

Bogdan Diveev

Lviv Polytechnic National University
12, Stepan Bandery Str. Lviv, 79000, Ukraine

© B. Diveev, 2020

<https://doi.org/10.23939/tt2020.02.065>

VIBRATION IMPACT AND NOISE PROTECTION DEVICES WITH DVA FOR WHEELED VEHICLES

Summary. *The article investigates vibration and noise protection devices for wheeled vehicles using dynamic vibration absorbers (DVA). Algorithms for modeling their dynamic characteristics based on adaptive calculation schemes are presented. A non-linear suspension with DVA and a noise-absorbing partition is considered, which is due to the introduction of a layered composite thin-walled structure with an intermediate damping layer with high damping properties and a DVA system, which provides better vibration and noise absorption. The problems of shock propagation during the overturn of the bus to passengers are also considered. The influence of the parameters of the shock absorber on the dynamic properties of the bus is investigated. The optimal parameters of the shock absorber are determined.*

Key words: *suspension, wheeled car, vibration, dynamic vibration absorber, noise barrier, bus overturn, shock absorber, optimization.*

1. INTRODUCTION

One of the important problems of designing modern vehicles, including wheeled vehicles, is the safety and comfort of the driver and passengers while maintaining the optimum technical and economic parameters, such as functionality, economy, energy and material consumption, operating costs, repair costs, and more. The primary objective in this area is the need to improve the analytical method of calculating the action of dynamic loads to approximate theoretical results to the experimental data and to achieve the rational and effective design of safety and comfort elements.

A variety of suspension designs with permanent or adjustable mechanical properties are used in transport technology. The use of suspensions with variable mechanical properties is explained by the following factors: the change in the load that falls on this wheel and the change in operating road conditions. Regulators are known for taking into account the change in weight in the air suspension of buses, rear suspension of modern cars (changing the number of passengers in the back seat). However, they are quite complex to implement and involve the use of electronic and electromechanical devices (controllers, regulators). In the meantime, it is important to develop simple and efficient suspensions for wheeled trailers, agricultural machinery, which would take into account the change in their weight. An important issue in the development of modern machines is the reduction of vibration. Dynamic vibration dampers (DVA) are widely used to reduce vibration and noise levels in vehicle cabs, to reduce the vibration of rotating vehicles, including unsprung wheels of wheeled vehicles.

Noise barriers are widely used in vehicle designs. They can be used to reduce vibration and noise levels in vehicle interiors. The sources of these perturbations can be both the process of interaction of wheels with the road and internal sources: engine, transmission, gearbox, brakes, mufflers, pumps, etc. A particularly large contribution to the acoustic load of the bus interior is played by the engine compartment. Dynamic vibration dampers (DVA) attached to a thin-walled layered element help reduce noise in the low-frequency region, where noise-insulating partitions are ineffective.

The degree of damage to the vehicle and the injury of the occupants in the crash depends on the ability of the body structure to absorb maximum kinetic energy while maintaining the integrity of the passenger compartment. To ensure the structural integrity of the vehicle and its ability to absorb impact energy, it is important to study the nature of the fracture to reduce peak forces and improve the impact absorption energy. These factors can be improved with special energy-absorbing devices.

2. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

An important issue in the development of modern machines is the reduction of vibration. DVA is widely used to reduce vibration and noise levels in vehicle cabs to reduce the vibration of rotating vehicles, including the unsprung mass of wheeled vehicles. The car suspension with DVA is considered. To reduce the vibrations of the unsprung mass of the car, we use the DVA attached to them.

In these cases, the use of DVA can be effective (the basics of DVA design were developed at the beginning of the last century in the writings of Dan Hartog, S. Timoshenko). Many DVA designs are currently being developed. They are used in different machines and structures; differ in weight and dimensions, design features, application ranges. An example of the different locations of DVA is analyzed in [1–9].

Noise reduction is also important. Devices consisting of a thin-walled element – the basis of the partition and the connected dynamic vibration dampers, which serve for vibration noise protection from the perturbations acting on the partition, are considered. Such a partition can be used in the floor of the bus, in the elements of the engine compartment of vehicles, and the walls of the cabin, which is under conditions of high acoustic loads, as well as the cabs of mining machinery, in the schemes of sound insulation of premises, aircraft salons, etc. [10–15].

Rolling over a bus is one of the most serious accidents. The risk of fatal consequences when tipping is greater than in other cases. It has been determined that although the share of bus tipping accidents is lower, the fatal consequences (31 %) and serious injuries (21 %) are higher than other accidents [16].

Since then, the danger of bus overturning has only increased. After all, the speed and, especially, the height of the buses have increased, which helps to increase the impact force when turning. The threat to humans in this type of accident is even like a plane crash. In the case of a bus, a significant reduction of the threat is possible with the help of shock-proof devices (although this type is offered even for aircraft). This paper explores a new type of energy-absorbing device.

Known various designs of impact devices for wheeled vehicles, devices whose principle of operation is the plastic deformation of materials during which the shock energy is absorbed due to structure displacement [17, 18] are the most widely used. Composite materials are widely used [16]. In recent times, functional-gradient materials have been used as energy-generators [19].

