DOI: 10.32703/2617-9059-2024-44-4

UDC 621.3

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Comparative assessment of resonant frequencies of the floor of suburban electric train cars

The work is devoted to the study of the resonant frequency characteristics of the floor of passenger cars for the modernization of suburban electric trains of the EPL2, EPL9 series. A horizontal vibrator loading scheme was adopted for the experiments. The aim of the study is an experimental comparative assessment of the resonant frequencies of the floor of cars with different variants of the structural structure. An experimental and computational method for determining the resonant frequency of the oscillatory system based on limited experimental data, namely the excitation frequency and the measured coefficient of reduction of the force amplitude, is proposed. Based on the tests, the frequency characteristics of the relative change in the amplitude of accelerations on the surface of the floor of different structures were obtained. The relative amplitudes are presented in the form of the amplitude growth coefficient k. It is found that the floor amplitude growth coefficient does not depend on the excitation amplitude and depends only on the frequency. The frequency characteristics of the floor design options with rubber shock absorbers differ significantly from the basic version in terms of resonant frequency – 104 Hz versus 69–75 Hz. The introduction of layers of the vibration damping membrane into the design scheme has a minor effect on the resonant frequency of the floor: 75–71–69 Hz.

Keywords: *railway car, passenger comfort, floor vibration isolation, vibration levels, experimental studies.*

Introduction. Suburban railway plays an important role in ensuring labor mobility of the population along the routes "suburban zone – megalopolis". The total time of daily work trips of citizens can be one to two hours. In fact, a trip to work or from work in an electric train is part of the worker's production activity. The level of passenger comfort during the trip determines the productivity of his activity and state of health. Suburban electric trains are characterized by a periodic mode of movement with frequent accelerations and braking in a wide range of speeds and frequencies of vibrations of the body floor. The level of vibrations perceived by passengers depends on the mechanism of their origin, the characteristics of the vibration transmission channel to the passenger and the quality of vibration insulation of the car floor.

Each of the above factors depends on the technical condition and characteristics of the passenger car. Today, the technical condition of passenger cars of the rolling stock of the railways of Ukraine is characterized by significant wear. About 85% of the rolling stock requires repair [1]. By order of JSC

«Ukrzaliznytsia» PJSC «Kyiv Electric Car Repair Plant» (KECRP) carries out major repairs and modernization of electric trains of the EPL2T, EPL9T etc. series. One of the main tasks of modernization is to increase the comfort and safety of passenger transportation and the working conditions of the locomotive crew. In addition, the modernized electric trains acquire a modern appearance and more efficient lighting devices. Fig. 1 shows the ED9M electric train before and after modernization at KECRP.

Fig. 1. **Electric train ED9M before and after modernization**

Modernization of electric trains includes measures aimed at reducing the level of vibrations transmitted to the passenger compartment through the floor. To form the modernization project, a study was conducted to determine the influence of the design and characteristics of the car floor on the level of vibrations transmitted to passengers in motion.

Analysis of recent research and problem statement. The main source of vibrations and noise perceived by passengers are dynamic processes in the contacts of wheel pairs with the track. In [2], an analysis of the parameters of vibrations arising from short rail irregularities with a wavelength of 15–20 mm is presented. It is proposed to reduce the impact of track vibrations on passengers by using vibration dampers in the range of 500–1000 Hz, which are characteristic of short rail irregularities.

In [3], the interaction of wheels with rails in the process of guiding wheel pairs along the track is considered. It is shown that significant interaction forces are a powerful source of vibrations transmitted to the running gear of the rolling stock. Passenger cars can also have additional sources of vibrations transmitted to the body from the drive and additional suspension equipment. In article [4], the level of vibrations, which depends on the technical condition of the gearboxes of passenger cars, is analyzed. The work conducted theoretical and experimental studies of the operation of gearboxes based on the analysis of the level and frequencies of vibration, on the basis of which a series of diagnostic features of the drive operation were proposed. The work did not consider the issue of reducing the level of vibrations transmitted to the body.

Article [5] contains the results of a study of the characteristics of the suspension of undercarriage equipment: transformers, traction converters, auxiliary converters, air compressors, etc. Due to the significant weight of the equipment and elastic suspension, it can be a source of powerful vibrations transmitted to the frame of the car. The conclusion is made about the need to take into account the influence of air temperature on the stiffness of rubber elements of the suspension of undercarriage equipment. It is determined that in the temperature range from −30 to +40°C the dynamic stiffness of rubber shock absorbers of the suspension of undercarriage equipment changes relative to normal conditions by −20% +50%.

