load shelf on magnetic holders, connected to the signal conversion unit (6) and registration equipment (7) by a system of connection cables (8).

The parameters of structural components with elastic-dissipative properties: rubber shock absorbers and anti-vibration membrane were studied at the bench.



a *b*

Fig. 2. Experimental bench for researching the elastic-dissipative parameters of rubber shock absorbers: *a* – a photograph of the installation of rubber shock absorbers; *b* – general view of the experimental setup: 1 – base plate; 2 – rubber shock absorbers under investigation; 3 – vibrating plates; 4 – load; 5 – accelerometers; 6 – signal conversion unit; 7 – registration equipment; 8 – connection cables

The method of free oscillations was used to experimentally determine the stiffness parameters and damping coefficient of rubber elastic elements (shock absorbers) on a vibration bench (Fig. 2). The method makes it possible to determine the necessary parameters based on deciphering the oscillogram of the load's oscillations after its removal from the state of equilibrium – disturbance. Disturbance – in the form of an initial vertical deviation is carried out with the help of a single hammer blow on the load from above. After the impact, the load carries out free oscillations with damping until it comes to a complete stop. The oscillogram of oscillations is recorded by the system of measurement and registration of oscillations.

The system of control running tests (SCRT) of rolling stock units was used to measure vibration parameters, signal conversion, and to record the results of the experiment. SCRT was designed at the Regional Branch of JSC "Ukrzaliznytsia" "Research and Design and Technology Institute of Railroad Transport" ("NDKTI UZ"). The system is based on the National Instruments CompactRIO platform. In the course of the research, the option of SCRT "Measurement of accelerations under the autonomous mode of a "black box" with further statistical processing was implemented. Fig. 3 shows a block diagram of part of SCRT, which was used for our experiment.

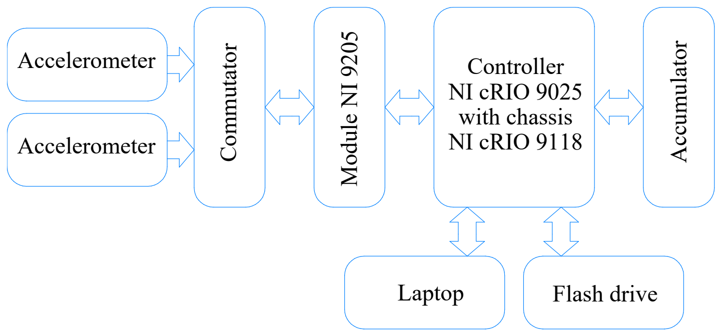


Fig. 3. Block diagram of part of the system of control running tests of rolling stock units, which was used to conduct the experiment

On the block diagram in Fig. 3 – accelerometer is low-frequency experimental UM type manufactured by "VD MAIS" (Ukraine). The accelerometer is built on the ADXL 278 microcircuit by Analog Devices (USA). The accelerometer has a magnetic attachment to the vibrating plate or load. Collection and visualization of measurement information is carried out by a National Instruments (USA) CompactRIO controller.

5. Investigating the influence of the structure and parameters of the floor of the electric train car on the level of vibration protection of passengers

5. 1. Mathematical modeling of the process of transmission of vibrations from the car floor to the passenger

The study of the influence of the parameters of the structural components of the car floor on its anti-vibration properties was performed based on a mathematical model. The calculation scheme for modeling vibrations transmitted from the base of the car through the floor and seats to the passenger is shown in Fig. 4.

In Fig. 4, the following designations are used:

– position 1 – base of the car;

– position 2 – elastic-dissipative element – a set of rubber shock absorbers with the following parameters:

$$C_2 = Csh \cdot S; \quad \beta_2 = \beta sh \cdot S, \tag{1}$$

where Csh, βsh are the specific stiffness and damping coefficient of the layer of rubber floor shock absorbers, determined experimentally, N/m^3 and N·s/m^3, respectively; S – area of the estimated floor area, m^2;

– position 3 – a system of wooden beams within the calculation area is considered as a single rigid body with a mass of m^3, kg:

$$m_3 = \gamma w \cdot F \cdot S / sp, \tag{2}$$

where γw – specific mass of timber, kg/m^3; F – cross-sectional area of the beam, m^2; sp – step of laying beams in the floor, m;

– position 4 – conditional elastic-dissipative element simulating a system of anti-vibration membrane strips glued to wooden beams. The stiffness and damping coefficient of the element is given as follows:

$$C_4 = Cam \cdot S \cdot a / sp; \quad \beta_4 = \beta am \cdot S \cdot a / sp, \tag{3}$$

where Cam, βam – specific stiffness and damping coefficient of the anti-vibration membrane, N/m^3 and N·s/m^3, respectively; a – width of bars, m;

– position 5 – a layer of plywood flooring is considered as a rigid body with a mass of m_5:

$$m_5 = d \cdot S \cdot \gamma p, \tag{4}$$

where d – thickness of the plywood decking, m; γp – specific mass of plywood flooring, kg/m^3;

– position 6 – conditional elastic-dissipative element simulating a layer of anti-vibration membrane between plywood sheets. The stiffness and damping coefficient of the membrane correspond to the experimentally determined specific elastic-dissipative parameters:

$$C_6 = Cam \cdot S; \quad \beta_6 = \beta am \cdot S; \tag{5}$$

– position 7 – conditional elastic-dissipative element simulating the floor covering layer. The stiffness and damping coefficient are taken in accordance with the determined specific elastic-dissipative parameters:

$$C_7 = C\,\mathrm{cov} \cdot S; \quad \beta_7 = \beta\,\mathrm{cov} \cdot S, \tag{6}$$

where Ccov, βcov are the specific stiffness and damping coefficient of the anti-vibration membrane, N/m^3 and N·s/m^3, respectively.

