UDC 625.282
DOI: 10.15587/1729-4061.2022.253561

IMPROVING VEHICLE DAMPING OF VIBRATION BY IMPROVING PNEUMATIC SPRING

Vyacheslav Masliyev
Corresponding author
Doctor of Technical Sciences, Professor
Department of Electric Transport and Heat Engineering*
E-mail: masliew@ukr.net

Vladislav Dushchenko
Doctor of Technical Sciences, Professor
Department of Information technologies and systems of wheeled and caterpillar machines named after A. A. Morozov*

Vitalii Yepifanov
PhD, Professor, Director
Primary and Scientific Institute of Mechanical Engineerind and Transport*

Roman Nanivskyi
PhD
Department of the Scientific and Organizational**

Yurii Cherevko
PhD
Scientific Center of Land Forces**

Anton Masliev
PhD
Individual entrepreneur
Yureva blvd, 3, Kharkiv, Ukraine, 61082

Alexander Demydov
PhD, Senior Lecturer
Department of Electrical Transport and Diesel Locomotive*

Yurii Makarenko
PhD
LLC STC «Technosintez»
Polevaya str., 67, Kharkiv, Ukraine, 61001

*National Technical University «Kharkiv Politechnic Institute»
Kyrpychova str., 2, Kharkiv, Ukraine, 61002
**Hetman Petro Sahaidachnyi National Army Academy
Heroes of Maidan str., 32, Lviv, Ukraine, 79026

It is known that the air suspension of vehicles, in which diaphragm-type air springs are used as an elastic element, do not provide the necessary vibration damping. The reason for this is that such air springs have a relatively large passive part. As a result, a relatively small mass of compressed air crosses through the throttle installed between the air spring and the additional reservoir. This mass of air contains thermal energy, into which the energy of vibrations, which enters through the walls of the additional reservoir into the environment, has turned. This is interpreted as vibration damping, which is insufficient due to the low air mass.

Therefore, hydraulic vibration dampers are installed parallel to the diaphragm air springs, which complicates and increases the cost of the vehicle. Increasing the damping properties of such air suspensions could eliminate these hydraulic vibration dampers, which would reduce costs and simplify operation.

An air suspension with an improved air spring has been proposed, which has an increased effective area and a reduced «passive» capacity, an empirical formula has been built to determine its damping coefficient, as well as an expression for the stiffness coefficient. Mathematical modeling of oscillations of vehicles with different designs of pneumatic springs was carried out in order to improve their damping. The mathematical model takes into account the change in the parameters of the air spring during vibrations. The study was carried out for the diesel train DL-02. Using mathematical modeling, the effectiveness of the air suspension with an improved air spring has been proven: its damping index reaches 0.263, and the vibration damping coefficient is 45,859 kg/s, which corresponds to the values recommended for vehicles

Keywords: vehicle, air suspension, air spring, throttle, additional reservoir, simulation, damping factor

Received date 17.12.2021
Accepted date 22.02.2022
Published date 28.02.2022

How to Cite: *Masliev, V., Dushchenko, V., Yepifanov, V., Nanivskyi, R., Cherevko, Yu., Masliev, A., Demidov, A., Makarenko, Yu. (2022). Improving vehicle damping of vibration by improving pneumatic spring. Eastern-European Journal of Enterprise Technologies, 1 (7 (115)), 59–66. doi: https://doi.org/10.15587/1729-4061.2022.253561*

1. Introduction

Oscillations and vibrational accelerations of vehicles during movement create the problem of providing comfortable conditions for passengers and saving the path (path) from damage.

Specialists solve this problem by optimizing the characteristics of suspensions, as well as creating fundamentally new designs.

One of the most common ways to solve this problem is the use of suspensions based on pneumatic springs (PS) on vehicles. This allows to implement in these dimensions much more metal springs of its static deflection, that is, to implement a «soft» suspension. This ensures that the natural frequency of vertical vibrations of the body is about 1 Hz, which is the most favorable for passengers. The pneumatic springs also filter out the noise and vibrations generated when the wheels roll on or along the track. This ensures proper comfort for passengers and reduces dynamic loads on the track, which helps to increase their service life.

The expediency of using air suspension on vehicles is proved by the experience of leading countries: France, Japan, etc., where almost all high-speed trains and heavy vehicles are equipped. Their use is considered to be an actual direction for improving the technical level of the vehicle.

Therefore, research on improving the damping of vehicle vibrations by improving pneumatic springs is relevant.

2. Literature review and problem statement

The paper [1] presents the results of the research air-hydraulic springs with an adaptive self-adjusting damper for a suspension that provides two-stage inelastic resistance. It is shown that this makes it possible to reduce the acceleration of «shaking» and the heating of the damper when driving over small irregularities, etc. But the question remained unresolved, which is associated with the excessive complexity of the damper in terms of design and manufacturing technology and the cost of repairs in operation. The reason for this may be the difficulties associated with the declared functions that the damper must perform. An option to overcome the corresponding difficulties may be to abandon such a damper altogether and replace it with an pneumatic spring, which has sufficient self-damping.