In an emergency, the impact energy is absorbed by the plastic deformation of the body elements during a collision of the vehicle with an obstacle. Such elements are specifically provided in the design and up to a certain level of load play the role of the frame of the machine. As the load increases, the above threshold value of the energy-absorbing elements begins to work as a shock device.

However, most of these devices work by flattening all sorts of beams. This does not allow the shock energy to be absorbed evenly. They do not allow them to absorb the impact energy in the side-turn of the bus, which is often accompanied by massive injury to passengers. An example of such a shock-absorbing structure is the shock-absorbing truss for the car seat (patent WO 01/40015 A1 of 07.06.2001), which contains energy-absorbing elements – rods. However, this device does not allow the use of angular inertial properties of the rods, which could significantly improve the energy-absorbing properties of the farm, nor does it provide for the regulation of effective shock-absorbing mass. In this case, the farm does not perform the functions of the supporting structure (at loads less than critical), and all the impact energy is absorbed only in one small element, the energy absorption efficiency is low. In [20] the design of a shock-absorbing roof of a bus is described.

3. VIBRATION PROTECTION DEVICES IN SUSPENSION USING DVA

The algorithms for calculation and optimization of a non-linear wheeled vehicle and DVA are considered. The work consists of three stages: the first is to obtain a general design scheme, the second is to determine the mechanical characteristics of the suspension, taking into account the change in static load on it and its nonlinear elastic-damping properties, to obtain simple analytical approximation expressions for the dynamic rigidity of the suspension and – optimization of trailer and suspension frames based on the first two items of the models.

Class of passive car suspension is known, consisting of elastic and damping elements, and is used for vibration protection against disturbances acting on wheels from the road surface. To increase the vibration protection properties of the suspension in their designs, rubber gaskets are provided in the places of the hinge fastening of the elastic element (springs, cylindrical springs) and dampers. Such elastic inserts are used both in the McPherson suspension and in the suspension on the longitudinal levers. However, along with the positive impact on the level of vibration protection in the higher frequencies, such elastic inserts adversely affect the steering of the car. Therefore, in real suspension designs, additional stabilizers, Panar beams are used to increase controllability, which increases handling but reduces the pliability of the suspension. Contact with the road and the force of the wheel press against the road is increasing, but at the same time, the vibration protection is reduced. There are some ways to improve the contact of wheels with the road. This is, for example, the use of additional elastic elements in the spring suspension. But they again increase the rigidity of the suspension, which is unacceptable for passenger cars. It is known that unsprung masses have some natural frequency of oscillation, which is much higher than the frequency of oscillation of the body and is 10-15Hz. When moving the car, unsprung masses are excited and the following undesirable phenomena occur:

- hum in the car at this frequency;
- periodic reduction of the force of the wheel to the road.

In recent years, some companies, including OPEL (Wheel suspension for vehicle roll control – has separately attached springs and shock absorbers with oscillation dumping mounting with the carrier, DE4121827, 1993-01-07) and Volkswagen (Vehicle axis vibration dumping system A new class of suspension – suspension with intermediate, attached to the unsprung masses with additional vibration-absorbing mass, is proposed for the damp wheel-specific vibrations, DE19915635, 2000-10-12). This design serves to reduce the vibrations of the unsprung masses and, thus, to reduce these negative phenomena. However, in addition to the positive impact, such constructive solutions have negative features. Namely, the use of a vibration-absorbing mass or a dynamic oscillator (DVA) is associated with phenomena such as the occurrence of two parasitic resonant frequencies in the immediate vicinity of the operating frequency of the DVA. The second drawback is the gradual change in the frequency response of DVA due to the fatigue of the material of the joints, microproducts, heating during operation, and other factors. These factors require not only precision manufacturing of DVA, but also constant monitoring of its dynamic properties, which significantly increases its cost, and in the case of a car suspension – is often a difficult task. Therefore, unlike the above design schemes (from OPEL and Volkswagen), the design is based on a new mechanical system.

The design solves the problem of ensuring a smooth control of the frequency and damping characteristics of the suspension with DVA and the implementation of effective volumetric vibration. This task is achieved in that in the suspension, equipped with attached to unsprung masses compact dynamic vibration dampers containing vibration-absorbing inertial element, made in the form of a curved rod, attached to a depreciated object, with attached, which is connected with taking into account the possibility of changing the shape, geometrical parameters and mechanical properties of the rod, the shape of the container and the nature of its filling to make a precise adjustment adjustable of this dynamic oscillator for a given operating frequency while ensuring its optimal volumetric damping properties.

In Fig. 1, a suspension with the DVA is shown, where: 1 is a tubular suspension element; 2 – clamping clips, 3 – clamping nodes; 4 – nodes of clamping rods DVA, 5 – curvilinear rod – elastic

element; 6 – inertial mass in the form of a hollow container with additional masses. In Fig. 2, the design scheme of the suspension is shown.