The article [6] presents the results of a study of vibrations affecting the level of passenger comfort using the example of the Tehran-Andymeshk train (Iran). The vibration transmitted to passengers was measured in accordance with ISO 2631-1-1997. Mechanical vibration and shock: Evaluation of human exposure to whole-body vibration. The statistical relationship of 15 psychological and physiological disorders in passengers with age, gender, number of trips and the level of equivalent acceleration of the car was studied. The results did not reveal statistical relationships between symptoms of discomfort and age. A significant impact on the well-being of passengers of the equivalent acceleration of the car was confirmed. The paper does not present an analysis of the influence of the parameters of the vibration of the car floor on the level of passenger discomfort.

In the article [7] local resonances of vibrations on the floor of high-speed trains, which negatively affect the comfort of passengers, are studied. Vibration peaks in the frequency range of 30–35 Hz are detected. Vibration attenuation is proposed using several subframe dynamic vibration absorbers. The results of the research show that the proposed method can reduce vibration, but only for certain modes, excluding the entire range of frequencies of vertical vibrations, which does not correspond to the modes of movement of suburban electric trains.

A feature of suburban diesel and electric trains is that passengers sit on seats or even stand on the floor throughout the trip. In the paper [8] the results of experimental studies of vibrations of the floor of railway cars are presented. The main aim of the research was to solve the problem of numbness of passengers' legs caused by vibration of the floor of railway cars. The analysis of the vibration characteristics of the floor shows that the vibration transmitted to the floor is significantly amplified in the frequency range of 20–50 Hz. This coincides with the frequency range in which the lower limbs of a person are most sensitive, which leads to numbness of the legs. To further study the vibration mechanism of the floor, a refined finite element model of the car body, including the floor panels, was developed. The results show that due to the inappropriate design of the rigidity of the elastic supports of the body, the deformations of the floor above the center of the bogie are significantly amplified for several typical modes in the frequency range of 20–50 Hz. Confirmation that at the oscillation frequencies of the car body frame, close to the natural frequencies of the floor, local resonance of the floor occurs, which is the main cause of numbness of the legs of passengers. The paper proposes optimization of the dynamic rigidity of the elastic support of the body. Experimental verification showed that as a result of the optimization of the suspension, the transmission of vibration from the chassis of the car body to the floor in the frequency range of 20–50 Hz was significantly reduced, due to which the problem of numbness of the legs was solved. Unfortunately, the proposed approach to the problem of reducing the vibration levels of the car floor based on the optimization of the spring suspension contradicts the tasks of the dynamics of the car movement.

In the article [9], the processes of vibration propagation in the material of the car floor and the influence of its structure on the properties of vibration absorption or insulation are considered. The processes of damping and vibration propagation are highlighted as separate. The types of vibrations transmitted from road irregularities and other sources from the car floor to the human body are considered. The article presents some results of the study of material vibration and an analysis of the propagation of material vibration in the floor is performed. The dominant frequency band of floor vibration is determined – these are low frequencies, between 20 and 40 Hz. In this range, the natural frequencies of the floor material can be identified. The results obtained allow determining the frequency range of the vibrator in experimental studies.

The article [10] presents the results of the assessment of vibration perceived by passengers of the Cairo metro. It is emphasized that high levels of body vibration of passengers can cause diseases and health problems, especially low back pain. Vibrations can also cause damage to the musculoskeletal system of the neck and back. After daily exposure for several years, these same body vibrations can lead to a number of health disorders, including damage to internal organs, muscles, joints and bone structure. The measurement data were analyzed according to ISO 2631-1 and ISO 2631-5. ISO 2631-1 (1997) uses daily exposure A(8) and VDV, while the new methodology of ISO 2631-5 (2004) uses Sed and R-factor parameters. The analysis revealed that the whole-body vibration absorbed by the human body increases with the magnitude and duration of vibration exposure experienced by passengers. It was found that subway passengers have a higher risk of exposure than bus passengers. The main conclusion of this study is that musculoskeletal symptoms and disorders of the lower back, neck and upper extremities in subway passengers may be the result of prolonged exposure to vibration, especially for passengers who sit during the trip.

