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DETERMINING PATTERNS IN THE INFLUENCE OF THE GEOMETRICAL PARAMETERS OF THE CONNECTING PIPELINE ON THE DYNAMIC PARAMETERS OF THE PNEUMATIC SPRING OF RAILROAD ROLLING STOCK

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The object of this study is the process of determining the dynamic indicators of the pneumatic spring, which is used as the main element of the second stage of spring suspension in the high-speed rolling stock of the railroad, based on the influence of the geometric parameters of the connecting pipeline.

It has been established that the dynamic indicators of rolling stock mainly depend on the operation of the pneumatic spring, the characteristics of which are largely determined by the geometric parameters of the connecting element between the pneumatic spring and the additional tank.

A mathematical model of oscillations of a dual-mass system has been constructed, the elements of which are connected through a pneumatic spring suspension system. The operation of the pneumatic system is described using the Boyle–Mariotte equations, the state of the ideal gas, the energy for the flow in the connecting pipeline, and the law of conservation of energy.

Theoretical studies into the influence of the diameter and length of the connecting element of the pneumatic spring suspension system on energy loss and damping coefficient for the cycle of its operation and the rigidity of the pneumatic spring have been carried out.

It was established that the dependence of the stiffness of the pneumatic spring, while changing the value of the diameter of the connecting element from 6 mm to 30 mm, is nonlinear.

In the process of compression, the rigidity of the pneumatic spring changes with the length of the connecting element of 1 m from 927 kN/m, static rigidity, to 497 kN/m, dynamic rigidity.

The dependences of energy loss and damping coefficient for the operation cycle of the pneumatic spring suspension system based on the hysteresis loop have been constructed.

It was found that the difference between the damping coefficient in the process of compression and expansion of the pneumatic spring is no more than 4 %.

It was established that the design of high-speed rolling stock is impossible without high-quality modeling of the process of operation of the pneumatic spring suspension system

Keywords: rolling stock of railroads, pneumatic spring, connecting pipeline, dynamic characteristics of pneumatic springs

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1. Introduction

According to the Draft Plan for the Restoration of Ukraine [1], one of the main tasks of railroad transportation is the implementation of the project for the construction of a high-speed railroad (HSR) connecting Kyiv to Warsaw. This will reduce the travel time on this route from 17–19 hours to 5–6 hours, by reaching an average speed of 180 km/h. To create conditions for safe movement and achieve the specified speed, you need modern rolling stock. The main models of rolling stock are electric trains EKr-1 “Tarpan” (Fig. 1) with a design speed of 200 km/h (depending on the modifications) and diesel trains DPKr-2 and DPKr-3 (Fig. 2) with a design speed of 154 km/h and 140 km/h, respectively.

The main structural elements of rolling stock, on the operation of which its safe movement depends, are systems and devices for transmitting and damping dynamic forces. In the vertical direction, they are installed between the body and bogies, bogies and wheel sets. One of such systems is the spring suspension system, which consists of a pneumatic spring and an additional tank interconnected by a connecting pipeline. It is this system that absorbs impact energy when the wheel set interacts with the rail track, while providing the necessary level of indicators of traffic safety and comfort [2, 3].

Fig. 1. Electric train EKr-1 "Tarpan"

Fig. 2. Diesel train DPKr-3

As noted, one of the main elements of the pneumatic spring suspension system is the connecting pipeline. The geometrical parameters of the connecting pipeline mainly affect the rigidity of the pneumatic spring, energy dissipation, and damping coefficient for the cycle of operation of the pneumatic spring suspension system. Often, at the design stage, pneumatic spring suspension systems are developed according to analogs of other rolling stock. The operation of such a system is checked and improved during operational tests based on the analysis of dynamic indicators and traffic safety indicators [4, 5].

Therefore, the theoretical study of the work of the pneumatic spring suspension system is an integral part of the design of high-speed rolling stock. In the process of research, it is necessary to take into account the geometric parameters of the connecting pipeline and their impact on the main characteristics of the pneumatic spring and the system as a whole.

Thus, investigating the dynamic characteristics of the pneumatic spring suspension system is a relevant task in establishing traffic safety indicators of high-speed rolling stock. This will allow for the safe operation of high-speed rolling stock under various conditions of its interaction with the rail track.

2. Literature review and problem statement

The authors of work [6] conducted analytical studies into the influence of the parameters of the classical and dynamic model of the pneumatic spring (Fig. 3, 4) on the nature of the vehicle's movement. The classic model of the pneumatic spring consisted of two springs of appropriate rigidity. This representation is valid only for low frequencies and does not take into account the damping process in the connecting pipeline between the pneumatic spring and the additional tank. However, in the dynamic model, the authors managed to more accurately reflect the operation of the pneumatic spring suspension system by introducing a fictitious piston into the connecting pipeline.