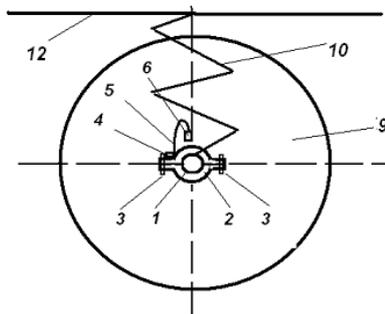


Fig. 1. Car suspension with DVA

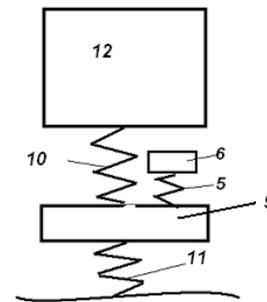


Fig. 2. The design scheme of the suspension

Fig. 3a shows in more detail: 1 – tubular suspension element; 2 – clamping clips, 3 – clamping nodes; 4 – nodes of clamping rods DVA, 5 – curvilinear rod – elastic element; 6 – inertial mass in the form of a hollow container with additional masses. In Fig. 3b, it is shown: 6 – inertial mass; 7 – hollow container, 8 – additional masses.

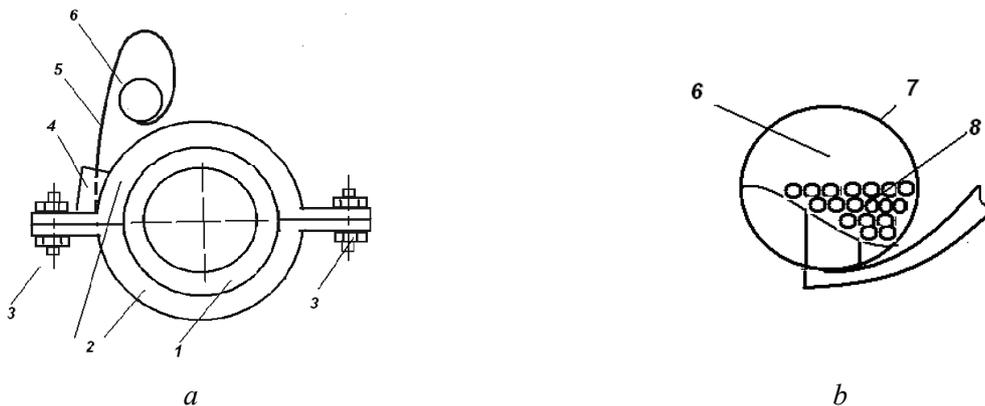


Fig. 3. More detailed image of the suspension bracket (a) and container with particles (b)

To the depreciated object 12 – the tubular element 1 of the suspension is attached to the clamp 2 using the clamping nodes 3. Through the clamping nodes 4 rod DVA, attached elastic element DVA – curved rod 5. To the rod 5 is attached to the container and extra masses.

DVA works as follows. The oscillations of the depreciated object (the frame of the car 12) and the tubular element 1 of the suspension cause the displacement of the clamps 3, which are transferred to the inertial mass 6 in the form of a hollow container with additional mass, which causes them to oscillate.

Adjustment of the operating frequency of the DVA to the resonant frequency of oscillation of the depreciated object 12 is carried out by selecting the shape, geometrical parameters, and mechanical properties of the rod 5 and mass 8. This allows for smooth regulation of the volume damping properties of the DVA. Thus, the developed DVA provides efficient, frequency-controlled, volumetric dynamic damping (damping) of the vibrations of the unsprung mass of the vehicle. This reduces the vibration and noise in the car.

Changing the number and mass of the balls 8, freely mounted in the hollow containers 7 of the inertial mass 6 (Fig. 3.4), provide the corresponding values of damping in the vicinity of a given operating frequency. This changes the operational properties of the integral elastic-damping system and provides precise tuning of the parameters of vibration absorption of DVA, which allows solving two main problems:

- to provide highly efficient absorption of energy of mechanical vibrations of the system;
- to prevent the occurrence of resonant oscillations in the area of the calculated frequencies, which are selected around the frequencies of the most intense oscillations of the rotor machine.

The procedure for calculating the dynamics of a container with balls is given in [6-8]. Dynamic equilibrium balloons in a container are reduced to a single mass model, the parameters of which depend on the geometry of the container, the degree of its filling, and the material of the balloons. These parameters are derived from experimental data.

4. CALCULATION AND OPTIMIZATION OF NON-LINEAR TRAILER SUSPENSION WITH DVA

Fig. 4 indicated: 1 – frame; 2 – elastic sleeve with a rubber element-plate; 3 – node attachment of the spring to the frame; 4 – profiled spring; 5 – spring with the variable free part; 6 – dry friction node with an intermediate layer with a high coefficient of friction and relatively low coefficient of friction. To increase the damping, element friction 6 is additionally introduced.