The article [11] presents the results of many years of research on the relationship between vibration perceived by a passenger in a train car and pain that occurs in the lower back of a person. Based on the analysis of a large number of publications on this topic, a direct relationship between vibration exposure and back pain has not been confirmed on a mass scale. However, it was observed in a certain proportion of passengers (25%).

The impact of vibrations on humans in vehicles is most clearly manifested in the work of drivers. The article [12] examines the causes of low back pain (LBP), which is a common phenomenon among professional bus drivers. Studies have shown a relationship between the impact of whole-body vibration (WBV) and low back pain (LBP). The differences in the WBV level are considered using the example of two bus designs. An intercity bus with a high floor and a city bus with a low floor were considered. Experiments were conducted on different roads, i.e. with different excitation characteristics. As a result, it was concluded that on average the driver's seat attenuates vibrations transmitted by the floor by only 10%. This conclusion is important in studies of the vibration-protective characteristics of the floor of a passenger car and passenger seats. In particular, it indicates the need to pay attention primarily to the absorbing properties of the floor.

In [13], some problems associated with comparing measured passenger vibration levels with the standard are outlined, and the need to revise the standard format is discussed. A method for generalizing the attenuation of vehicle seat vibration is proposed. Two criteria for assessing the threshold impact of vibrations on the passenger are considered: the "limit of reduced comfort" and the "limit of qualification reduced by fatigue". It is shown that the vibration isolation of most vehicle seats is not effective enough.

In [14] it is emphasized that vibrations of the body perceived by passengers mainly occur in the low frequency range – up to 50 Hz. According to the researchers, they arise due to geometric defects of the wheels and rails. Vibrations are transmitted to passengers through the suspension stages and, finally, reach the body, floor and seats. In addition, the coupling effects of the assembled system should be properly taken into account. The dynamics of two subsystems of the car are studied, namely the floating floor and the passenger seat. The test results demonstrate a strong influence of the coupling between the floor and the seat on the frequency response of the vibrating unit. The methodology proposed in the article, which is based on both laboratory tests and numerical models, is a promising approach to the design and optimization of railway transport subsystems. The results obtained open up the prospect of improving the vibration protection of passengers based on vibration-isolating inserts between the floor and the seat.

A comprehensive model of the track-train-seat-passenger vibration system for lateral, vertical and roll vibrations is considered in [15]. The comfort characteristics of a high-speed train were studied. Ride comfort was assessed by the total equivalent acceleration determined according to ISO 2631-1. It was found that the flexibility of the rail bed has a negligible effect on floor vibrations. The flexibility of the body caused a sharp vibration of the floor in the lateral, vertical and tilt directions at frequencies above 7.5 Hz. Comfort indicators were the worst near the ends of the body. The vertical acceleration on the seats did not depend on the position of the seat.

The work [16] is devoted to the search for lightweight, laminated vibration-absorbing materials for the floors of railway cars. In particular, the possibility of using bamboo for floor elements is considered. Its important properties are emphasized: vibration-proof, noise-proof, fire resistance, strength. The developed composite material with bamboo inclusions combines the unique advantages of laminated construction and composite interface bionics. The low density (0.73 g/cm^3) of the layered composite provides a specific modulus of elasticity of 13.03 GPa cm^3/g, a vibration damping coefficient of 6.61% and an impact strength of 14.16 J/cm^2. This is significantly higher than that of other wood-based composite materials used for the floors of high-speed train cars. This bamboo-wood composite material has great potential for use in the arrangement of floors of environmentally friendly railway cars.

The paper [17] presents the results of field tests of the dynamic characteristics of low-floor railway cars. The tests revealed the presence of predominant vertical vibrations at a frequency of about 8 Hz. It was proven that these vibrations are the result of the pitching motion of the cars. According to the results of theoretical analysis, it was found that the cause of the pitching motion with a frequency of 8 Hz is the elastic characteristics of the articulated connection between the body and the bogies.