The paper [2] presents the results of studies of a regenerative shock absorber, which can transform the vibration energy into electrical energy. In the future, it is used to charge the battery or is converted into heat by a resistor, which is released into the environment. This is treated as knee damping.

But the issue related to the excess cost of the damper remained unresolved, which is due to the use of rare earth metals for the manufacture. An option to overcome the corresponding difficulties may be to abandon such a damper altogether and replace it with an pneumatic spring, which has sufficient self-damping.

The paper [3] presents the results of studies of the influence of changes in the PS parameters on its dynamic stiffness. As a result, it was found that the rigidity increases in proportion to the PS deformation and the equilibrium value of the air pressure in it. But the question remained unresolved, how the presence of an additional reservoir affects the dynamic rigidity of the PS.

The paper [4] presents the results of studies of vibration damping of a car on a suspension with the help of a shock absorber in which an electrorheological fluid is used. The shock absorber is controlled by current from an electric generator that converts vibration energy into electrical energy.

But the question remained unresolved, which is associated with the instability of the characteristics of the shock absorber, which is associated with the dependence of the viscosity of the electrorheological fluid on the ambient temperature, which has sufficient self-damping.

The work [5] defines the requirements for magnetorheological elastomers used to control the damping characteristics of the suspension. Damping control has been found to have an effect on ride comfort. However, the authors considered only the dependence of damping on the relative speed of movement of the suspension components, and its dependence on the speed of movement was not evaluated.

In [6], it is proposed to use electromagnetic vibration dampers in the vehicle suspension. The authors have compiled a procedure for choosing the optimal parameters for an electromechanical damper, where there is no mechanical contact and friction between its components. This should increase its resource and reliability in operation. The disadvantage of this damper is a complex and expensive control system. A solution to this problem may be to use a different physical principle of the damper.

The article [7] considers modeling of dynamic processes in an electromechanical damper. The dependences of the damping force are implemented, the dependence of damping on the relative speed of movement of the suspension components, and its dependence on the speed of movement was not evaluated.

The article [8] compares two dampers that differ in the physical principle of operation: friction dampers, in which external friction of the working bodies occurs, and hydraulic dampers, in which internal friction is inherent. The studied influence on mobility indicators showed that friction dampers provide better machine mobility compared to hydraulic dampers. However, the authors considered only the mobility of the machine, and did not investigate its smoothness of movement.

In [9], a nonlinear PS equation was derived, and an analysis was made of the influence of various parameters on the dynamic stiffness with a change in the amplitude and frequency of excitation. But the question remained unresolved, how the presence of an additional reservoir affects the dynamic rigidity of the PS.

In [10], a method for studying the dynamic properties of an pneumatic spring using a hole (throttle) to create damping is given, since exact approaches have not yet been proposed. Recovery and damping forces in the pneumatic spring vary depending on the amplitude of the oscillations. A model is proposed for calculating the coefficients of elasticity and damping. But the question remained unresolved, connected with taking into account in the model the ratio of its «active» and «passive» parts. An option to overcome the corresponding shortcomings may be to refine the model in the direction of taking into account the relevant parameters.

The results of theoretical studies of air suspension based on the «active» capacity of the pr are presented in [11]. The authors proposed an empirical formula for calculating the vibration damping coefficient, which gives satisfactory results for air suspensions with balloon PS. As for diaphragm PS, this formula gives significantly overestimated results. This is due to the fact that it does not take into account the «passive» capacity of the diaphragm pr, which has a significant impact on the mass of compressed air flowing through the throttle between the pr and the additional reservoir.

Thus, the empirical formula for calculating the oscillation damping coefficient requires further development.

All this suggests that it is advisable to conduct a study on improving the damping of vehicle vibrations and improving this formula. This will make it more universal and suitable for calculating the damping coefficients of oscillations of air suspensions with balloon and diaphragm PS.

3. The aim and objectives of research

The aim of research is to draw up empirical formulas for the coefficients of stiffness and damping of an improved air suspension and a comparative assessment of its effectiveness. This will allow at the design stage to determine the rational parameters of the improved air suspension.

To achieve the aim, the following objectives were set:

– determine the structural features, parameters and draw up a design scheme for an improved air suspension;

– finalize the mathematical model of mass fluctuations on an improved air suspension;

– create empirical formulas for the stiffness and damping coefficients of the improved air suspension;

– carry out computer simulation of mass fluctuations on an improved air suspension;

– conduct a comparative analysis of the improved air suspension with other types of suspensions in terms of stiffness, damping and damping indicators and develop recommendations for its application in practice.

4. Materials and methods of research

The object of research is an improved air suspension for vehicles.

A hypothesis has been put forward that it is possible to increase the damping of the air suspension by increasing the amount (mass) of compressed air flowing through the throttle between the pneumatic spring and the additional reservoir.

The following assumptions are accepted: air pressure and its mass flow through the throttle are determined by polytropic thermodynamic processes occurring in the air suspension during oscillations.

Due to the small range of air temperature changes in air suspensions, let's consider constants: its density, heat capacity at constant volume, gas constant and heat capacity at constant pressure. For simplicity, let's assume that the ambient temperature is 293 K.