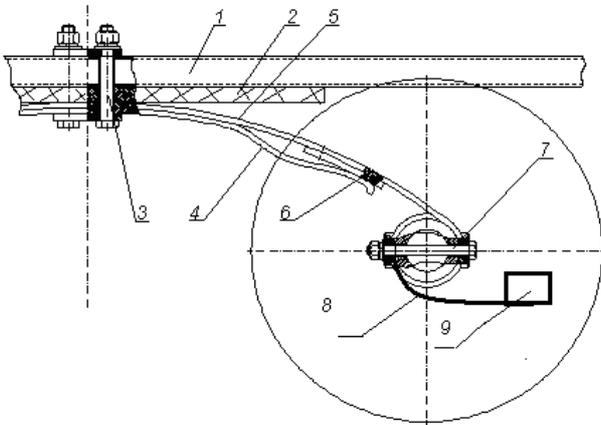


Fig. 4. Design of non-linear trailer suspension with DVA

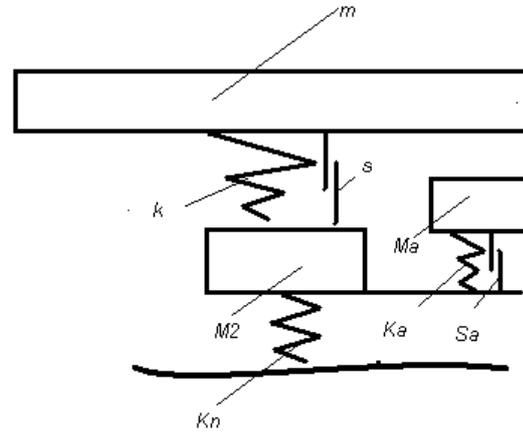


Fig. 5. The three-mass design scheme of suspension with DVA

Let's consider a three-mass design scheme of suspension with DVA (Fig. 5). Here, m is the mass of the part of the machine corresponding to that part of the suspension, M_2 is the corresponding mass of the unsprung parts and M_A is the mass of the DVA. K, S is the corresponding elastic and damping element. Dynamic equilibrium equations will be

$$\begin{aligned}
 m \ddot{w}_0 + S \left(\dot{w}_0 - \dot{w}_1 \right) + K (w_0 - w_1) = 0, \quad m_A \ddot{w}_A + s_A \left(\dot{w}_A - \dot{w}_1 \right) + k_A (w_A - w_1) = 0 \\
 M_2 \ddot{w}_1 - S \left(\dot{w}_0 - \dot{w}_1 \right) - K (w_0 - w_1) - s_A \dot{w}_A - k_A w_A = K_n (w_n(t) - w_1),
 \end{aligned}
 \tag{1}$$

where S, K – some nonlinear functions of relative velocities and displacements, K_n – stiffness of wheels, $w_n(t)$ – kinematic perturbation (in general, some random function depending on both terrain and velocity).

Compound structures determine their functional and durable properties and require refined calculation. For units such as rubber-metal shock absorbers and plate vibration absorbers, it is important not only to determine their dynamic stiffness but also to the stress fields that determine their durability. To adequately determine these quantities, it is necessary to apply refined computational schemes that allow studying the physic-mechanical fields in inhomogeneous structures with a priori unknown contact surfaces and nonlinear properties of materials. These problems were considered in both simplified and refined [2-8] formulation.

In Fig. 6 the values of the beam stiffness for the different ratio of the bed coefficient of the rubber layer and the bending rigidity of the beam k/EI are presented (the shape of the stamp was chosen in the form of a parabola $f=x^2/2R$).

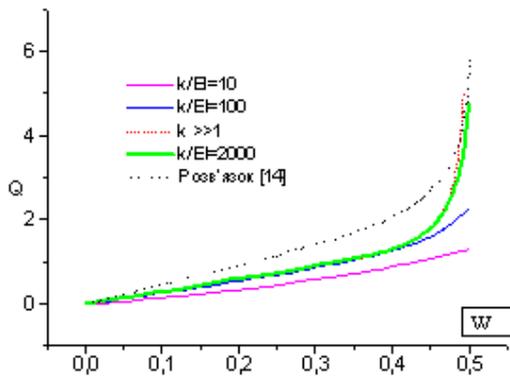


Fig. 6. Beam stiffness for the different ratio of the bed coefficient

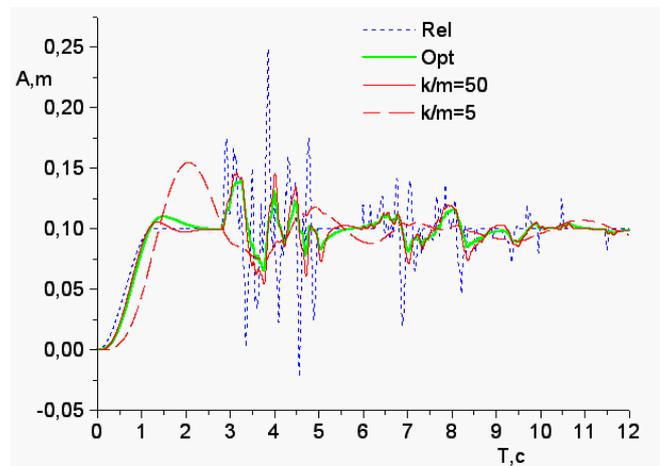


Fig. 7. Examples (including the optimal one) of implementing a dynamic non-stationary random process

Unlike traditional suspension in the frequency domain, the real dynamic process is considered, taking into account all the nonlinearities that undoubtedly take place in real structures. Advanced numerical methods have been used to analyze such processes – the Gear solution method and the small-parameter computational schemes obtained using physical condensation methods [5–7]. Genetic methods were used for optimization. Fig. 7 gives examples (including the optimal one) of implementing a dynamic non-stationary random process for kinematic perturbation with an initial smooth monotone transient process (which characterizes controllability).

5. NOISE BARRIER WITH DVA

The design refers to mechanical engineering, in particular to the class of passive vibration protection devices, consisting of a thin-walled element – the basis of the partition and the associated dynamic vibration dampers, which serve for vibration protection against perturbations acting on the partition. Such a partition can be used in the floor of the bus, in the elements of the engine compartment of vehicles, and the walls of the cabin, which are under conditions of high acoustic loads, as well as cabs of machines of mining equipment, in the schemes of sound insulation of premises, cabin salons and the like.