In the article [18], the results of experimental studies of the influence of viscoelastic damping materials on vibration reduction in railway cars were published. In particular, the paper emphasizes that the problem of noise and vibration has become particularly acute for high-speed train traffic. To reduce vibration and noise in railway vehicles, three types of viscoelastic damping materials were proposed: bitumen-based damping material; water-based damping coating and butyl rubber damping material. Measurements of vibration and noise levels were carried out on experimental cars, finished with new materials in various structural modifications. It was found that the effect of vibration reduction significantly depends on the speed of movement. One of the results suggests that the first two materials reduce vibration in a wider frequency range of 63–1000 Hz than the last damping material. They also reduce the level of internal noise in the car by 5–8 dBA, and the latter by 1–6 dBA.

Many studies claim that the main source of car vibrations is dynamic processes in the contacts of wheel pairs with the track. In particular, work [19] presents the results of the analysis of vibrations caused by irregularities in the railway track. The focus is on the mechanism of vibration excitation of the car body and its impact on passenger comfort. The hypothesis of the leading role of wheel pair vibrations in the vibration excitation of the car floor is confirmed based on the results of a numerical experiment on a multi-body model and experimental test data.

Works [6–10, 13, 15] contain important data on “dangerous” vibration frequencies for the human body, which are undesirable to allow in passenger cars. Therefore, before making a decision on floor modernization, in order to choose the most rational variant of its structure, it is necessary to conduct comprehensive theoretical and experimental studies. One of the directions of such studies is obtaining resonant frequency characteristics of the floor. This article is devoted to the first stage of resonant studies – on compact floor samples of different structures.

The aim and objectives of research. The aim of research is an experimental and theoretical comparative assessment of the resonant frequencies of the floor of suburban electric train cars on samples of different structural structures.

To achieve the aim, the following objectives were set:

- justification of the parameters of experimental samples of the floor of passenger cars of different structures;
- experimental studies of the resonant frequencies of the floor of different structures.

Materials and methods of research. The object of research is the resonant frequency characteristics of the floor of a suburban electric train car.

The main hypothesis of research: the introduction of rubber shock-absorbing elements into the structure of the floor of the car significantly affects its resonant frequency characteristics.

The study was carried out on experimental full-scale samples of the floor of passenger cars with different structural schemes. To obtain the frequency-amplitude characteristics (FAC) of the floor, an experimental complex was used, which consists of the following elements:

– vibration source – portable vibrator ВСВ -131/2 (produced by LLC NVF "Promvitech", Ukraine);

– UM 3-050 accelerometer (manufactured by the “SRDTI” branch of JSC “Ukrzaliznytsia”, Ukraine);

– equipment for recording the results of the experiment – analog-to-digital converter NI 9237 (manufactured by National instruments, USA);

– certified software for conducting research and processing results – LabVIEW (developed by National Instruments, USA).

In general, the model of the vibration protection system can be represented as three components: a source of vibrations; vibration isolation; object of vibration protection. The source of vibrations of a passenger car is the base (bottom) of the car, which is under the action of excitations from bogies and attachments. Vibrations are transmitted to the base of the car through its frame. Vibration isolation is a multi-component layered floor of a car, which includes elastic and dissipative elements, the function of which is to absorb vibration, and rigid power structural elements, the function of which is to support the load from passengers, seats, partitions and internal equipment of the car. The object of vibration protection is passenger seats installed on the surface of the car floor.

The reaction perceived by the car floor from the vibration isolator side can be considered proportional to the deformation Δ and deformation rate $\frac{d}{dt}\Delta$ of the vibration isolator

$$R = C \cdot \Delta + \beta \cdot \frac{d}{dt}\Delta, \tag{1}$$

where C, β – respectively, the stiffness coefficient (stiffness) and the damping coefficient of the linear inertialess vibration isolator.

The stiffness coefficient of the vibration isolator C and the reduced mass of the vibration system m determine its natural frequency of oscillations ω_o

$$\omega_o = \sqrt{\frac{C}{m}}. \tag{2}$$

The damping properties of the vibration isolator are characterized by the vibration damping coefficient δ

$$\delta = \frac{\beta}{2 \cdot m}, \tag{3}$$

According to the theory of resonance [20], the efficiency of the vibration isolator during force harmonic excitation can be estimated by the coefficient of reduction of the force amplitude k, which is transmitted to the vibration protection object

$$k = \frac{F}{F_o} = \frac{\sqrt{\omega_o^2 + 4 \cdot \delta^2 \cdot \omega^2}}{\sqrt{(\omega_o^2 - \omega^2)^2 + 4 \cdot \delta^2 \cdot \omega^2}}, \tag{4}$$

where F_o – the amplitude of force excitation according to the law $F_o \cdot sin(\omega \cdot t)$;
ω – the frequency of force excitation.