The specific mass of loading the car mp – the mass of passengers and passenger seats per unit area of the car – depends on the degree of filling the car with passengers and is determined from the formula:

$$mp = \frac{np \cdot kp \cdot mpas + mseat}{Sv}, \tag{7}$$

where np – passenger capacity of the car, pas.; kp – degree of filling of the car with passengers; $mpas$ – conditional weight of one passenger including luggage, kg; $mseats$ – total mass of passenger seats in the carriage, kg; Sv – floor area of the car, m^2.

With a passenger capacity of 136 passengers (electric train EPL2t) the specific mass of loading can vary from 100 to 450 kg/m^2 depending on the filling of the car with passengers.

The mathematical model of the oscillating system "floor-passenger" is built on the classical Lagrange equation of the II kind:

$$\frac{d}{dt}\frac{\partial K}{\partial \dot{q}_i} + \frac{\partial D}{\partial \dot{q}_i} + \frac{\partial P}{\partial q_i} = 0, \tag{8}$$

where K, P are the kinetic and potential energies of the oscillating system, respectively;

D is the scattering function of conditional dissipative elements;

q_i, $\dot{q}_i$ – vertical generalized coordinates and generalized velocities of the inertial elements of the oscillating system.

The following generalized coordinates are adopted for the calculation scheme:

q_1=q_3 – displacement of the center of the system of wooden beams;

q_2=q_{51}, q_3=q_{52} – movements of the centers of plywood floors of the first and second layers, respectively;

q_4=q_p – movement of the floor under the passenger.

The system of generalized velocities is accepted according to displacements:

$$\dot{q}_1 = \dot{y}_3; \; \dot{q}_2 = \dot{y}_{51}; \; \dot{q}_3 = \dot{y}_{52}; \; \dot{q}_3 = \dot{y}_p. \tag{9}$$

The kinetic energy of the oscillating system according to the calculation scheme in the adopted coordinate system is determined from the expression:

$$2K = m_3 \cdot \dot{y}_3^2 + m_5 \cdot \left(\dot{y}_{51}^2 + \dot{y}_{52}^2\right) + m_p \cdot \dot{y}_p^2. \tag{10}$$

The potential energy of the oscillating system associated with the presence of conditional elastic elements depends on the relative movements of their attachment points:

$$\begin{aligned} 2P = {} & C_2 \cdot \left(y_o - y_3\right)^2 + C_4 \cdot \left(y_3 - y_{51}\right)^2 + \\ & + C_6 \cdot \left(y_{51} - y_{52}\right)^2 + C_7 \cdot \left(y_{52} - y_p\right)^2, \end{aligned} \tag{11}$$

where y_o are the vertical movements of the car base associated with the external kinematic disturbance of the oscillating system.

Dispersion function of rubber shock absorbers and anti-vibration diaphragm:

$$\begin{aligned} 2D = {} & \beta_2 \cdot \left(\dot{y}_o - \dot{y}_3\right)^2 + \beta_4 \cdot \left(\dot{y}_3 - \dot{y}_{51}\right)^2 + \\ & + \beta_6 \cdot \left(\dot{y}_{51} - \dot{y}_{52}\right)^2 + \beta_7 \cdot \left(\dot{y}_{52} - \dot{y}_p\right)^2. \end{aligned} \tag{12}$$

The system of differential equations describing the oscillations of the modeled floor in the Cauchy form is as follows:

$$\begin{cases} \ddot{y}_3 = \dfrac{\left(\beta_2 + \beta_4\right) \cdot \dot{y}_3 - \beta_2 \cdot \dot{y}_o - \beta_4 \cdot \dot{y}_{51} + \left(C_2 + C_4\right) \cdot y_3 - C_2 \cdot y_o - C_4 \cdot y_{51}}{m_3}; \\ \ddot{y}_{51} = \dfrac{\left(\beta_4 + \beta_6\right) \cdot \dot{y}_{51} - \beta_4 \cdot \dot{y}_3 - \beta_6 \cdot \dot{y}_{52} + \left(C_4 + C_6\right) \cdot y_{51} - C_4 \cdot y_3 - C_6 \cdot y_{52}}{m_5}; \\ \ddot{y}_{52} = \dfrac{\left(\beta_6 + \beta_7\right) \cdot \dot{y}_{52} - \beta_6 \cdot \dot{y}_{51} - \beta_7 \cdot \dot{y}_p + \left(C_6 + C_7\right) \cdot y_{52} - C_6 \cdot y_{51} - C_7 \cdot y_p}{m_5}; \\ \ddot{y}_p = \dfrac{\beta_7 \cdot \left(\dot{y}_p - \dot{y}_{52}\right) + C_7 \cdot \left(y_p - y_{52}\right)}{m_p}. \end{cases} \tag{13}$$

The external kinematic disturbance imposed on the base of the car is described by harmonic functions:

$$y_o(t) = Y_o \cdot \sin(\omega \cdot t); \quad \dot{y}_o(t) = \frac{d}{dt} y_o(t), \tag{14}$$

where Y_o, ω are the amplitude and frequency of oscillations of the perturbation of the car base.