The noise-absorbing properties of thin-walled structures are considered without taking into account the possibility of improving the characteristics of DVA. However, the thin-walled layered noise absorbers used today are only effective in the higher frequency range and ineffective in the low-frequency range characteristic of many machines. According to this utility model, their noise-absorbing properties can be significantly improved in the low-frequency range by DVA.

The basis of the development is to create a noise-absorbing partition for the engine compartment, which would be due to the introduction of a layered composite thin-walled structure with an intermediate damping layer with high damping properties and a system of dynamic dampers with the device of frequency control would be better noise absorption in vehicles.

This object is achieved by the fact that in the known noise-absorbing partition for the engine compartment containing a layered thin-walled structure, additionally contains adjustable precision dynamic vibration dampers, with high damping with the possibility of compatible with thin-walled elements of vibration.

The additional introduction into the thin-walled partition of the intermediate damping layer and the system of dynamic dampers with given damping will provide an expansion of the spectrum of vibration absorption of DVA and stability of the sound-protective properties of the partition, eliminates the danger of near resonance excitation of oscillations.

The reduction of vibration and internal noise of vehicles is considered, in particular the use of sound insulation partitions that separate the engine compartment from the passenger compartment (Fig. 8). The design scheme of the partition is shown in Fig. 9.

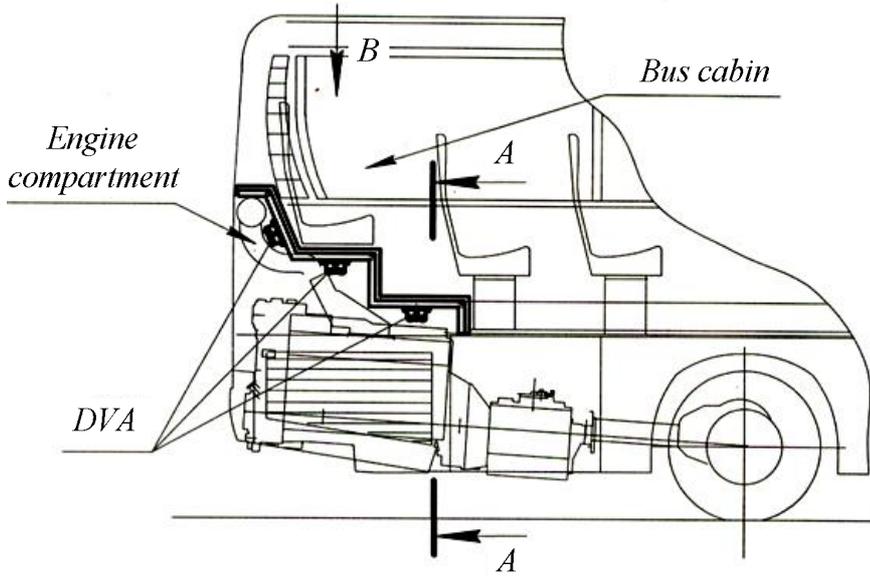


Fig. 8. Sound insulation partitions

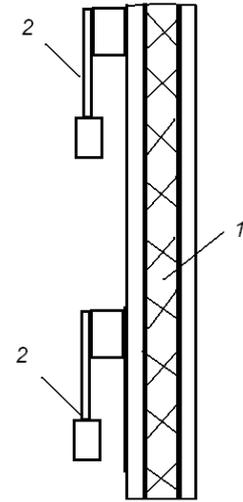


Fig. 9. The design scheme of the partition

Known design solutions that reduce noise in the passenger compartment to the level regulated by regalement documents. The disadvantages of sound insulation partitions are the low noise absorption coefficient at low frequencies, which leads to an increase in the acoustic load in the interior of the vehicle. The following tasks are considered:

1. Obtaining a condensed calculation scheme of a layered plate by bringing it to a single-layer Timoshenko plate with corresponding dynamic properties.
2. Finding the main dynamic characteristics (natural frequencies, natural oscillation forms) of the partition together with its attached elements.
3. Obtaining and analyzing discrete-continuous models with DVA.

In literature [11–14] provide methods for calculating three-layer composite plates with filler, and reducing the layered thin-walled plates to their equivalent beam of Timoshenko. For the partition as a Timoshenko plate, we obtain a system of algebraic equations depending on the frequency [10-14] (taking into account the DVA)

$$\begin{aligned} (-EIk^2 - SG - \rho I \omega^2) \gamma_0 - SGikw_0 = 0, \quad SGik\gamma_0 + (-SGk^2 + \rho S \omega^2) w_0 - (K_A + i\omega C_A)(w_A - w_0) = q_0 \\ -M_A \omega^2 w_A + (K_A + i\omega C_A)(w_A - w_0) = 0, \quad q_0 = 2(A_i - A_t), \quad -A_t ik \cos \varphi = \rho_a \omega^2 w_0, \end{aligned} \quad (2)$$

where E , G – are the modulus of elasticity of the plate (in cylindrical bending); S , I – the cross-sectional area and its moment of inertia; γ_0 , w_0 – the angular and normal displacements; ρ – the running density; A_i , A_t – the amplitudes of the incident and transmitted waves; C_v – the speed of sound; ρ_a – the density of the air; φ – the angle of incidence of the sound wave; w_A – displacement of DVA; M_A – its mass; K_A , C_A – its rigid and damping properties.