Fig. 2 shows typical amplitude-frequency characteristics of the vibration system for several values of the vibration damping coefficient δ.

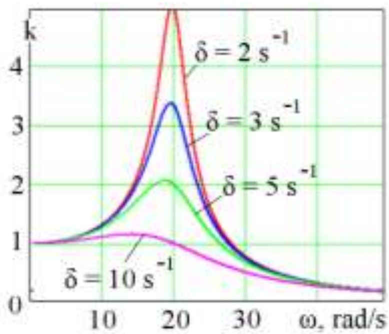


Fig. 2. **Typical amplitude-frequency characteristics of an oscillatory system for different values of the oscillation damping coefficient**

An experimental and computational method for determining the resonant frequency of an oscillatory system based on limited experimental data, namely the excitation frequency and the measured force amplitude reduction coefficient λ, is proposed.

To do this, it is necessary to solve equation (4) with respect to ω_o:

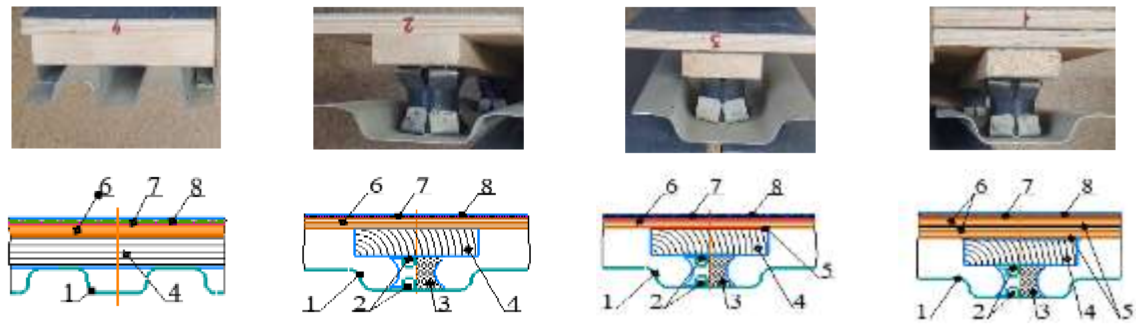
$$k(\omega_o) = \frac{\sqrt{\omega_o^2 + 4\cdot\delta^2\cdot\omega^2}}{\sqrt{(\omega_o^2-\omega^2)^2 + 4\cdot\delta^2\cdot\omega^2}}. \qquad (5)$$

At this oscillation damping coefficient δ, cyclic excitation frequency ω and force amplitude reduction coefficient k are considered known.

This method is important for cases when the experiment has the possibility of creating external excitation only in a limited frequency range, or for fixed frequencies. It is such equipment that was used in this study. To solve equation (5), the Given-Find procedure of the MathCAD computing package (Parametric Technology Corporation – USA) was used.

Justification of the parameters of experimental samples of the floor of passenger cars of different structures. In the process of forming a project for the deep modernization of electric trains at the "KECRP" the task was set of choosing a structural floor structure that would best meet the criteria of comfort and smoothness of movement (SOU MPP 45.060-204:2007).

To conduct experimental studies of the vibration-proof properties of the floor of the modernized cars, four variants of the structure used on passenger cars and electric trains of Ukrzaliznytsia were selected. Fig. 2 shows photos and diagrams of the structural structure of the floor samples selected for the experiments.



a *b* *c* *d*

Fig. 2. **Structural structure of the floor of the cars under study.**
a – Ammerndorf car; b – KCBP car with rubber shock absorbers; c – modernized floor of car 61-776 with vibration-damping membrane; d – modernized floor of car 61-776 with two layers of vibration-damping membrane and plywood flooring: 1 – floor base; 2 – fasteners; 3 – rubber shock absorber; 4 – floor beam; 5 – vibration-damping membrane; 6 – floor covering; 7 – key layer; 8 – floor covering

Floor option a – Ammerndorf car – model 47-K (GDR). Ammerndorf cars have long been the most popular passenger cars on Ukrainian railways and have extensive experience in operation. Each of the other three floor samples is a certain step towards improving the vibration isolation structure: rubber shock absorbers – a layer of vibration damping membrane – an additional solid layer of vibration damping membrane. Floor option b – a car built by PJSC "Kriukiv Car Building Plant" (KCBP) (Ukraine) model 61-776 using a system of rubber shock absorbers. Floor option c – a modernized floor based on the 61-776 car model with the addition of a layer of vibration damping membrane. Floor variant d is a further modernized floor of the 61-776 model car with an additional solid layer of vibration damping membrane and a second layer of plywood flooring.