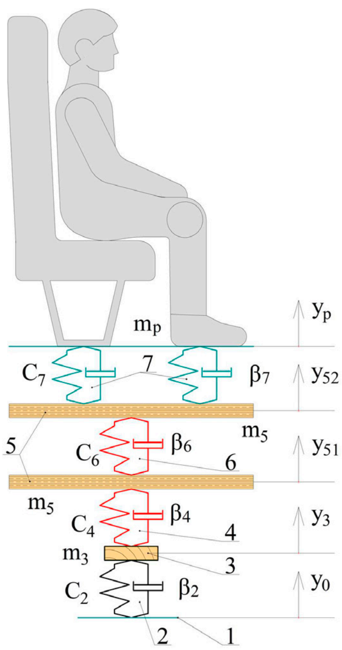


Fig. 4. Calculation scheme for modeling vibrations transmitted from the base of the car through the floor and seats to the passenger

5. 2. Experimental determination of the elastic-dissipative parameters of the structural components of the car floor

For the mathematical modeling of vibration protection of the car floor, data on the elastic-dissipative parameters of its components are required: a layer of rubber shock absorbers; anti-vibration membrane; the top layer of the floor.

According to the classical theory of oscillations, the natural oscillation period of a linear mechanical oscillator is determined from the formula:

$$T = 2\pi\sqrt{\frac{m}{C}}, \tag{15}$$

where T is the oscillation period, s;

m is the weight of the oscillating load, kg;

C is the stiffness of the elastic element of the oscillating system, N/m.

Hence, the stiffness of a separate rubber shock absorber of the car floor Cra [N/m] can be determined from the formula:

$$Cra = 4\pi^2 \frac{m_W}{n \cdot T^2}, \tag{16}$$

where m_W is the total mass of the load and the upper vibrating plate of the vibrating bench, kg;

n is the number of rubber shock absorbers installed on the bench during the experiment.

According to ISO/TR 19201:2013. Mechanical vibration. Methodology for selecting appropriate machinery vibration standards, the logarithmic decrement of vibration damping δ of the elastic support can be determined from the ratio:

$$\delta = \ln\frac{A_i}{A_{i+1}}, \tag{17}$$

where A_i, A_{i+1} are pairs of adjacent amplitudes from the experimental oscillogram recording of oscillations on the bench.

Hence, the damping coefficient of an individual rubber shock absorber βra [N·s/m] for a system of n shock absorbers can be determined from the formula:

$$\beta ra = \frac{2 \cdot m_W}{n \cdot T}\delta. \tag{18}$$

Stiffness and damping coefficient of rubber elastic elements (shock absorbers) were determined from experimental oscillograms of free oscillations of mass m_W.

Fig. 5 shows a fragment of the oscillogram recording the free oscillations of the vibrating plate together with the load. For the analysis, a part of the oscillogram in the time range of 0.3–0.85 s was used, when the oscillations associated with the structural stiffness of the vibrating plate and the load are damped.

Due to its layered construction, the floor of a passenger car is a classic example of a system with distributed parameters. Such parameters are the mass of solid layers of the floor and the stiffness and damping coefficient of its elastic-dissipative structures. To simplify the calculation scheme of the floor during the mathematical modeling of vibration protection, it is proposed to introduce the concept of specific parameters. Such parameters include specific stiffness and specific damping coefficient. This method makes it possible to replace the system with distributed parameters by a discrete system with concentrated parameters per unit area.

The specific stiffness of the floor layer with rubber shock absorbers Csh [N/m^3] depends on the diagram of their installation Era:

$$Csh = Cra \cdot Era, \tag{19}$$

where Era is the diagram of installation of rubber shock absorbers – the number of shock absorbers per unit of floor area, m^{-2}.

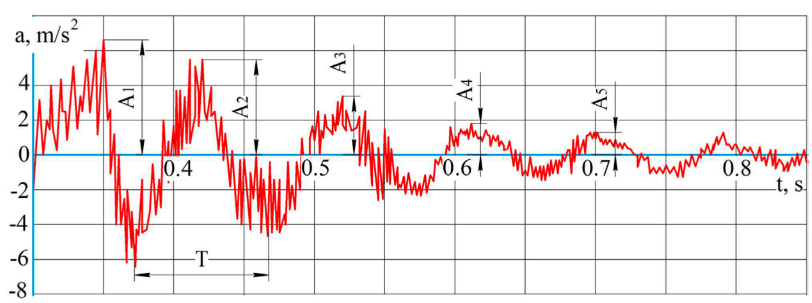


Fig. 5. Example of processing an experimental oscillogram in the time range of 0.3–0.85 s

The specific damping coefficient of the floor layer with rubber shock absorbers – βsh [N·s/m^3] also depends on the diagram of their installation Era:

$$\beta sh = \beta ra \cdot Era. \tag{20}$$

An example of determining the elastic-dissipative parameters of rubber shock absorbers for one of the variants (number of rubber shock absorbers n=4; load weight n_W=146.7 kg) is given in Table 1.

Unlike rubber shock absorbers, the parameters of which can be considered concentrated, the anti-vibration membrane and floor covering have a distributed nature of elastic-dissipative parameters. It is customary to measure them in specific units per unit area: specific stiffness, N/m^3; specific damping coefficient, N·s/m^3. Square samples of anti-vibration membrane measuring 500×500 mm of the Vibrofix ML-2.6 brand were used in the experiment. For the experimental determination of stiffness and damping coefficient, the same bench and methodology were used as for rubber shock absorbers (Fig. 2).