However, the rigidity of the pneumatic spring, damping, and mass air flow through the connecting pipeline are considered constant values and do not depend on the deformation of the pneumatic spring during the operation of the rolling stock, but are the result of specified initial conditions.

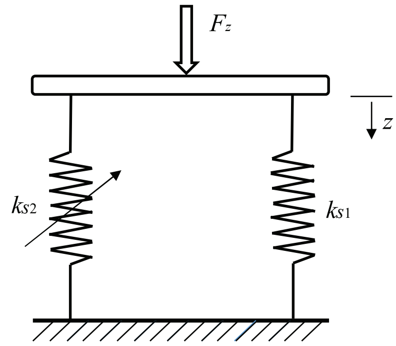

Fig. 3. Classic model of pneumatic springs [6]

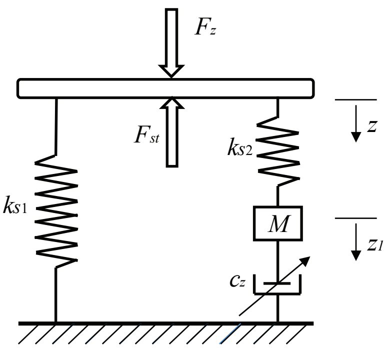

Fig. 4. Dynamic model of pneumatic springs [6]

In [7], experimental studies of the pneumatic spring suspension system using the ASTB bench were carried out. The influence of the diameter of the connecting pipeline on the dynamic rigidity of the pneumatic spring in the frequency range of 0–20 Hz, which is of interest to the dynamics of railroad vehicles, has been investigated. It has been established that the use of an additional tank leads to significant discrepancies between the rigidity of the pneumatic system at low and high oscillation frequencies of the rolling stock. This transition mainly depends on the parameters of the connecting pipeline. Theoretical studies were also conducted on the basis of the use of various mechanical models, namely modeling using one element of rigidity, the model Nishimura (Japan) [8], models VAMPIRE and GENSYS (Sweden) [9]. However, the use of these models involves the preliminary experimental determination of a certain number of input parameters, which greatly complicates the implementation of this task. In addition, equivalent mechanical models do not make it possible to explore dynamic behavior of the pneumatic spring suspension system during the complication of its shape and design.

In [10], the influence of the volume of the additional tank, the length and diameter of the connecting pipeline on the characteristics of the pneumatic spring suspension system was investigated. The model was developed in the MATLAB/Simulink software environment and takes into account the effects of elasticity, friction, and viscosity of the pneumatic spring in the vertical, transverse, and longitudinal directions. However, the mathematical model does not take into account the thermodynamic processes that arise as a result of the deformation of the pneumatic spring and the flow of air into an additional tank through the connecting pipeline, as well as the heat exchange of the pneumatic spring with the environment.

In [11], a dynamic model of vertical rigidity based on the laws of thermodynamics and hydrodynamics was developed, and the geometric parameters of the pneumatic spring were determined by the analytical method. The process of changing the state of the thermodynamic system was considered as iso-entropic. Dynamic vertical stiffness was found for the case of connecting a pneumatic spring to an additional tank through a hole of a certain diameter (there is no connecting pipeline). The use of such a pneumatic suspension system is possible in the case when the additional tank is the internal cavities of the bogie frame. However, on most types of rolling stock, a connecting pipeline with certain geometric parameters is used between the pneumatic spring and the additional tank. This allows for a wide range to regulate the dynamic rigidity of the pneumatic spring and energy loss in the pneumatic spring suspension system.

Work [12] reports an analytical model of the pneumatic spring suspension system based on the experimental characteristics obtained. To determine the mass flow of air through the connecting pipeline, Poiseuille's law was used, which establishes that the flow is laminar and one-dimensional. However, during the operation of the pneumatic spring suspension system, which may have a different configuration of the connecting pipeline, the fluid movement is turbulent.

In [13], a nonlinear adiabatic polytropic model of the pneumatic spring was built using the basics of fluid mechanics and thermodynamics. The analysis in that work was aimed at assessing the dynamic behavior of the proposed pneumatic spring model. However, no additional tank and connecting pipeline with corresponding geometric parameters were taken into account.

In [14], a generalized dynamic model was constructed, taking into account the thermodynamics of the pneumatic system "pneumatic spring-connecting pipeline-additional tank", effective friction and viscoelastic damping of the rubber chord shell. It should be noted that the parameters of the proposed model must be determined and additionally checked by conducting bench tests. In addition, in [14] there is no assessment of the influence of the geometric parameters of the connecting pipeline on the characteristics of the pneumatic spring.