The study was carried out by computer simulation of mass fluctuations on an improved air suspension.

The research methodology was assumed using the MATLAB Simulink software package (USA) do the following:

– obtain characteristics of dynamic load;

– check the performance of the air suspension with the selected variant of the set of parameters, especially the size of the throttle cross-sectional area in order to prevent it from "locking" when the air flow reaches the speed of sound;

– prove the adequacy of the proposed dependencies for the stiffness and damping coefficients;

– determine the optimal parameters of the improved air suspension according to the criterion of maximum approximation to the upper limit of vibration damping indicators recommended for the vehicle.

5. Results of a study of the dynamics of a vehicle equipped with an air suspension with an improved pneumatic spring

5. 1. Features of the structure, parameters and design scheme of the improved air suspension

Diaphragm PSs (Fig. 1) are used on vehicles as elastic elements of a spring suspension. At the same time, PSs install devices for dissipating vibrational energy into the environment in the form of heat [11]. Usually, hydraulic dampers are used for this, the reliability of which in operation is low due to the presence of friction forces between its components and, accordingly, their wear [12].

To improve the reliability of the air suspension, it was proposed to implement oscillation damping by throttling the air flow, which, during oscillations, flows between the PS and the additional reservoir. This air contains thermal energy, into which the energy of vibrations has turned. It is transferred through the walls of the additional tank to the environment, which is interpreted as vibration damping.

An experimental freight diesel locomotive 2TE116 No. 184 was equipped with such an air suspension based on a balloon PS (Fig. 2, *a*). It has successfully passed operational tests at the Yelets depot with the speed of freight trains up to 100 km/h. Hydraulic dampers were not used here: vibration damping was carried out exclusively by air suspension [13].

In the process of dynamic tests of the diesel locomotive 2TE116 No. 184, it was found that in order to increase the damping properties of the air suspension, it is necessary to increase the mass of air, which, during oscillations, flows between the PS and the additional tank. This can be effectively implemented under certain conditions.

First, each PS must be connected (by air) to an additional tank, the capacity of which is 3–10 times its capacity.

Secondly, the PS should have as large an active capacitance as possible and a smaller passive capacitance.

The fulfillment of the first requirement is complicated by the overall limitations of the vehicle, and the second is difficult to fulfill in the diaphragm PS, in which the «passive» capacitance significantly exceeds the «active» one (Fig. 1.)

The compressed air filling it during the oscillation process does not flow through the throttle into an additional reservoir, but mainly between its «active» 5 and «passive» 6 parts. In this case, there is no loss of oscillation energy. Therefore, an air suspension based on a diaphragm PS does not have the ability to realize sufficient vibration damping.

But diaphragm pneumatic springs are still used on vehicles due to the fact that their shells are manufacturable and provide the necessary vertical, transverse and rotating deformations that occur during vehicle movement.

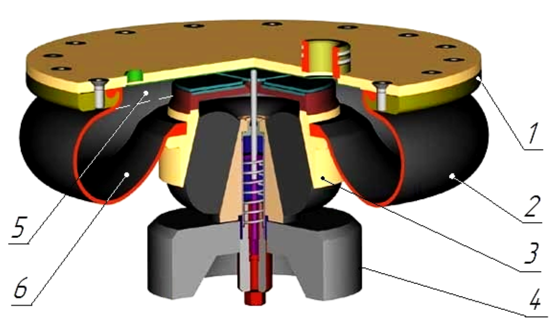

Fig. 1. Pneumatic springs of the «diaphragm» type: 1 — upper bottom; 2 — flexible shell; 3 — elastic buffer; 4 — base plate; 5, 6 — «active» and «passive» parts of the pneumatic spring capacity — respectively

Experts see the solution to the problem of increasing the damping qualities of the PS in the use of a balloon PS, Fig. 2, *a* and proper selection of the throttle section [14]. During full-scale tests of the diesel locomotive 2TE116 No. 184, the vibration damping index was 0.21, which is close to the indicators (0.2–0.35) recommended for the vehicle [15].

This value of the vibration damping index is sufficient for speeds up to 100 km/h, but insufficient for high speeds.

This prompted the authors to create an improved air suspension with increased damping, Fig. 3, the design of which is protected by a patent [16].

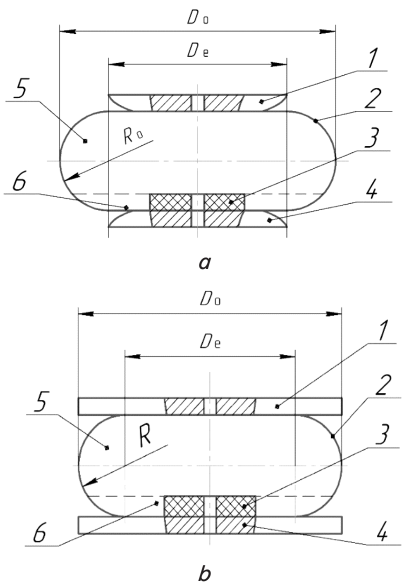

Fig. 2. Pneumatic springs of balloon type: *a* – serial; *b* – improved: 1 – upper bottom; 2 – flexible shell; 3 – elastic buffer; 4 – base plate; 5 – «active», 6 – «passive» parts of the pneumatic spring capacity