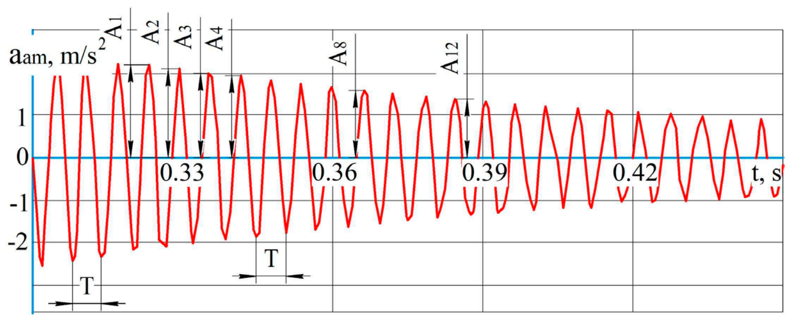


Fig. 6. Fragment of an oscillogram of free oscillations of an oscillating system for determining the elastic-dissipative parameters of an anti-vibration membrane

Table 1

Example of determining the elastic-dissipative parameters of rubber shock absorbers for one of the variants for floor parameters

n_W=146.5 kg; n=4	$t(A_i)$, s	0.350	0.437	0.524	0.611	0.698	T_{ave}
	T, s	0.087	0.087	0.087	0.087	0.087	
Oscillation amplitudes A_i, 10^{-3}m/s^2	A_i, m/s^2	A_1	A_2	A_3	A_4	A_5	A_{ave}
		7.50	5.65	3.50	2.05	1.20	
Ratio A_i/A_{i+1}		1.45	1.61	1.71	1.71	1.62	
Logarithmic decrement of oscillation attenuation – δ						0.482	
Free oscillation period – T, s						0.087	
Stiffness of 4 shock absorbers, N/m						190 835	
Stiffness of one shock absorber – Cra, N/m						47 710	
Damping coefficient of 4 shock absorbers, N·s/m						1625	
Damping coefficient of one shock absorber – βra, N·s/m						406	

Fig. 6 shows a fragment of the oscillogram recording the free oscillations of the vibrating plate together with the load installed on the anti-vibration membrane. For analysis, part of the oscillogram in the time range of 0.3–0.45 s was used, when the oscillations associated with the structural rigidity of the vibrating plate are attenuated.

An example of determining the elastic-dissipative parameters of an anti-vibration membrane is given in Table 2.

Table 2

Example of determining the elastic-dissipative parameters of an anti-vibration membrane (load mass n_W=146.7 kg)

t_i, s	0.203	0.211	0.218	0.226	0.235	0.243	0.251	0.261	0.268	0.281
T, s	0.007	0.008	0.008	0.009	0.008	0.008	0.009	0.008	0.008	0.008
A_i, m/s^2	A_1	A_2	A_3	A_4	A_5	A_6	A_7	A_8	A_9	A_{10}
	2.65	2.46	2.29	2.17	2.02	1.89	1.72	1.60	1.49	–
A_i/A_{i+1}	1.077	1.072	1.057	1.072	1.072	1.097	1.072	1.077	1.075	
Logarithmic decrement of oscillation attenuation – δ									0.072	
Free oscillation period – T, s									0.008	
Anti-vibration membrane rigidity – Cam, N/m									102,715,100	
Coefficient of anti-vibration membrane damping – βam, N·s/m									1,360	

A similar procedure was used to determine the elastic-dissipative parameters of the floor covering.

5. 3. Integration of the system of differential equations and construction of amplitude-frequency characteristics of the structural components of the floor

The integration of the system of differential equations (13) was performed using the Given-Odesolve procedure of the computer mathematical package Mathcad. The solution to the system of differential equations is the time functions of the movements and velocities of points of the oscillating floor system corresponding to the independent coordinates of the system of equations. Such functions are: $y_3(t)$; $y_{51}(t)$; $y_{52}(t)$; $y_p(t)$; $\dot{y}_3(t)$; $\dot{y}_{51}(t)$; $\dot{y}_{52}(t)$; $\dot{y}_p(t)$.

When integrating the system of equations, the following initial conditions were adopted: amplitude of disturbance fluctuations Y_o=10^{-3} m; the disturbance frequency ω varied for different calculation variants from 1 Hz to 100 Hz; the initial values of displacements and velocities were equal to 0.

In the process of treating the results of solving the system of equations, the time dependences of the accelerations of the floor elements were derived:

$$a_3(t)=\frac{d}{dt}\dot{y}_3(y);\quad a_{51}(t)=\frac{d}{dt}\dot{y}_{51}(y);$$

$$a_{52}(t)=\frac{d}{dt}\dot{y}_{52}(y);\quad a_p(t)=\frac{d}{dt}\dot{y}_p(y).$$

Fig. 6 shows an example of one of the variants of the resulting solutions-functions for the following conditions: Cra=4.8·10^4 N/m^3; βra=0.4·10^3 N·s/m^3; Cam=1.03·10^8 N/m^3; βam=1.36·10^3 N·s/m^3; $Cvol$=2.8·10^8 N/m^3; βvol=1.8·10^3 N·s/m^3; mp=150 kg; Era=4 units/m^2; S=1 m^2.