In [15], theoretical and experimental studies into the vertical rigidity of the pneumatic spring were carried out. The developed mathematical model of the pneumatic spring provided for additional consideration of the rate of change in its effective area and effective volume due to deformation. However, the pneumatic spring was considered without an additional tank and a connecting pipeline, which differs significantly from the actual operating conditions of the pneumatic spring suspension system.

In [16], different types of connection of the pneumatic spring with an additional tank are investigated. It was established that for work in the low-frequency range of perturbations, the "pneumatic spring-hole-connecting pipeline-additional tank" system is effective. For high-frequency perturbations, the "pneumatic spring-hole-additional reservoir" system is effective. However, that paper does not establish the dependence of the characteristics of the pneumatic spring suspension system on the geometric parameters of the connecting pipeline.

The influence of pneumatic resistance of the connecting element on the elastic and damping properties of the pneumatic spring was investigated in [17, 18]. A model of a pneumatic spring as a dynamic system with three phase coordinates (pressure in a pneumatic spring and an additional tank, air mass in a pneumatic spring) has been developed. The process of changing the state of air inside the pneumatic spring was considered as adiabatic and the mass air flow through the connecting element depended on the pressure difference in the pneumatic spring and the additional tank. However, this model did not take into account the thermodynamic processes that will arise as a result of changes in body temperature and its heat transfer with the environment. This will affect the mass air flow through the connecting pipeline with certain geometric parameters and the rigidity of the pneumatic spring and the damping of the pneumatic system as a whole.

It should be noted that the reviewed works [6–18] report studies into the characteristics of the pneumatic spring suspension system based on equivalent mechanical and thermodynamic models.

So, an unresolved task is to assess the influence of the geometric parameters of the connecting pipeline on the dynamic characteristics of the pneumatic spring suspension system. At the same time, the mathematical model should provide for the possibility of taking into account different configurations of the pneumatic spring suspension system, heat transfer with the environment, and thermodynamic processes of changes in body temperature and, accordingly, mass air flow through the connecting pipeline.

3. The aim and objectives of the study

The aim of this work is to determine the influence of the geometric parameters of the connecting pipeline installed between the pneumatic spring and the additional tank on the characteristics of the pneumatic spring suspension system using a thermodynamic mathematical model. This will make it possible to more perfectly design a pneumatic spring suspension system that would be effective under different operating conditions of rolling stock.

To accomplish the aim, the following tasks have been set:

– to conduct theoretical studies into the influence of the diameter and length of the connecting element on the rigidity of the pneumatic spring;

– to estimate the influence of the diameter and length of the connecting element on the energy loss and damping coefficient for the operation cycle of the pneumatic spring suspension system;

– to compare the characteristics of the pneumatic spring suspension system during compression and expansion of the pneumatic spring.

4. Materials and methods of research

4. 1. Mathematical model of a flat system with two degrees of freedom

The estimation scheme of a dynamic model with two degrees of freedom (k=2) in kinematic perturbation is shown in Fig. 5.

During the development of the estimation scheme, the following prerequisites are accepted:

– the centers of mass of solids coincide with their geometric centers;

– the stiffness of the journal stage is considered nominal;

– stiffness in wheel and rail contact is not taken into account;

– the track is considered absolutely rigid;

– the wheelset and the track mass interacting with it move continuously.

Geometric irregularities of the rail track are taken as perturbations and jumping fluctuations are considered.

During the movement of the system, kinematic perturbation causes vertical oscillations of the structure over the suspension, which are characterized by two generalized coordinates z_k and z_b (bouncing).

Using D'Alembert's principle, the oscillation equation of the analyzed dynamic model:

$$\begin{cases} m_k \cdot \ddot{z}_k - P_1(t) \cdot A_1 + m_k \cdot g = 0, \\ m_b \cdot \ddot{z}_b + P_1(t) \cdot A_1 + k \cdot \left(z_b - z_{w.p.}\right) = 0, \end{cases} \tag{1}$$

where m_k is the body weight reduced to one spring, kg; m_b – the mass of the bogie reduced to one spring, kg; P_1 – pressure in the pneumatic spring, Pa; A_1 – effective pneumatic spring area, m^2; k – equivalent stiffness of the journal stage of spring suspension, N/m; g – acceleration of gravity, m/s^2.

Solving the equation of oscillations will make it possible to obtain the values of vertical displacements, velocities, and accelerations of mass m_k and m_b and further evaluate the dynamic properties of the model.