Table 1 shows the structural structure of the floor samples studied.

Table 1. **Structural structure of experimental floor samples**

Floor structure	Floor element material	Presence of an element on the floor option			
		a	*b*	*c*	*d*
Floor base	Profile sheet	+	+	+	+
Rubber shock absorber mount	Sheet B 1.5	-	+	+	+
Rubber shock absorber	Rubber B-14 (vulcanization)	-	+	+	+
Floor joist	Pine board – 50 mm	+	+	+	+
Vibration damping membrane-first layer	Vibrofix ML, 5kg/m^2	-	-	+	+
Floor sheet-first layer	Plywood PFA-T 20mm	+	+	+	+
Vibration damping membrane-second layer	Vibrofix ML, 5kg/m^2	-	-	-	+
Floor sheet-second layer	Plywood PFA-T 20mm	-	-	-	+
Linoleum adhesive	Ke 2000S glue	+	+	+	+
Floor covering	PVC (linoleum)	+	+	+	-
	PVC, 2TEC (Belgium)	-	-	-	+

Research objectives: to determine the impact of improving the structural structure of the car floor on its vibration-protective properties, for example, a vibration-damping membrane; rubber shock absorbers; the presence of a second layer of plywood flooring, etc.

Experimental studies of vibration isolation of floor options of different designs. The main element of the experimental complex for studying the vibration isolation of the floor of experimental cars is the source of exciting vibration – the BCB-131/2 vibrator ("Promvitech", Ukraine). The vibrator has the following characteristics:

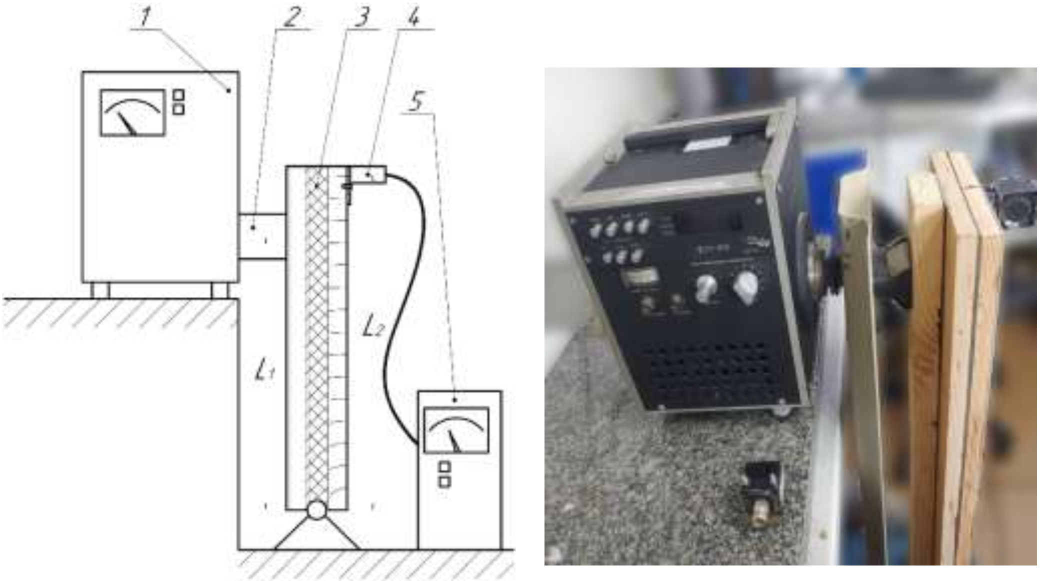
- the direction of vibration creation – horizontal;
- the range of acceleration values – 1...20 m/s^2;
- the range of vibration velocity values – 0.5...70.7 mm/s;
- vibration frequencies – fixed – three options – 45; 64; 79.6 Hz;
- the limit of permissible amplitude error – 3%;
- the limit of permissible frequency error – 0.2%.