Fig. 7 shows examples of solving the system of differential equations of vibrations of the floor layers: Fig. 7, *a–d* – movement of floor elements; Fig. 7, *e–h* – acceleration of floor elements.

Fig. 8, 9 illustrate the results of processing the solution to the system of differential equations. Fig. 8 shows examples of AFR for each element of the car floor in the form of dependences $Y/Y_o(f)$ for different variants of the structural structure: variant No. 1 and variant No. 2. In Fig. 8, we marked: Y_o – amplitude of movement of the base of the car (kinematic disturbance); Y_3 – amplitude of movement of wooden beams; Y_{51}, Y_{52} – displacement amplitudes of plywood floors; Y_p – amplitude of movement of the floor surface.

Fig. 9 shows the estimated dependences of the resonant frequency of the car floor dependent on the parameters of the oscillating system: Fig. 9, *a* – dependent on the specific weight of loading the car mp and the diagram of rubber shock absorber installation Era; Fig. 9, *b* – dependent on the diagram of installation of rubber shock absorbers Era and the stiffness of anti-vibration diaphragm Cam.

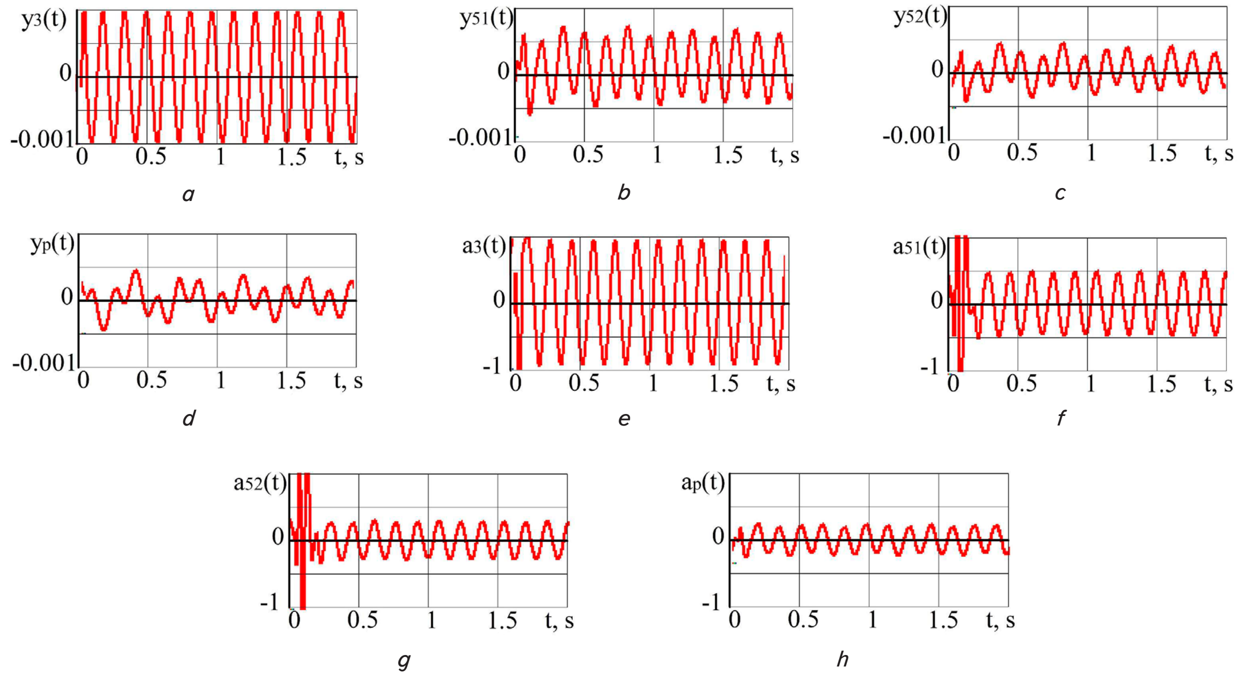


Fig. 7. Examples of solving a system of differential equations: *a*–*d* – movement of floor elements; *e*–*h* – acceleration of floor elements

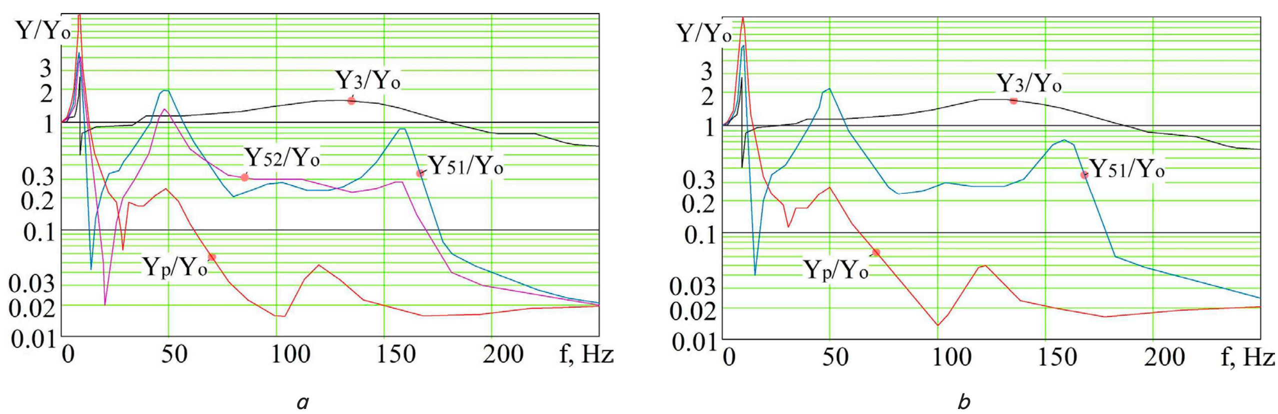


Fig. 8. Examples of AFR for elements of the modernized floor: *a* – option No. 1; *b* – option No. 2