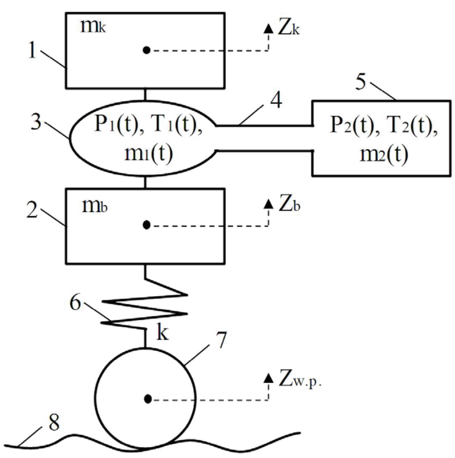

Fig. 5. Estimation scheme of a dynamic model with two degrees of freedom: 1 – body; 2 – bogie; 3 – pneumatic spring; 4 – connecting pipeline; 5 – additional tank; 6 – spring of the journal stage of spring suspension; 7 – wheel set; 8 – vertical irregularities of the rail track

4. 2. Mathematical model of the pneumatic spring suspension system

The air inside the pneumatic spring suspension system was considered as an ideal gas since the relative pressure and temperature arising during the movement make it possible to adopt assumptions about the behavior of the gas inside the spring as ideal.

Considering the ideal gas equation for the volume of the pneumatic spring and differentiating by time, we get:

$$\frac{dP(t)}{dt}V(t)+\frac{dV(t)}{dt}P(t)= \\ =\frac{dm(t)}{dt}RT(t)+\frac{dT(t)}{dt}m(t)R, \tag{2}$$

$$\dot{P}(t)=-\dot{h}(t)\frac{P(t)}{h(t)}+\dot{m}(t)\frac{RT(t)}{h(t)A_1}+\dot{T}(t)\frac{m(t)R}{h(t)A_1}, \tag{3}$$

where P, V, T – respectively, pressure, volume, and temperature of the working fluid of the pneumatic spring, Pa; m^3; K; m – air mass, kg; R – universal gas constant, J/(kg·K); h – current height of the pneumatic spring, m.

Since the pneumatic spring is connected to an additional tank through the connecting pipeline, the continuity equation will take the form:

$$\frac{dm_1(t)}{dt}=-\dot{m}(t), \tag{4}$$

$$\frac{dm_2(t)}{dt}=\dot{m}(t).$$

Using the Bernoulli equation, which determines the relationship between the flow velocity v(t), the pressure $p(t)$, and the height h of a certain point in the liquid, the mass flow of air through the connecting pipeline with certain geometric parameters was found:

$$\dot{m}(t)=\rho(t)A_{per}\sqrt{\frac{2\Delta P(t)}{\rho(t)\left(f\frac{l}{d}+K_s+K_p\right)}}, \tag{5}$$

where ρ is the density of the working fluid, kg/m^3; A_{per} – cross-sectional area of the connecting pipeline, m^2; l, d – length and diameter of the connecting pipeline, m; K_s – compression loss ratio; K_p – coefficient of losses on expansion.

Given that the flow of air through the connecting pipeline is turbulent in nature, an equation was used that takes into account the loss of pressure due to flow constraints. Pressure losses consisted of three components, namely friction losses, which were determined through the Darcy-Weisbach equation, as well as local losses due to instantaneous narrowing and expansion of the flow.

Since the pneumatic spring suspension system is closed, in its energy balance it is necessary to take into account the transfer of energy between the pneumatic spring and the additional tank.

So, the law of conservation of energy, taking into account the heat transfer between the pneumatic spring and the environment, as well as the transfer of energy between the pneumatic spring and the additional tank will take the form:

$$\Delta U(t)=\Delta Q(t)-\Delta W(t)+\Delta E(t), \tag{6}$$

where U is the internal energy of the pneumatic spring, J; Q – heat transfer, W; W – work done, J; E – energy transfer between the pneumatic spring and the additional tank, J.

Changes in internal energy and energy transfer between the pneumatic spring and the additional tank, heat transfer, and limit work were found according to the formulas:

$$\Delta U = m \cdot C_v \cdot \Delta T(t), \tag{7}$$

$$\Delta Q(t)=h_T \cdot A_s(t)\cdot\left(T_s - T(t)\right)\Delta t, \tag{8}$$

$$\Delta W(t)=F(t)\cdot\Delta h(t)=A_1\cdot P(t)\cdot\Delta h(t), \tag{9}$$

$$\Delta E(t)=\Delta m(t)\cdot C_p\cdot T(t), \tag{10}$$

where Δh is the deviation of the height of the pneumatic spring from the state of equilibrium, m; A_s – heat transfer area, m^2; h_T – heat transfer coefficient, W/(m^2·K); C_v – specific heat capacity at a constant volume, J/(kg·K).

According to a similar principle, an equation was built for an additional tank, taking into account the absence of heat transfer with the environment and changes in volume.

4. 3. Mathematical model for determining the characteristics of the pneumatic spring suspension system

Evaluation of the characteristics of the pneumatic spring suspension system was carried out with the initial data specified in Table 1.