Solving the system of equations (2), we obtain:

$$A_t = F(\omega)A_i, \quad (3)$$

The power transfer coefficient due to the obstacle is set $\tau = \frac{|A_t|}{|A_i|}$ as well as the absorption coefficient in decibels $T_L = 10 \log |\tau^{-1}|$. Where $T_L = 10 \log \left| \frac{1}{F(\omega)^2} \right|$.

Fig. 10 shows this value in the presence and absence of DVA with specially selected parameters.

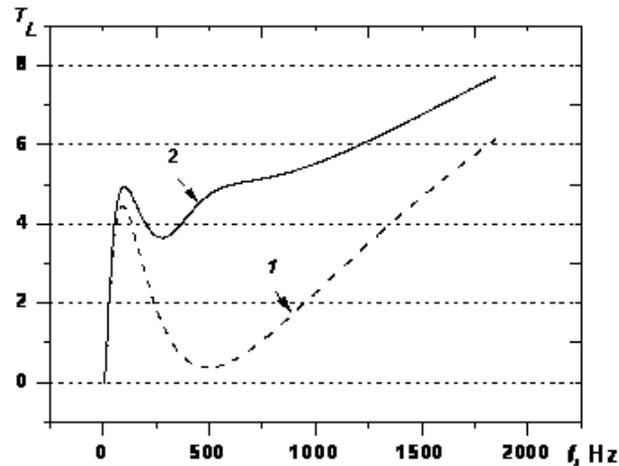


Fig. 10. The sound loss ratio for a plate without DVA (1) and with DVA adjusted for sound insulation failure (2)

Improved noise-absorbing properties in the vicinity of the failure of this characteristic for the partition without DVA may be seen.

6. SHOCK ABSORBING ROOF OF THE BU

We also investigated the problem of creating a shock-absorbing roof of a bus, a new design of which would increase the efficiency of absorption of impact energy during lateral turning, to preserve the volume of the bus and thus – the level of safety.

The impact-absorbing roof of the bus containing energy-absorbing elements, where the energy absorbing elements are made in the form of a bearing spatial rod symmetric system, consisting of supporting rods, one of which is pivotally secured to the lateral rack, interconnected by energy-absorbing composite heat absorbing in the mounting points of the bearing rods. This embodiment of the device provides energy absorption in the event of a load on the load-bearing structure of the body in lateral impact without significant deformation, as the mechanism of multipoint and controlled energy absorption, which is characterized by high efficiency, allows providing minimal spatial deformation of the vehicle body and thus significantly its safety in emergencies by changing the area of the composite layer.

The shock-absorbing roof of the bus contains (Fig. 11): 1 – energy absorbing composite rod; 2 – horizontal roof beams; 3 – the breaking rod; 4 – side bus racks; 5 – shock absorber coupling mounted in the places of attachment of the bearing rods, hinged fastening of one half of the horizontal beam of the roof to the side rack.

Fig. 11b shows a structure in a deformed (post-shock) state. For intense shock loads, the structure retains integrity and functions as a normal structural member of the bus roof. The shock-absorbing roof of the bus contains energy-absorbing elements, which are made in the form of a bearing spatial rod symmetric system, which consists of bearing rods 2, interconnected by a splitting rod 3 and connected with one shock absorbing composite rod 4 and shock absorbing muff. The main structure is the body of a vehicle, such as a bus.

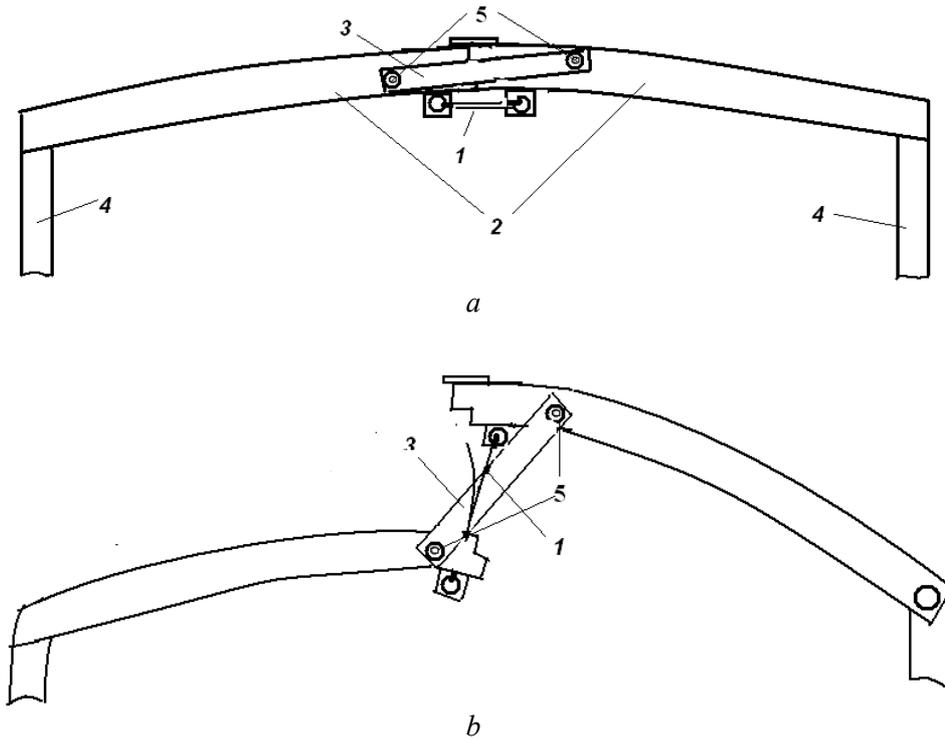


Fig. 11. Roof structure in an undeformed condition (a) and the shock-absorbing roof of the bus after reaching critical force (b)