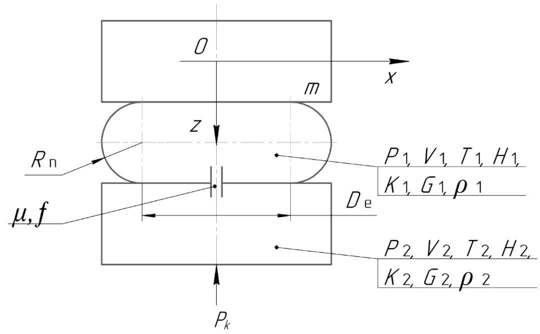

Fig. 3 Calculation scheme of the air suspension: P_j, V_j, T_j, H_j, K_j, G_j, ρ_j – respectively: pressure, volume, temperature (in Kelvin), surface area, heat transfer coefficient, mass and density of air in the PS (j=1), and in additional tank (j=2); D_e – effective diameter of the PS under a stable nominal load; μ, f – respectively: coefficient of air leakage through the throttle and its cross-sectional area; RP – radius of the flexible shell under a stable rated load; m – part of the mass of the body, which falls on the PS; P_k – function of the excitation force, selected in the form of a P-like impulse or a sinusoid; XOZ – fixed coordinate system

The increased damping properties of the air suspension with improved PS are due to the following energy processes that occur with the air in its cavity during vibrations during movement.

When the vehicle oscillates on a certain mariner, its upper bottom moves to the bottom, Fig. 4. This causes compression (and temperature increase) of the air that fills the PS. As a result, it flows into an additional reservoir 3, where its thermal energy enters the environment through the walls. On the return stroke (rebound), this air returns to the pneumatic spring 2 already cooled and denser and, accordingly, smaller in volume. Therefore, the displacement of the upper bottom 3 during rebound will be reduced. Such a process is identified as damping.

An analysis of the physical processes that occur with compressed air filling the air suspension system proves that the damping coefficient β depends on changes in the geometry of the constituent elements of the system and the modes of flow of gas-thermodynamic processes during vibrations.

In [14], a hypothesis was put forward that the coefficient β depends not only on the cross section of the throttle opening f, but also on the amount of air G flowing through it from the active capacity of the PS to the additional reservoir and in the opposite direction.

Subsequent studies revealed the possibility of further increasing the amount of air flowing through the throttle. To do this, the design of the reinforcement was changed for the PS containing a flexible shell of a balloon type, namely: the diameters of the upper and lower bottoms are made equal to the outer diameter of the flexible shell. When PS is compressed, this provides an increase in the effective area and, accordingly, the mass of air that will flow through the throttle into the additional reservoir.

This necessitated the refinement of the mathematical model given in [14] and the study of the stiffness and damping characteristics of the air suspension with improved PS.

The mathematical model is compiled in the form of a system of differential equations of vertical oscillations of the part of the body mass per one air suspension, according to the scheme shown in Fig. 3.

5. 2. Mathematical model of mass fluctuations on an improved air suspension

Using the d'Alembert principle, a differential equation for fluctuations of the part of the body mass m, which falls on the PR when its compressed air supply is turned off, is compiled:

$$m\ddot{z} + \beta\dot{z} + Cz = P_k, \tag{1}$$

where denoted: $\beta=\beta(\omega, \rho, VA, V1, V2, H2, f...)$ – damping coefficient; $C=C(n, D_e, V1, V2, P)$ – air suspension stiffness factor.

The air pressure $P_1$and P_2 and its mass flow through the throttle $G1=G2$ are determined by polytropic thermodynamic processes that occur in the air suspension during vibrations.

Due to the small range of air temperature changes in air suspensions, we will consider constant: its density $\rho_1=\rho_2$, heat capacity at constant volume C_V, gas constant R and heat capacity at constant pressure $C_P=C_V+R$, as well as ambient Kelvin temperature T_0.

The first law of thermodynamics for air in the PS during its compression: $dQ1=dU1+P1dV1$, where $dU1=G1CvdT1+$ $+T1CvdG1$ determines the change in the internal energy of the air in the PS. The second term $i=U+P1V1=$ $=CP\,T1=(Cv+R)T1$ represents the enthalpy of the elementary amount of air $dG1$ flowing out or entering the PS through the throttle opening connecting the PS with an additional tank: $dG1$ will be positive when air leaks from the PS and negative when air enters it $dQ1=-k1H1(T1-T0)dt+idG1$ – the heat lost by the PS air (or supplied to it from outside).

So, the first law of thermodynamics for air at the PS in the course of its compression will take the following form:

$$G_1C_v dT_1 + P_1 dV_1 - RT_1 dG_1 + k_1 H_1 (T_1 - T_0) dt = 0. \tag{2}$$

Air equation of state in differential form:

$$P_1 dV_1 + V_1 dP_1 = RG_1 dT_1 + RT_1 dG_1. \tag{3}$$

If to consider the process to be adiabatic, then the equation for the air flow from the PS to the additional tank through the throttle has the form:

$$\mathrm{d}G_1 \cong -\mu f \sqrt{2\rho(P_1 - P_2)}\, \mathrm{d}t. \quad (4)$$

Thus, the gas-thermodynamic processes in the air suspension during compression of the pneumatic spring are described by equations (2)–(4).