The main geometric parameters of the connecting pipeline, the influence of which on the characteristics of the pneumatic spring suspension system will be estimated in this paper, are its diameter and length (Fig. 6).

Table 1

Initial data for determining the characteristics of the pneumatic suspension system

Name		Designation	Value
Concentrated mass, kg:	body	m_k	10,900
	bogie	m_b	3,350
Initial pressure in pneumatic spring, Pa		P_1	470,000
Effective pneumatic spring area, m^2		A_1	0.2275
Equivalent stiffness of the journal stage of spring suspension, N/m		k	1,136,000
Initial height of the pneumatic spring during the modeling process, m		h	0.2
Geometric parameters of the connecting pipeline:	length, m	l	0.4...2.0
	diameter, mm	d	6.0...30.0
Compression loss ratio		K_s	0.5
Expansion loss ratio		K_p	1.0
System speed, m/s		v	30
Amplitude of vertical irregularity, m		H	0.015
Length of vertical irregularity, m		L	10

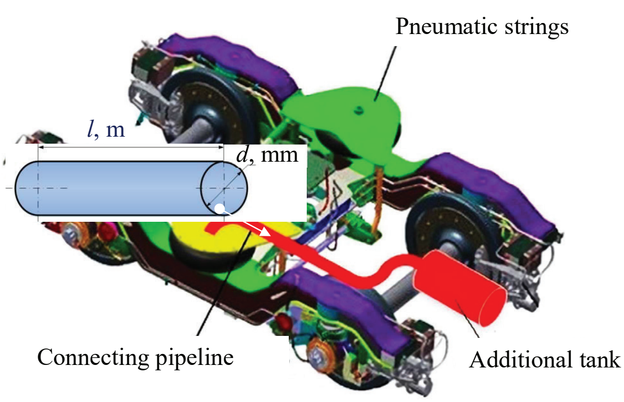

Fig. 6. The main structural units in the bogie of the diesel train DPKr-2

Determining the dynamic stiffness of the pneumatic spring preliminarily involves obtaining the dependence of the force on deformation. From the resulting “force-deformation” dependence, rigidity is determined by the formula:

$$S = \frac{F_0}{x_0}, \tag{11}$$

where F_0 is the amplitude of the force, kN; x_0 – amplitude of deformation, m.

The damping coefficient, which takes into account energy losses for the cycle of operation of the pneumatic spring suspension system, is determined by the formula:

$$D = \frac{E}{F_0 \cdot x_0}, \tag{12}$$

where E – energy dissipation per cycle, J.

So, according to these geometric parameters, we shall evaluate the features of their influence on the characteristics of the pneumatic spring suspension system.

5. Investigation of the characteristics of the pneumatic spring suspension system

5. 1. Evaluation of the influence of the geometric parameters of the connecting pipeline on the rigidity of the pneumatic spring

During the research, numerical stiffness values were found (Table 2), on the basis of which the dependence of the dynamic rigidity of the pneumatic spring on the diameter of the connecting pipeline at its different lengths was built (Fig. 7).

Table 2

Values of the dynamic stiffness of the pneumatic spring during the compression process

Length, m	Diameter of the connecting pipeline, mm						
	6	10	14	18	22	26	30
0.4	924	885	806	686	583	515	496
0.6	922	882	819	701	595	518	496
0.8	923	889	825	712	604	522	494
1	928	890	833	727	612	527	497
1.2	927	894	837	735	619	529	499
1.4	923	900	842	736	627	536	500
1.6	931	900	853	746	629	540	501
1.8	933	901	854	748	639	544	504
2.0	932	898	858	756	648	549	506

With a diameter of 6 mm or less, there is a constant stiffness value of the pneumatic spring for each individual value of the length of the connecting pipeline. With a length of 1 m, the stiffness is 928 kN/m. This feature is that the pneumatic spring in this case works as a separate elastic element and there is practically no air flow between the pneumatic spring and the additional tank.

With a connecting pipeline diameter of 30 mm or more, this feature is repeated, but in this case the entire pneumatic spring suspension system acts as an elastic element. In this case, with a connection pipeline length of 1 m, the stiffness of the pneumatic spring is 497 kN/m.

Consequently, with a connection pipeline length of 1 m and a diameter of 6 mm to 30 mm, the rigidity of the pneumatic spring will decrease from 925 kN/m to 497 kN/m, which as a percentage will be 46 %. It is in this range that the pneumatic spring suspension system will work as an elastic-viscous element that will correspond to its purpose.

Fig. 7 demonstrates that with an increase in the diameter of the connecting pipeline, at any length of it, the dependence of stiffness is nonlinear.