The shock-absorbing roof of the bus operates in such a way that its principle of operation involves the presence of a mechanism of multipoint absorption of impact energy. During the overturning of bus 1 in the process of loading, for example, the left wall 1 (Fig. 11), there is a crushing of the shell of the left lateral or right lateral stand 4 and the subsequent movement of the left or right bearing rod 2, which caused the movement of the composite rod 3, which, in turn, causes the deformation of the composite rod 4 (detachment of one layer from another) (Fig. 11b) and the rotation of the shock-absorbing couplings 5, altering their internal construction, and the projection of the entire roof structure when rotating relative to the hinge 6 upwards, which significantly increases the efficiency of energy absorption, provides minimal spatial deformation of the body of the vehicle (in the direction of the passenger compartment) and increases the level of safety in emergencies.

In Fig. 12, energy-absorbing elements are presented: adhesive of unity, coupling working on rubbish.



Fig. 12. Energy-absorbing elements: (a) adhesive bonding, (b) clutch working

Several models have been developed to investigate the mechanics of machine destruction [17–19]. These models are simple, mostly single-mass. For a more accurate result, it is necessary to consider a generalized discrete-continuous calculation scheme [3–7]. To study the dynamics of the system: basic design – passengers, we use a relatively simple two-mass model (Fig. 13) with variable parameters. Fig. 14 shows the optimization results with constant damping and controlled friction.

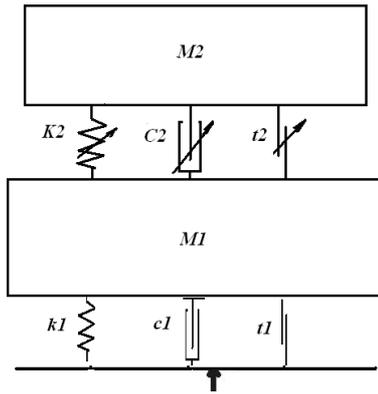


Fig. 13. Two mass model

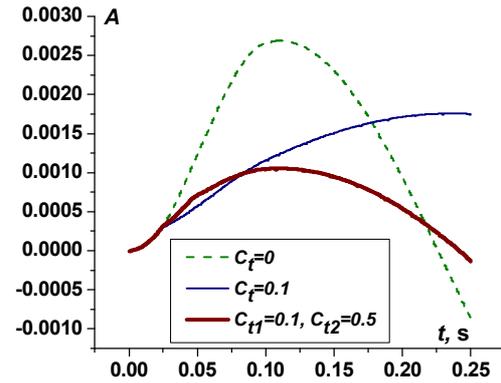


Fig. 14. Optimization results with constant damping and controlled friction

Friction, and in particular, controlled friction, reduce the acceleration of passengers and, consequently, their trauma when the bus is tipped over. Damping in the shock absorber can be controlled by changing the width of the adhesive layer along the lengths of the composite rod (Fig. 15).

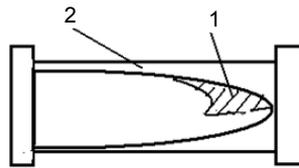


Fig. 15. Composite rod (2) with variable width adhesive layer (1)

To optimize the absorber, the algorithms given in [4–7] were used.

7. CONCLUSIONS AND RESEARCH PERSPECTIVES

The nonlinear suspension of a wheeled car with a DVA is considered. Based on the discrete-continuous approach, a three-mass design scheme was obtained, taking into account the unsprung masses and the associated DVA's. The optimization of the suspension parameters for the kinematic perturbation realizations as a random non-stationary process is carried out. An adaptive design scheme for noise barrier with DVA has been developed. The layered partition plate is modeled by a beam of Timoshenko. Significant improvement of the sound insulation properties in the low-frequency region has been demonstrated due to the connection of the DVA's system. A shock-absorbing roof of a bus is shown containing energy-absorbing elements, where the energy-absorbing elements are made in the form of a bearing spatial rod symmetric system, which consists of supporting rods, one of which is pivotally secured to the lateral rack, interconnected by energy-absorbing and composite energy-absorbing composite mounted on the mounting points of the bearing rods. This embodiment of the device provides energy absorption in the event of a load on the load-bearing structure of the body in lateral impact without significant deformation, as the mechanism of multipoint and controlled energy absorption, which is characterized by high efficiency, allows providing minimal spatial deformation of the vehicle body and thus significantly increases its safety in emergencies. By changing the area of the layer in the composite plate controlled changing of absorbing force is achieved.