Taking into account the fact that the capacity of the additional tank is constant, i.e. $\mathrm{d}V2=0$, the equations of gas thermodynamics will look like this:

$$G_2 C_v \mathrm{d}T_2 + k_2 H_2 (T_2 - T_0)\mathrm{d}t + \\ +(T_2 - T_1) C_v dG_2 - RT_1 \mathrm{d}G_2 = 0. \quad (5)$$

$$V_2 P_2 = RG_2 dT_2 + RT_2 dG_2. \quad (6)$$

Since the amount of air in the air suspension does not change, the air mass balance equation will be:

$$\mathrm{d}G_1 + \mathrm{d}G_2 = 0. \quad (7)$$

The system of thermodynamic alignments is valid for the compression stroke of the pneumatic spring: $\mathrm{d}G1<0$, $\mathrm{d}T1>0$, $\mathrm{d}V1<0$.

During the expansion of the PR: $\mathrm{d}G1>0$, $\mathrm{d}T1<0$, $\mathrm{d}V1>0$, the system may look offensive.

For PS:

$$G_1 C_v \mathrm{d}T_1 + C_v T_1 \mathrm{d}G_1 + P_1 \mathrm{d}V_1 - C_v T_2 \mathrm{d}G_1 - \\ -RT_2 \mathrm{d}G_1 + k_1 H_1 (T_1 - T_0)\, \mathrm{d}t = 0. \quad (8)$$

$$P_1 \mathrm{d}V_1 + V_1 \mathrm{d}P_1 = RG_1 \mathrm{d}T_1 + RT_1 \mathrm{d}G_1. \quad (9)$$

$$\mathrm{d}G_1 \cong -\mu f \sqrt{2\rho(P_2 - P_1)}\, \mathrm{d}t. \quad (10)$$

For additional tank:

$$G_2 C_v \mathrm{d}T_2 + k_2 H_2 (T_2 - T_0)\mathrm{d}t + \\ +(T_2 - T_1) C_v \mathrm{d}G_2 - RT_2 \mathrm{d}G_2 = 0. \quad (11)$$

$$V_2 \mathrm{d}P_2 = RG_2 \mathrm{d}T_2 + RT_2 \mathrm{d}G_2. \quad (12)$$

The mathematical model contains 12 non-linear differential equations of the second and first order with variable coefficients. Therefore, it is expedient to study it by numerous methods using computers.

5. 3. Empirical formulas for stiffness and damping coefficients of improved air suspension

When using the graph-analytical method, the dependence of the effective area of the improved PS on the deformations z was obtained:

$$F=\pi((D_e/2)+z)^2. \quad (13)$$

Let's assume that the «active» capacity of the PS (m^3) is equal to:

$$VA=\pi((D_e/2)+z_{max})^2 \cdot 2z_{max}, \quad (14)$$

where z_{max} – the maximum allowable vertical deformation of the PS during vibrations.

The stiffness coefficient (N/m) at the nominal position of the air suspension with serial PR is [14]:

$$C_0 = \frac{nP_1\left(\pi(De/2)^2\right)^2}{V_1 + V_2}, \quad (15)$$

and for the air suspension with improved PS, the following expression is proposed:

$$C = \frac{nP_1\left(\pi\left((De/2)+z_{max}\right)^2\right)^2}{V_1 + V_2}. \quad (16)$$

The following empirical formula for the damping factor is proposed:

$$\beta = \omega \cdot \rho \cdot \pi\left((De/2)+z_{max}\right)^2 \cdot 2 \cdot z_{max} \cdot V_2 \cdot H_2 / V_1 \cdot \mu \cdot f. \quad (17)$$

For air suspension with conventional balloon or diaphragm PS:

$$\beta_0 = \omega \cdot \rho \cdot \pi(De/2)^2 \cdot 2 \cdot z_{max} \cdot V_2 \cdot H_2 / V_1 \cdot \mu \cdot f. \quad (18)$$

The above empirical formulas take into account the dependence of the damping coefficient on the oscillation frequency and air density, since they affect the value of its mass, which, during oscillations, flows through the throttle between the PS and the additional reservoir. This value of the air mass is also affected by: the ratio of the active capacity of the PS (14) and its capacity V_1 in the nominal position. This takes into account the value of the «passive» capacitance of the PS and the plane of the surface of the additional reservoir, through which the main part of the vibration energy is transferred to the environment.

5. 4. Computer simulation of mass fluctuations on an improved air suspension

To assess the adequacy of the empirical formulas (16)–(18), the coefficients calculated from them, calculated from the formulas known from the theory of vibrations and from computer simulation (1)–(12), were compared. Wheel frequency of natural vibrations of the mass on air suspension:

$$\omega = (C/m)^{0.5}. \quad (19)$$

The value of the critical value of the damping coefficient, kg/s:

$$\beta_{cr}=2\,(C \cdot m))^{0.5}. \quad (20)$$

For vehicles, it is recommended to choose the damping coefficient at the level:

$$\beta=(0.2...0.35)\beta_{cr}. \quad (21)$$

Table 1, 2 show the parameters of the air suspension in the position of static equilibrium in relation to the DEL-02 diesel train.