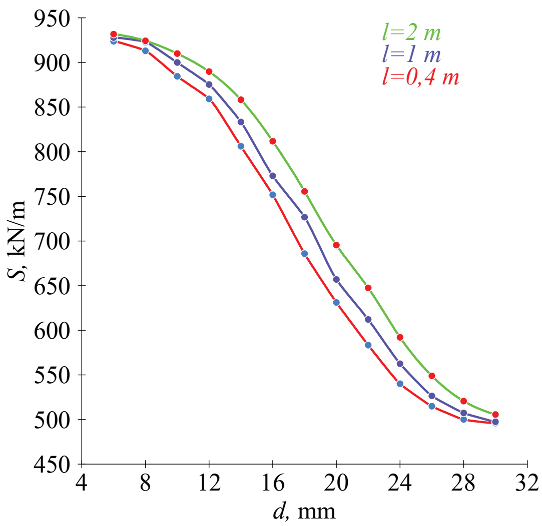

Fig. 7. Dependence of the dynamic rigidity of the pneumatic spring on the diameter of the connecting pipeline at different values of its length

In addition, analyzing Fig. 8, it was found that an increase in the length of the connecting pipeline leads to an increase in the stiffness of the pneumatic spring. It is observed that with a diameter of the connecting pipeline of 6 mm or less, the difference in stiffness at different lengths will be no more than 1 %. However, with increasing diameter, this difference increases and reaches its maximum value with a diameter of 22 mm and will be no more than 10 %. With a diameter of more than 22 mm, this difference decreases again and will be no more than 2 % with a diameter of the connecting pipeline of 30 mm.

5. 2. Evaluation of the influence of geometric parameters of the connecting pipeline on energy loss and damping coefficient

When performing research, the value of energy losses for the cycle of operation of the pneumatic spring suspension system was derived (Table 3). On the basis of this, the dependences of the damping coefficient of the pneumatic spring suspension system are constructed, depending on the diameter and length of the connecting pipeline (Fig. 8).

Analyzing Table 3, we established that the maximum values of energy losses for the cycle of operation of the pneumatic spring suspension system are observed with a diameter of the connecting pipeline of 20 mm, at any length of it. With a connection pipeline length of 0.6 m, the maximum value of energy losses per cycle of operation of the pneumatic spring suspension system is 620 J, with a length of 1 m – 587 J, with a length of 1.4 m – 575 J.

Fig. 9 shows that the damping coefficient of the pneumatic spring suspension system with an initial increase in the diameter of the connecting pipeline increases at any length. This is mainly due to an increase in mass air flow through the connecting pipeline and, accordingly, an increase in pressure losses.

However, with a diameter of the connecting pipeline of 18 mm, the damping coefficient reaches its maximum values and is 0.87, after which it begins to decrease. The decrease in the damping coefficient is explained by a decrease in energy losses during the operation cycle of the spring suspension system. It was established that the maximum difference in the damping coefficient when changing the length of the connecting pipeline from 0.6 m to 1.4 m is no more than 15 % of its maximum value.

Table 3

Values of energy losses for the cycle of operation of the pneumatic spring suspension system

Diameter, mm	The length of the connecting pipeline, m		
	0.6	1	1.4
6	65	56	50
10	202	187	163
14	418	378	355
18	549	536	529
20	620	587	575
22	592	579	573
26	448	438	542
30	339	326	374

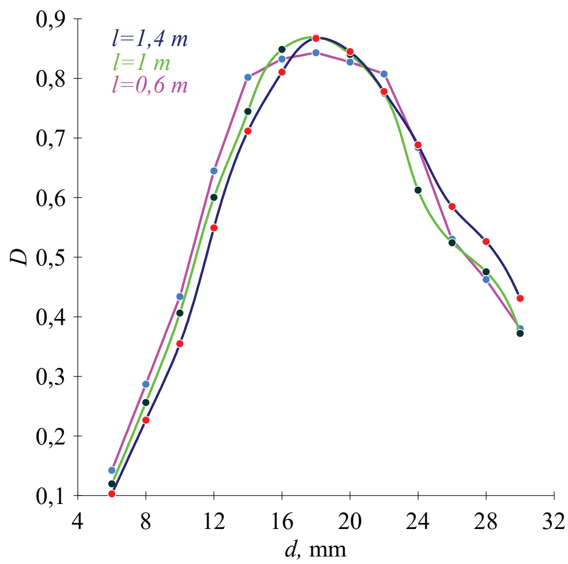

Fig. 8. Dependence of the damping coefficient on the diameter of the connecting pipeline at different lengths

5. 3. Comparative assessment of the characteristics of the pneumatic spring suspension system during compression and expansion

Fig. 9 shows the results of comparing the dynamic rigidity of the pneumatic spring, and in Fig. 11 results of comparison of the damping coefficient of the pneumatic spring suspension system during compression and expansion.