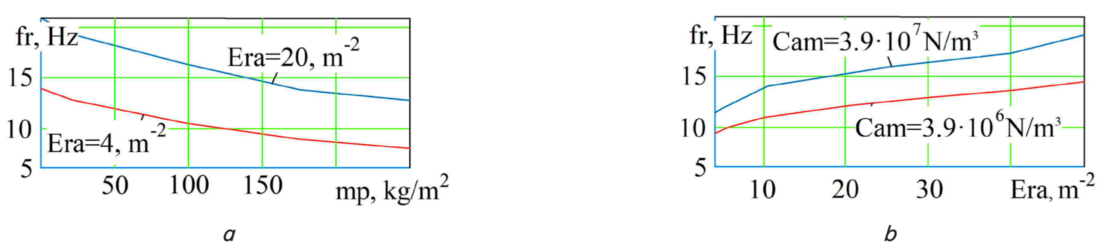


Fig. 9. Dependence of the resonant frequency *fr* of the floor on the parameters of oscillating system: *a* – on the specific weight of the loading of the car *mp* and the diagram of installation of rubber shock absorbers *Era*; *b* – on the diagram of installation of rubber shock absorbers *Era* and the rigidity of anti-vibration diaphragm *Cam*

For the calculations, two variants of the values of the rubber shock absorber installation diagram *Era* were used – 4 units/m^2 and 20 units/m^2. This choice is explained as follows:

1) *Era*=4 units/m^2 is the diagram of shock absorber installation, which is used on the basic model of the car – 61-776;

2) *Era*=20 units/m^2 corresponds to the densest arrangement of shock absorbers according to the technological limitations of installation.

6. Discussion of results related to investigating the anti-vibration properties of the passenger car floor

The study of the impact of vibration isolation characteristics of two variants of the multi-layer floor of an electric train car (Fig. 1) was performed on a mathematical model. Before performing the floor AFR theoretical study, two sets of input data had to be determined. The first is the parameters of the kinematic perturbation of the oscillating system, which is close to the operating conditions. The second is the elastic-dissipative characteristics of the floor layers.

The parameters of the kinematic disturbance were determined based on analysis of the characteristic frequencies of the sources of floor vibrations in operation, given in works [2, 7, 8, 17, 18]. A forced kinematic disturbance according to the harmonic law (14) was used as the input of the oscillatory system. A series of AFR calculations were performed for a retrofitted oscillating floor system. For each of the calculations, fixed amplitude and frequency of movements of the base of the car were assumed. As a development of studies [2, 7, 8, 17, 18], the task was set to compare the resonant frequencies of the floor elements under investigation with the characteristic operating frequencies of the car's vibrations in operation. Special attention was paid to frequencies that can cause physiological disorders [4], numbness of legs [5], or general discomfort of passengers [6, 13]. Most of the calculation variants were performed for these frequencies.

The experimental determination of the elastic-dissipative characteristics of the floor layers was performed on a vibration test bench (Fig. 2, 3) using the method of free oscillations. The method of free oscillations described in [14–16] was improved. Two aspects of the method have undergone improvements: the method of primary excitation of oscillations and the method of filtering measurement results. The shock method was used for the primary excitation of oscillations. The essence of the shock method is that the initial vertical deviation of the oscillating load relative to the equilibrium position was carried out using a single external shock. The peculiarity of the technique for filtering the measurement results is that the oscillograms are decoded (Fig. 5, 6), starting from the time when the oscillations associated with the structural rigidity of the elements of the vibration bench subside.

The calculation scheme of the oscillating system (Fig. 4) was improved in comparison with [12] in the part of the biodynamic model of the passenger. The introduction of specific load parameters and elastic-dissipative characteristics of floor elements is a development of the approach to vibration modeling given in [14–18]. In particular, the carriage loading is considered as the specific mass of passengers and passenger seats per unit of area (7). Similar to the approach used in [19], passengers are considered as a conditionally solid body resting on an elastic-dissipative floor covering.

The calculation scheme is constructed as a linear four-mass oscillator with elastic-dissipative connections between layers. The mathematical model is a system of four Lagrange differential equations (13). The solutions to the system of equations are the time functions of movements and velocities of the floor elements (Fig. 7).

For comparison, two variants of the floor structure were considered (Fig. 1). Option No. 1 is a five-layer scheme with two solid and three elastic-dissipative elements. Option No. 2 is a seven-layer floor scheme with three solid and four elastic-dissipative elements.

The obtained time functions of movements and velocities of the floor elements (Fig. 7) show that at all frequencies of disturbance that were studied, a decrease in movements and accelerations in the chain from the base of the car to the floor surface is observed. Movements are reduced by 2.7–4 times, and accelerations are reduced by 6–9 times.