References

1. Yao, H., Wang, T., Wen, B., & Qiu, B. (2018). A tunable dynamic vibration absorber for unbalanced rotor system. *Journal of Mechanical Science and Technology*, Volume 32 (4), 1519–1528. doi: 10.1007/s12206-018-0305-7 (in English).

2. Hu, H. L., & He, L. D. (2017). Online control of critical speed vibrations of a single-span rotor by a rotor dynamic vibration absorber at different installation positions. *Journal of Mechanical Science and Technology*, Volume 31(5), 2075–2081. doi: 10.1007/s12206-017-0404-x (in English).
3. Kernytskyy, I., Diveyev, B., Pankevych, B., & Kernytskyy, N. (2006). Application of variation-analytical methods for rotating machine dynamics with absorber. *Electronic Journal of Polish Agricultural Universities, Civil Engineering*, Volume 9, Issue 4 (in English).
4. Diveev B.M. Optyimizatsiia protsesiv zakhystu vid vibratsii na osnovi napivavtomatychnoho heneratora [Optimization of vibration protection processes based on a semi-automatic oscillator]. *Avtomatyzatsiia vyrobnychyykh protsesiv u mashynobuduvanni ta pryladobuduvanni [Automation of production processes in mechanical engineering and instrument making]*, Volume 39, 71–76 (In Ukrainian).
5. Diveyev, B. (2015). Impact and particle buffered vibration absorbers optimization and design. *Ukrainian journal of mechanical engineering and materials science*, Volume 1, Issue 2, 35–50 (in English).
6. Diveyev, B., Vikovych, I., Martyn, V., & Dorosh, I. (2015). Optimization of the impact and particle vibration absorbers. In *22nd International Congress on Sound and Vibration, ICSV 2015* (in English).
7. Cherchuk, H., Diveyev, B., Martyn, V., & Sava, R. (2014). Parameters identification of particle vibration absorber for rotating machines. *Proceeding of ICSV21, Beijing, China. (Electronic edition)* (in English).
8. Inoue, M., Yokomichi, I., & Hiraki, K. (2013). Design of particle/granules damper for vertical vibration with approximate analysis. *Journal of System Design and Dynamics*, Volume 7(4), 367–377. doi: 10.1299/jsdd.7.367 (in English).
9. Diveev B.M., Globchak M.V. & Gorbay O.Z. (2018) *Declaration patent for utility model No. 122780*. Ukraine (In Ukrainian)
10. Renji, K. (2005). Sound transmission loss of unbounded panels in bending vibration considering transverse shear deformation. *Journal of Sound and Vibration*, Volume 283(1-2), 478–486 (in English).
11. Diveyev, B., Horbay, O., Pelekh, R., & Smolskyy, A. (2012). Acoustical and vibration performance of layered beams with the dynamic vibration absorbers. In *19th International Congress on Sound and Vibration 2012, ICSV 2012* (pp. 1491–1498) (in English).
12. Diveyev, B., Butiter, I., & Shcherbina, N. (2008). Identifying the elastic moduli of composite plates by using high-order theories. *Mechanics of Composite Materials*, Volume 44(1), 25–36. doi: 10.1007/s11029-008-0004-z (in English).
13. Diveyev, B., Konyk, S., & Crocker, M. J. (2018). Dynamic properties and damping predictions for laminated plates: High order theories–Timoshenko beam. *Journal of Sound and Vibration*, Volume 413, 173–190. doi: 10.1016/j.jsv.2017.10.017 (in English).
14. Diveyev, B. (2016). Sound Transmission Properties of Composite Layered Structures in the Lower Frequency Range. *Ukrainian journal of mechanical engineering and materials science*, Volume 2/2, 11–32. (in English).
15. Diveev B.M., Ostashuk M.M., Gorbay O.Z., Tarasjuk U.I. & Bojkiv M.V (2014) *Declaration patent for utility model No. 93600*. Ukraine (In Ukrainian).
16. Olivares, G., & Herman, T. (2005). *Mass Transit Crashworthiness Statistical Data Analysis*. NIAR Technical Report No. FTA40002, Retrieved from: <http://www.fta.dot.gov/documents/Crashworthiness^Report.pdf>. (in English).
17. Yang, Z., Yan, H., Huang, C., Diao, X., Wu, X., Wang, S. & et al. (2014). Experimental and numerical study of circular, stainless thin tube energy absorber under axial impact by a control rod. *Thin-Walled Structures*, Volume 82, 24–32. doi: 10.1016/j.tws.2014.03.020 (in English).
18. Li, G., Xu, F., Sun, G., & Li, Q. (2015). A comparative study on thin-walled structures with functionally graded thickness (FGT) and tapered tubes withstanding oblique impact loading. *International Journal of Impact Engineering*, Volume 77, 68–83. doi: 10.1016/j.ijimpeng.2014.11.003 (in English).
19. Obradovic, J., Boria, S., & Belingardi, G. (2012). Lightweight design and crash analysis of composite frontal impact energy absorbing structures. *Composite Structures*, Volume 94(2), 423–430 doi: 10.1016/j.compstruct.2011.08.005 (in English).
20. Diveev B. M., Ostashuk M. M., Gorbay O. Z., Kernitsky I.S. & Pelech Y. A (2017) *Declaration patent for utility model No. 114977*. Ukraine (In Ukrainian).

Received 17.06.2020; Accepted in revised form 19.08.2020.