Vibration sensors – UM 3-050 accelerometers (manufactured by the branch of "SRDTI" of JSC "Ukrzaliznytsia", Ukraine). The accelerometer is built on the ADXL 278 Analog Devices chip (USA). The collection and initial processing of experimental data is carried out by the CompactRIO controller of National Instruments (USA).

To measure the vibration parameters, convert signals and record the experimental results, a system of control running tests of rolling stock units (CRT) was used. CRT was developed at the Regional Branch of JSC "Ukrzaliznytsia" "Scientific Research and Design and Technological Institute of Railway Transport" ("SRDTIUZ").

Processing of measurement signals and recording of results - analog-to-digital converter NI 9237 (manufactured by National Instruments, USA) based on the LabVIEW software of the National Instruments company.

The equivalent loading scheme allows for the creation of artificial vibrations of experimental samples and their registration. Fig. 3 shows the loading scheme and the experimental complex.



a *b*

Fig. 3. **Equivalent load scheme (a) and experimental complex for studying the parameters of floor vibration isolation (b):**
1 – vibrator; 2 – movable vibrator rod; 3 – floor test sample; 4 – accelerometer; 5 – equipment for recording results

Research results. In accordance with the technical capabilities of the BCB-131/2 vibrator, the test was carried out at three fixed excitation frequencies – 45; 64; 79.6 Hz, with three amplitudes of vibrator excitation accelerations: 1.0; 1.5; 2.5 m/s^2.

As a result of the tests, the frequency characteristics of the relative change in the amplitude of accelerations on the floor surface of different structures were obtained. Relative amplitudes are presented as the acceleration amplitude growth factor k, as the ratio of the amplitude at a certain frequency to the amplitude at a frequency of 45 Hz, which was taken as the base (Fig. 4).

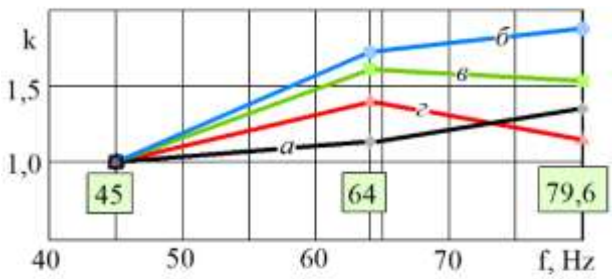


Fig 4. **Frequency characteristics of the relative amplitude of accelerations on the floor surface k for different structures:** a, b, c, d – variants of the structural structure of the floor

In addition, the following results were obtained.

For all investigated variants of the structural structure of the floor, the coefficient of growth of the acceleration amplitude (*k*) does not depend on the excitation amplitude. For all variants of the excitation amplitude k has close values and depends only on the frequency.

To calculate the resonant frequencies of the floor of the car f_r, the proposed experimental and computational method for determining the resonant frequency based on limited experimental data was used, namely, by the excitation frequency and the measured coefficient of reduction of the excitation force amplitude (5). The results obtained are presented in Fig. 5.

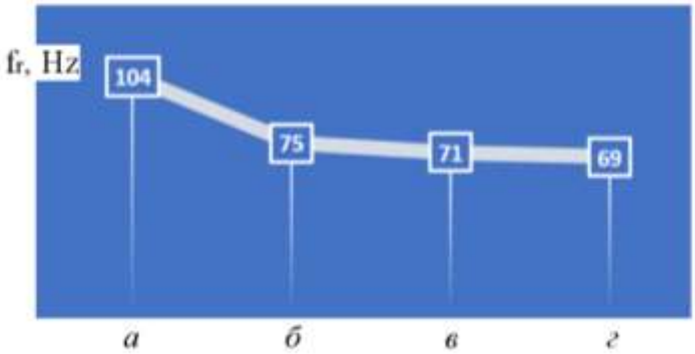


Fig 5. **Calculated values of the resonant frequency for floor samples of different structures**: a, b, c, d – variants of the structural floor structure

The frequency characteristics of the floor design variants with rubber shock absorbers – variants b, c, d – differ significantly from the basic variant a without rubber shock absorbers. In this case, the influence of rubber shock absorbers is manifested in a decrease in resonant frequencies from 104 Hz (floor without rubber shock absorbers) to 69–75 Hz (floor with rubber shock absorbers). The influence of the vibration damping membrane also manifested itself, namely, a shift of the resonant frequency towards low frequencies is observed. Thus, for variant b (without vibration damping membrane), the resonant frequency is 75 Hz, and for variants c and d – 71 and 69 Hz, respectively. The introduction of an additional solid layer of the anti-vibration membrane (variant d) slightly shifts the resonant frequency towards low frequencies compared to variant c.