The initial data given in Table 1, the rational: they are selected based on the results of theoretical studies and full-scale tests of a row of experimental diesel locomotives that were equipped with air suspension [13].

Table 1

Output data of design air suspensions

Pneumatic spring	ω, rad/s	ρ, kg/m^3	R_s, m	m, t	D_e, m	V_1, m^3	V_2, m^2	H_1, m^2	H_2, m^2	f, m^2	μ	P_1, MPa
1	6.24	4.9	0.05	12.3	0.56	0.022	0.07	0.6	1.97	$2\cdot10^{-4}$	0.75	0.49
2	7.1	3.8	0.03	12.3	0.64	0.019	0.070	0.67	1.97	$2\cdot10^{-4}$	0.75	0.38
3	6.24	4.9	0.10	12.3	0.56	0.062	0.030	0.90	0.3	$2\cdot10^{-4}$	0.75	0.49

Table 2

Thermotechnical data of air suspensions for steady state

n	C_V	C_P	R	K_i	T_0
–	J/(kg·deg)	J/(kg·deg)	J/(kg·deg)	W/(m^2·deg)	–
1.4	717	1000	287	1000	293°

In the process of computer simulation, the cross-sectional area of the throttle was varied. With its decrease in magnitude, given in Table 1, a gradual increase in the frequency of natural oscillations was observed. This is due to a decrease in the mass of air flowing between the PS and the additional reservoir as a result of the air flow reaching the speed of sound.

Fig. 4 shows the characteristics of quasi-static 1 and dynamic load 2 on the air suspension with improved PS, obtained in the study of the mathematical model (1)–(12). With a quasi-static load, an isothermal process occurs at a certain suspension, so the characteristic is linear. Under a dynamic load with a constant frequency, but with an oscillation amplitude varying within $\pm z_{max}$, characteristic (2) becomes non-linear. It practically coincides with the characteristics of the loading of air suspensions, created on the basis of diaphragm and serial balloon PS.

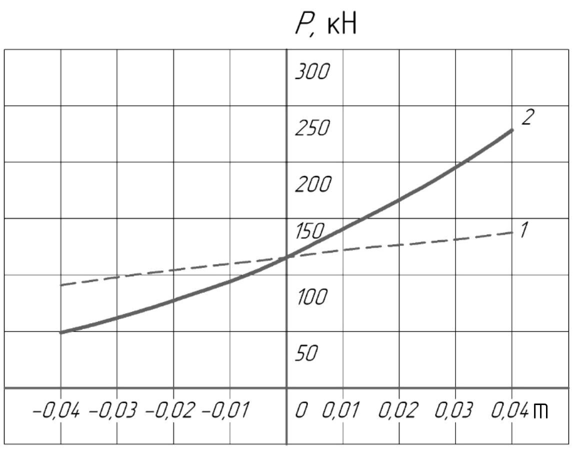

Fig. 4. Characteristics of the load of the air suspension with an improved pneumatic spring with an additional tank capacity of 0.07 m^3: 1 — quasi-static load (isothermal process); 2 — dynamic load with an oscillation frequency of 1 Hz

The load on the sprung part of the vehicle, transmitted through the air suspension with improved PS when driving along the track or track, is shown in Fig. 4. They are calculated as the product of the amplitude value of the air pressure and the corresponding value of the effective area of the pneumatic spring.

Oscillograms of natural oscillations of the mass on air suspensions with different PSs for: with diaphragm, balloon and improved balloon, are shown in Fig. 5–7. The analysis of the oscillograms proved that the calculated and simulated frequencies of natural oscillations of all three air suspensions are almost the same and are about 1 Hz. They are fading.

Comparison of air suspensions was also made according to the damping index D, which was calculated from the oscillograms:

$$D=\left(\ln\left(A_1/A_2\right)\right)/2\pi, \tag{22}$$

where A_1 and A_2 are successive oscillation amplitudes.

The air suspension damping coefficient is calculated using its critical value using the formula:

$$\beta=D\cdot\beta_{cr}. \tag{23}$$

This will make it possible to compare the damping properties of all three types of suspensions: with diaphragm, serial balloon and improved balloon air springs.

5. 5. Comparative analysis of air suspensions in terms of stiffness, damping and damping indicators

The results of calculations by (15)–(21) and computer simulation by (1)–(12), (23) are presented in Table 3.

Table 3

The results of calculating the stiffness and damping coefficients

Pneumatic springs	ω, rad/s (19)	C, n/m (15), (16)	β, kg/s (17), (18)	β_{cr}, kg/s (20)	β, kg/s, (21)	β, kg/s, (23)
Balloon	6.06	451885	25815	149106	29821–52187	33312
Improved	7.10	617989	34210	174370	34874–61030	45859
Diaphragm	6.06	451885	11121	149106	29821–52187	7650

As can be seen from Table 3, the difference in the values of the damping coefficients obtained by the proposed empirical formulas (17), (18) and the results obtained by mathematical modeling does not exceed 25 %.