The calculated AFR of the floor elements (Fig. 8) made it possible to determine the resonance frequency zones of the oscillating system of the car floor. The resulting layer-by-layer AFRs show a stepwise change in vibration parameters in the floor vibration protection chain. When comparing the two considered floor variants, the advantages of the second option (Fig. 8, *b*) are not observed. The characteristics of links 51 and 52 are practically identical. A layer of rubber shock absorbers has a decisive influence on the vibration protection characteristics of the floor. At the same time, the coefficient of increase of oscillations at the resonant frequency reaches sufficiently high values — up to 10, which indicates the insufficient degree of damping.

The resulting AFRs showed that the region of resonant frequencies of the first harmonic is in the range of 8–22 Hz (Fig. 9). An important result is the obtained dependence of the resonance frequency on the specific weight of the car loading. It is important that when the car is loaded with passengers from the minimum (empty car) to the maximum (150 % passenger capacity), the resonant frequency can decrease from 15–22 Hz to 8–12 Hz. Our results are a further development of study [19], in which the influence of the floor structure on the propagation of vibrations from the base of the car to the top layer of the floor was not considered.

The advantage of the structural execution of the first layer of the modernized floor is the possibility to vary its elastic and dissipative parameters by changing the diagram of installation of shock absorbers – *Era*. For example, by increasing *Era* from 4 to 20 units/m^2, the resonance frequency can be increased by 2.25 times (Fig. 9). Works [4–8, 12, 13] contain important data on vibration frequencies "dangerous" for humans, which should not be allowed in passenger cars, for example, 7.5 Hz, 38 Hz. Based on this, it can be considered preferable to install shock absorbers *Era* in the range of 5–6 units/m^2. In contrast to studies [4–8, 12, 13], in which only data on the characteristic frequencies of car vibrations are given, our results make it possible to determine the resonant characteristics of the floor under investigation.

The resulting AFR of the floor component vibrations confirm the hypothesis about the possibility of improving the vibration-isolating properties of the passenger car based on a multilayer structure with a significant difference in the elastic-dissipative characteristics of the structural layers.

The conditions for the practical use of our results are the use of the diagram of installation of rubber shock absorbers of the car floor in the range of 5–6 units/m^2.

Meanwhile, this study is limited to a specific floor structure that is already in use on another rolling stock. This makes it difficult to perform the optimization of the structural structure according to the given objective function.

A certain disadvantage of the proposed method of transition from distributed to concentrated parameters is the impossibility of taking into account the unevenness of the distribution of passengers on the car, which is typical for electric trains.

The verification of the mathematical model of floor vibrations is planned to be carried out during the tests of the modernized car with the proposed floor parameters, in particular, the diagram of installation of rubber shock absorbers.

This work could be advanced by studying the influence of the "floating" floor structure on the indicators of vibration protection of passengers. This may be related to some difficulties, namely the technical solutions for connecting the floor with the side walls of the car.

7. Conclusions

1. A mathematical model of the propagation of vibrations from the base of the car through the floor to the passengers has been built. The multi-layer calculation scheme of the floor in two variants was considered: five-layer and seven-layer. The parameters of the floor layers obtained experimentally by the method of free oscillations were used in the calculations. The specific mass, elastic, and dissipative parameters of the layers were applied. Thus, a transition has been performed in modeling from distributed parameters to linear concentrated ones.

2. Input data for the mathematical model of floor vibrations were obtained at the experimental bench. In particular, the following elastic-dissipative parameters of the structural components of the car floor were determined:

– stiffness of the rubber shock absorber – $4.7 \cdot 10^4$ N/m;

– damping coefficient of the rubber shock absorber – 406 N·s/m;

– specific stiffness of the anti-vibration membrane – $2.6 \cdot 10^7$ N/m^3;

– the specific damping coefficient of the anti-vibration membrane is $3.5 \cdot 10^2$ N·s/m^3.

3. AFR of the structural components of the car floor ha been built. AFR analysis revealed the presence of a region of resonance frequencies of the first harmonic in the range of 8–22 Hz. No advantages of a seven-layer floor over a five-layer floor were found. AFR for the first and second layers of plywood flooring are identical. It has been confirmed that the influence of the layer of rubber shock absorbers on AFR of the floor is decisive. An increase in the shock absorber installation diagram from 4 to 20 units/m^2 increases the resonance frequency of floor vibrations by 2.25 times. Based on data on "dangerous" vibration frequencies for humans, the preferred value for the shock absorber installation diagram was derived – 5–6 units/m^2.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study, as well as the results reported in this paper.

Funding

The study was conducted without financial support.

Data availability

All data are available, either in numerical or graphical form, in the main text of the manuscript.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

Acknowledgments

The authors are grateful to Pavel Prokopenko, head of the engineering department of the Research and Design and Technology Institute, AT "Ukrzaliznytsia", for technical assistance in conducting experiments, the results of which are used in the paper.