Conclusions.

1. The parameters of experimental samples of the floors of passenger cars of different structural structures were substantiated for the experimental assessment of their vibration-protective properties. Four floor options were selected. The Ammerndorf (47-K) car was chosen as the basis for comparison, as it was the most popular passenger car on the railways of Ukraine for a long time and has extensive operating experience. Each of the other three cars is a certain step, a significant expansion of the means of improving vibration insulation: rubber shock absorbers – a layer of vibration-damping membrane – an additional solid layer of vibration-damping membrane.

2. An assessment of resonance zones for floors of different structural structures was carried out. The frequency characteristics of the relative change in the amplitude of accelerations on the surface of the floor of different structures were obtained. The relative amplitudes are presented in the form of a coefficient of growth of the acceleration amplitudes k, as the ratio of the amplitude at a certain frequency to the amplitude at a frequency of 45 Hz, which was taken as the base. It was found that the coefficient of increase in the amplitude of floor accelerations does not depend on the excitation amplitude and depends only on the frequency.

3. An experimental and computational method for determining the resonant frequency of the oscillatory system is proposed based on limited experimental data, namely the excitation frequency and the measured coefficient of reduction of the excitation force amplitude.

The frequency characteristics of the floor design options with rubber shock absorbers differ significantly from the basic version in terms of resonant frequency – 104 Hz versus 69–75 Hz.

The introduction of layers of a vibration damping membrane into the design scheme has a negligible effect on the resonant frequency of the floor: 75–71–69 Hz.

General conclusion: experimental studies are necessary on a full-scale car sample in a wide range of excitation frequencies to obtain more reliable characteristics of the influence of the passenger car floor structure on its resonant frequencies.

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Порівняльна оцінка резонансних частот підлоги вагонів приміських електропоїздів

Робота присвячена дослідженню віброізоляційних властивостей підлоги пасажирських вагонів з метою вибору конструктивної структури при модернізації електропоїздів приміської залізниці серій ЕР2, ЕР9, ЕПЛ2, ЕПЛ9, ЕД9, яку проводить ПрАТ «Київський електровагоноремонтний завод». На основі аналізу відомих досліджень вібрацій підлоги вагонів вибрано характеристики вібратора збудження механічних коливань – частоти і амплітуди прискорень. Для проведення експериментів прийнято горизонтальну схему навантаження. Метою дослідження є експериментальна порівняльна оцінка вібрprincipal захисних властивостей підлоги вагона з різними варіантами конструктивної структури. Були поставлені задачі дослідження: обґрунтування параметрів експериментальних зразків підлоги пасажирських вагонів різної конструктивної структури; проведення експериментальної оцінки резонансних зон підлоги пасажирського вагона різної конструктивної структури. Запропоновано експериментально-розрахунковий метод визначення резонансної частоти коливальної системи на основі обмежених експериментальних даних, а саме частоти збудження і виміряного коефіцієнта зменшення амплітуди сили. На основі випробувань отримано частотні характеристики відносної зміни амплітуди прискорень на поверхні підлоги різної структури. Відносні амплітуди представлено у вигляді коефіцієнта зростання амплітуд прискорень k, як відношення амплітуди на певній частоті до амплітуди на частоті 45 Hz, яку було прийнято як базову. З'ясовано, що коефіцієнт зростання амплітуди прискорень підлоги не залежить від амплітуди збудження і залежить тільки від частоти. Частотні характеристики варіантів конструкції підлоги, що мають гумові амортизатори суттєво відрізняються від базового варіанту і сприяють появі ознак резонансу у зоні відносно низьких частот в діапазоні 55–70 Hz. Ведення в конструктивну схему шарів вібродемпферної мембрани також сприяє зсуву максимальних коефіцієнтів зростання амплітуд у бік низьких частот. Але цей вплив є вкрай незначним.

Ключові слова: *залізничний вагон, комфорт пасажирів, віброізоляція підлоги, рівні вібрацій, експериментальні дослідження.*