Fig. 9 shows that the rigidity of the pneumatic spring is different when it is compressed and expanded. Taking into account the average geometric parameters of the connecting pipeline, it was established that with a diameter of 20 mm and its length of 1 m, the rigidity of the pneumatic spring during compression is 657 kN/m, and at expansion – 430 kN/m. Difference is 227 kN, which in percentage ratio is the difference of 34.55 %. Obtaining such a difference indicates the need to take this factor into account when designing high-speed rolling stock.

It was found that an increase in the diameter of the connecting pipeline gradually leads to a decrease in the stiffness difference of the pneumatic spring during compression and expansion. The biggest difference is with a diameter of 6 mm and is 334 kN/m, and the smallest with a diameter of 30 mm and is 164 kN/m.

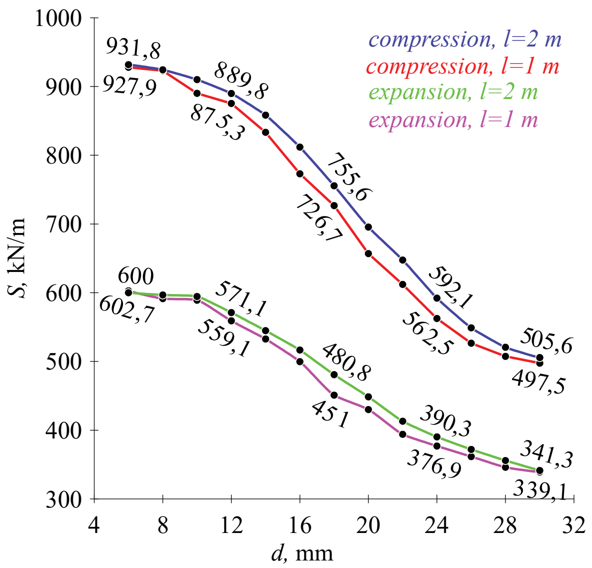

Fig. 9. Stiffness of the pneumatic spring during its compression and expansion

It was also found that a change in the length of the connecting pipeline from 0.4 m to 2 m leads to a change in the difference in stiffness during compression and expansion of not more than 19 %.

Analyzing Fig. 10, it was found that the difference between the damping coefficient in the process of compression and expansion of the pneumatic spring is not more than 4 %.

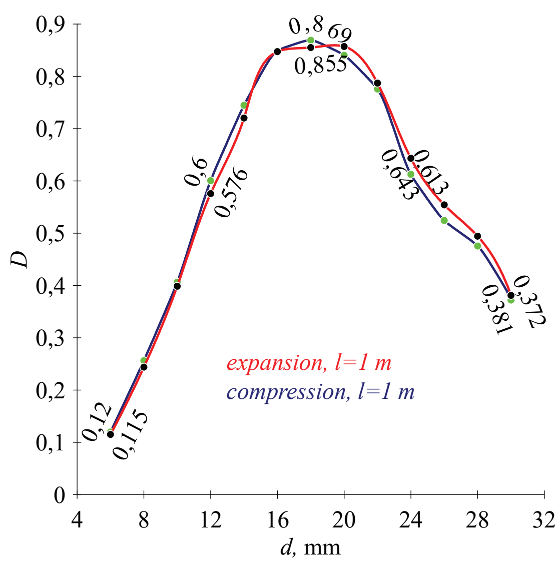

Fig. 10. Damping coefficient of the pneumatic spring suspension system during compression and expansion of the pneumatic spring

Having built the dependence of the dynamic rigidity of the pneumatic spring and the damping coefficient of the pneumatic spring suspension system, an effective zone of operation of the pneumatic spring was found (Fig. 11). In this case, the specified zone will depend on the initial data of the mathematical model.

Fig. 11 it demonstrates that the effective operation of the pneumatic spring suspension system will occur with a diameter of the connecting pipeline from 14 mm to 23 mm. At the same time, dynamic rigidity varies from 377 kN/m to 870 kN/m, and the damping factor is from 0.66 to 0.87.