Oscillograms of natural vibrations of a part of the mass of the body on air suspension, obtained by computer simulation, are shown in Fig. 5–7.

Analysis of the obtained results proves that the damping coefficient of the air suspension with improved PS increased by 25 % compared to the serial PR. It is close to the recommended values calculated by (21) and included in the range of damping coefficients (resistance parameters) of vibration dampers UG190.32.32 for vehicle (24,000–42,000 kg/s) [15]. This indicates the adequacy of the proposed empirical formulas (17), (18).

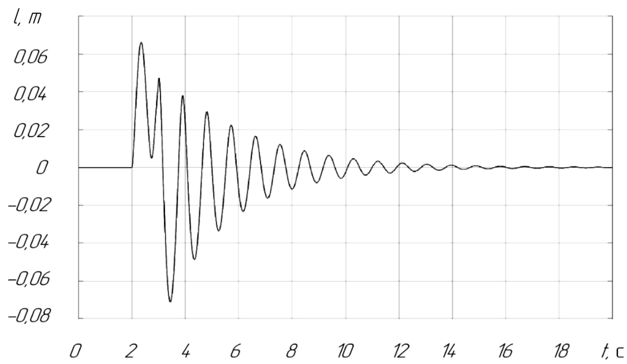

Fig. 5. Oscillogram of natural vibrations of the body mass on air suspension with diaphragm PS: $(VA/V1)=0.1$

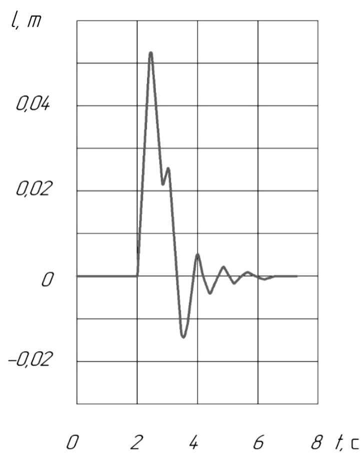

Fig. 6 Oscillogram of natural vibrations of the mass of the body on air suspension with balloon PS: $(VA/V1)=0.7$

Fig. 7. Oscillogram of natural oscillations of the mass of the body on air suspension with improved balloon PS: $(VA/V1)=0.9$

Table 4 shows the results of studies of the damping properties of air suspension for diesel trains DL 02.

As can be seen from Table 4, the performance and damping coefficients of the air suspension with improved PS are significantly higher than others and are within the ranges recommended for the vehicle.

Table 4

The results of studies of the damping properties of air suspension

Indicators and damping coefficients (opposite) of pneumatic suspension	with diaphragm PS	with serial balloon PS	with improved PS
D	0.058	0.210	0.263
β, kg/s	7650	33312	45859

6. Discussion of the results of the study of vibration damping of vehicles by improving the pneumatic springs

The dynamic load on the sprung part of the vehicle, transmitted through the air suspension with improved PS when driving along the track or track, is shown in Fig. 4. Its value for all three air suspensions is almost the same, but the number of oscillation periods is greatest for the air suspension of the diaphragm PS, which is associated with too weak damping (Fig. 5).

Most likely, vibrations are damped in air suspensions with improved PS (Fig. 7).

Damping coefficient (resistance parameter) calculated from oscillograms of natural vibrations of an air suspension with an improved balloon PS (β=45859 kg/s), close to the upper limit of the range (24,000–42,000 kg/s) of the hydraulic resistance parameter 2.

The damping index of the air suspension with a conventional balloon PS is close to the lower limit of the range.

The damping index and the resistance parameter of the air suspension with diaphragm PS are significantly less than those recommended for the vehicle, so the installation of hydraulic shock absorbers is necessary here.

The results obtained confirm the adequacy of the mathematical model of mass oscillations on air suspensions.

The accepted assumptions to a certain extent influence the results of the research, in particular in terms of the parameters of the components of the air suspension. Therefore, they should be considered as a first guess. Parameters (throttle cross-section, additional tank capacity, etc.) must be corrected during full-scale bench and field tests of vehicles equipped with improved air suspension.

7. Conclusions

1. The structural features of the improved air suspension are determined, namely, that its air spring (balloon type) has an increased diameter and, accordingly, the effective area of the upper and lower bottoms. With this in mind, a calculation scheme has been developed for compiling a mathematical model.

2. The mathematical model of vibrations of a part of the body mass on an improved air suspension has been improved in terms of taking into account the dependences of the effective area and the stiffness and damping coefficients on vertical displacements of the upper bottom during vibrations.

3. Empirical formulas have been compiled for the stiffness and damping coefficients of the improved air sus-

pension, which take into account the ratio of the active and total capacity of the air spring and its variable effective area.

4. Computer modeling of mass oscillations on an improved air suspension was carried out, which made it possible to obtain the characteristics of its quasi-static and dynamic load, as well as oscillograms of natural mass oscillations on an improved air suspension.

5. Comparison based on the results of simulation of an improved air suspension with a balloon air suspension in terms of damping coefficients and damping indicators revealed its advantages: they increased by 25 %.