References

1. Kulbovskyi, I., Sapronova, S., Holub, H., Tkachenko, V., Afanasieva, I., Safronov, O. (2019). Development of a model for managing the quality of repair and maintenance of rolling stock in transport infrastructure projects. Proceedings of 23rd International Scientific Conference. Transport Means 2019, 201–205. Available at: https://transportmeans.ktu.edu/wp-content/uploads/sites/307/2018/02/Transport-means-2019-Part-1.pdf
2. Shostak, Y., Holovashchenko, O., Reshetnikov, Y., Tkachenko, V. (2024). Improving the technological process of balancing electric machine rotors on a balancing machine. Applied Mechanics, 4 (7 (130)), 60–69. LOCKSS. https://doi.org/10.15587/1729-4061.2024.309867
3. Gong, D., Duan, Y., Wang, K., Zhou, J. (2019). Modelling rubber dynamic stiffness for numerical predictions of the effects of temperature and speed on the vibration of a railway vehicle car body. Journal of Sound and Vibration, 449, 121–139. https://doi.org/10.1016/j.jsv.2019.02.037
4. Nassiri, P., Koohpaei, A. R., Zeraati, H., Shalkouhi, P. J. (2011). Train Passengers Comfort with Regard to Whole-Body Vibration. Journal of Low Frequency Noise, Vibration and Active Control, 30 (2), 125–136. https://doi.org/10.1260/0263-0923.30.2.125
5. Gong, D., Wang, K., Duan, Y., Zhou, J. (2019). Car body floor vibration of high-speed railway vehicles and its reduction. Journal of Low Frequency Noise, Vibration and Active Control, 39 (4), 925–938. https://doi.org/10.1177/1461348419850921
6. You, T., Zhou, J., Thompson, D. J., Gong, D., Chen, J., Sun, Y. (2021). Vibration reduction of a high-speed train floor using multiple dynamic vibration absorbers. Vehicle System Dynamics, 60 (9), 2919–2940. https://doi.org/10.1080/00423114.2021.1928248
7. Dižo, J., Blatnický, M., Gerlici, J., Leitner, B., Melnik, R., Semenov, S. et al. (2021). Evaluation of Ride Comfort in a Railway Passenger Car Depending on a Change of Suspension Parameters. Sensors, 21 (23), 8138. https://doi.org/10.3390/s21238138
8. La Paglia, I., Li, Q., Ripamonti, F., Corradi, R. (2023). Experimental and numerical investigation of the dynamics of the floating floor and passenger seats of a railway vehicle. In Proceedings of the 29th International Congress on Sound and Vibration, 1–7. Available at: https://re.public.polimi.it/bitstream/11311/1249627/1/full_paper_36_20230429103100641.pdf

9. Wu, J., Qiu, Y. (2021). Modelling and ride comfort analysis of a coupled track-train-seat-human model with lateral, vertical and roll vibrations. Vehicle System Dynamics, 60 (9), 2988–3023. https://doi.org/10.1080/00423114.2021.1933088
10. Han, S., Chen, F., Yu, Y., Zheng, Z., Chen, L., Wang, G. (2022). Bamboo-Inspired Renewable, Lightweight, and Vibration-Damping Laminated Structural Materials for the Floor of a Railroad Car. ACS Applied Materials & Interfaces, 14 (37), 42645–42655. https://doi.org/10.1021/acsami.2c09785
11. Fan, R., Meng, G., Yang, J., He, C. (2009). Experimental study of the effect of viscoelastic damping materials on noise and vibration reduction within railway vehicles. Journal of Sound and Vibration, 319 (1-2), 58–76. https://doi.org/10.1016/j.jsv.2008.03.071
12. Kumar, V., Rastogi, V., Pathak, P. (2016). Simulation for whole-body vibration to assess ride comfort of a low–medium speed railway vehicle. SIMULATION, 93 (3), 225–236. https://doi.org/10.1177/0037549716679254
13. Sugiono, S., Oktavianty, O., Sulistyarini, D. H., Nugroho, W. S., Wiryawan, E. et al. (2021). Analysis of Train Passenger Comfort Related to the Vibration and Heat It Creates. Proceedings of the Second Asia Pacific International Conference on Industrial Engineering and Operations Management, 3518–3525. Available at: http://ieomsociety.org/proceedings/2021indonesia/605.pdf
14. Rébillat, M., Boutillon, X. (2011). Measurement of relevant elastic and damping material properties in sandwich thick plates. Journal of Sound and Vibration, 330 (25), 6098–6121. https://doi.org/10.1016/j.jsv.2011.07.015
15. McIntyre, M. E., Woodhouse, J. (1988). On measuring the elastic and damping constants of orthotropic sheet materials. Acta Metallurgica, 36 (6), 1397–1416. https://doi.org/10.1016/0001-6160(88)90209-x
16. Botelho, E. C., Campos, A. N., de Barros, E., Pardini, L. C., Rezende, M. C. (2005). Damping behavior of continuous fiber/metal composite materials by the free vibration method. Composites Part B: Engineering, 37 (2-3), 255–263. https://doi.org/10.1016/j.compositesb.2005.04.003
17. Cheli, F., Corradi, R. (2011). On rail vehicle vibrations induced by track unevenness: Analysis of the excitation mechanism. Journal of Sound and Vibration, 330 (15), 3744–3765. https://doi.org/10.1016/j.jsv.2011.02.025
18. Wang, Q., Zeng, J., Wei, L., Zhou, C., Zhu, B. (2017). Reduction of vertical abnormal vibration in carbodies of low-floor railway trains by using a dynamic vibration absorber. Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, 232 (5), 1437–1447. https://doi.org/10.1177/0954409717731234
19. Tomioka, T., Takigami, T. (2015). Experimental and numerical study on the effect due to passengers on flexural vibrations in railway vehicle carbodies. Journal of Sound and Vibration, 343, 1–19. https://doi.org/10.1016/j.jsv.2015.01.001