References

1. Novikov, V. V., Ryabov, I. M., Chernyshov, K. V., Pozdeev, A. V., Pohlebin, A. V. (2020). Pneumohydraulic spring with adaptive self-adjusting damper for suspension of a high-speed tracked vehicle and its calculation procedure. Traktory i Sel'hozmashiny, 1 (2), 13–20. doi: https://doi.org/10.31992/0321-4443-2020-2-13-20
2. Zuo, L., Scully, B., Shestani, J., Zhou, Y. (2010). Design and characterization of an electromagnetic energy harvester for vehicle suspensions. Smart Materials and Structures, 19 (4), 045003. doi: https://doi.org/10.1088/0964-1726/19/4/045003
3. Takahashi, M., Wakui, S. (2009). Improvement of Isolated table Using Auxiliary Tank. Journal of the Japan Society for Precision Engineering, 75 (4), 542–547. doi: https://doi.org/10.2493/jjspe.75.542
4. Choi, S.-B., Seong, M.-S., Kim, K.-S. (2009). Vibration control of an electrorheological fluid-based suspension system with an energy regenerative mechanism. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 223 (4), 459–469. doi: https://doi.org/10.1243/09544070jauto968
5. Dushchenko, V. V., Masliev, V. G., Nanivskyi, R. A., Masliev, A. O. (2019). Application of magnetorheological elastomers for performance control of cushioning systems for wheeled vehicles. Electrical Engineering & Electromechanics, 5, 50–59. doi: https://doi.org/10.20998/2074-272x.2019.5.09
6. Liubarskyi, B., Lukashova, N., Petrenko, O., Pavlenko, T., Iakunin, D., Yatsko, S., Vashchenko, Y. (2019). Devising a procedure to choose optimal parameters for the electromechanical shock absorber for a subway car. Eastern-European Journal of Enterprise Technologies, 4 (5 (100)), 16–25. doi: https://doi.org/10.15587/1729-4061.2019.176304
7. Liubarskyi, B., Lukashova, N., Petrenko, O., Yeritsyan, B., Kovalchuk, Y., Overianova, L. (2019). Procedure for modeling dynamic processes of the electromechanical shock absorber in a subway car. Eastern-European Journal of Enterprise Technologies, 5 (5 (101)), 44–52. doi: https://doi.org/10.15587/1729-4061.2019.181117
8. Dushchenko, V., Vorontsov, S., Masliyev, V., Agapov, O., Nanivskyi, R., Cherevko, Y., Masliiev, A. (2021). Comparing the physical principles of action of suspension damping devices based on their influence on the mobility of wheeled vehicles. Eastern-European Journal of Enterprise Technologies, 4 (5(112)), 51–60. doi: https://doi.org/10.15587/1729-4061.2021.237312
9. Jang, J.-S. (2014). The Effects of Parameter Changes on the Properties of an Air Spring. Journal of the Korea Society For Power System Engineering, 18 (2), 77–82. doi: https://doi.org/10.9726/kspse.2014.18.2.077
10. Asami, T., Yokota, Y., Ise, T., Honda, I., Sakamoto, H. (2013). Theoretical and Experimental Analysis of the Nonlinear Characteristics of an Air Spring With an Orifice. Journal of Vibration and Acoustics, 135 (1). doi: https://doi.org/10.1115/1.4007677
11. Korobka, B. A., Shkabrov, O. A., Kovalenko, Yu. N., Nazarenko, V. F. (2010). Otechestvennaya passazhirskaya telezhka na pnevmopodveske. Vagonniy park, 6, 48–51.
12. Vol'pert, A. G., Zholobov, V. A. (1985). Vibrogasiteli podvizhnogo sostava: kakimi oni dolzhny byt'? Zheleznodorozhniy transport, 3, 54–57.
13. Masliev, V. G., Shevchenko, P. M., Evstratov, A. S., Alekseev, A. O. (1981). Pnevmaticheskoe ressornoe podveshivanie teplovoza 2TE116. Transportnoe mashinostroenie. Moscow: NIIINFORMTYaZhMASh.
14. Masliiev, A. O., Makarenko, Yu. V. Masliiev, V. H. (2015). Dempfiruvannia kolyvan kuzoviv transportnykh zasobiv, yaki obladnano pnevmatychnymy resoramy. Visnyk Natsionalnoho tekhnichnoho universytetu «KhPI». Zb. nauk. prats. Seriya: Transportne mashynobuduvannia, 43 (1152), 59–64.
15. Posibnyk z tekhnichnoho obsluhovuvannia ta remontu hidravlichnykh ta fryktsiynykh hasnykiv kolyvan pasazhyrskykh vahoniv. Rada z zh. d. transportu derzhav-uchasnyts Spivdruzhnosti, Protokol vid 21-22 zhovtnia 2014 r. No. 61.
16. Masliiev, A. O., Masliiev, V. H., Dushchenko, V. V. (2017). Pat. No. 113641 UA. Pnevmatychna pidviska. No. u201607535; zaiavl. 11.07.2016; opubl. 10.02.2017, Bul. No. 3.