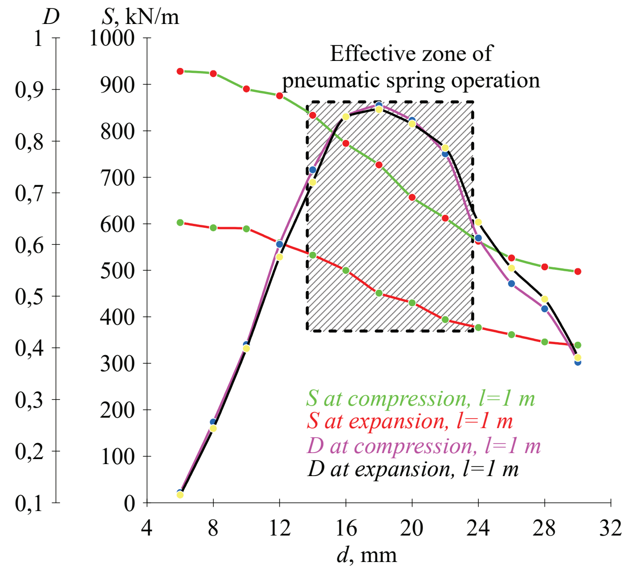

Fig. 11. Pneumatic spring efficient operation area

6. Discussion of results of the effect of geometric parameters of the connecting pipeline on the characteristics of the pneumatic spring

The influence of the diameter and length of the connecting pipeline on the rigidity of the pneumatic spring, energy loss, and damping coefficient for the cycle of operation of the pneumatic spring suspension system were investigated.

To perform the research, an estimation scheme of a dynamic model with two degrees of freedom was used (Fig. 5), which makes it possible to set the perturbation of the system with a certain frequency and take into account the main structural elements of the rolling stock.

It was found that with a diameter of the connecting pipeline less than 6 mm and more than 30 mm, the rigidity of the pneumatic spring is constant and is 928 kN/m and 497 kN/m, respectively (Table 2). Between these diameters, the rigidity of the pneumatic spring is dynamic, and the pneumatic spring itself is an elastic-viscous element. Comparison of the stiffness values of the pneumatic spring during its compression and expansion showed different values, the difference of which, with an increase in the diameter of the connecting element, decreases by 51 % with respect to the greatest difference (Fig. 9). An increase in the length of the connecting pipeline leads to an increase in the rigidity of the pneumatic spring. At different lengths, the difference in stiffness is no more than 10 %.

It was established that the greatest energy dissipation for the cycle of operation of the pneumatic spring suspension system with a connection pipeline length of 1 m occurs with a diameter of 20 mm and is 587 J. Then, with an increase in the diameter of the connecting pipeline from 6 mm to 20 mm, the scattering increases from 56 J to 587 J. With an increase in the diameter of the connecting pipeline of more than 20 mm, the opposite effect occurs, due to a decrease in the resistance of losses in the connecting pipeline. With an increase in the length of the connecting pipeline from 0.6 m to 1.4 m with a constant diameter of 20 mm, a decrease in energy dissipation by 3.2 % is observed.

The damping factor of the pneumatic spring suspension system with an increase in the diameter of the connecting pipeline also first increases to its maximum value, after which it begins to decrease. The maximum difference in the damping coefficient when changing the length of the connecting pipeline from 0.6 m to 1.4 m is no more than 15 % of its maximum value. The difference between the damping coefficient in the process of compression and expansion of the pneumatic spring is not more than 4 %.

Effective operation of the pneumatic spring suspension system occurs with a diameter of the connecting pipeline from 14 mm to 23 mm (Fig. 11).

The results of our research have made it possible to establish the dependence of the main characteristics of the pneumatic spring hanging system on the geometrical parameters of the connecting pipeline. This complements the studies reported in works [2–14]. Our results make it possible to reasonably choose the geometrical parameters of the connecting pipeline under the conditions of design and subsequent operation of rolling stock. The obtained results of rigidity, energy losses, and damping coefficient of the pneumatic spring suspension system can have practical application. This regards the possibility of their use by engineers and researchers in the study of dynamic indicators of rolling stock at the design stage.

The disadvantage of the study is the determination of the characteristics of the pneumatic spring suspension system without taking into account the effect of the rigidity of the rubber band of the pneumatic spring. However, this feature has a slight impact that can be neglected.

One of the limitations of the research is the determination of the characteristics of the pneumatic spring suspension system only from the geometric parameters of the connecting pipeline. Therefore, a continuation of research implies investigating the amplitude-frequency characteristics of the pneumatic spring suspension system under the action of dynamic loads.

7. Conclusions

1. The results of theoretical studies have shown that with a connection pipeline length of 1 m, an increase in its diameter from 6 mm to 30 mm leads to a decrease in the dynamic rigidity of the pneumatic spring from 925 kN/m to 497 kN/m, which as a percentage is 46 %. The maximum difference in the stiffness of the pneumatic spring with the lengths of the connecting pipeline from 0.4 m to 2.0 m is no more than 10 % with a diameter of 22 mm.

2. The influence of the diameter and length of the connecting pipeline on energy loss and damping coefficient for the operation cycle of the pneumatic spring suspension system has been investigated. It was established that the maximum values of the studied parameters are achieved with a pipeline diameter of 18–22 mm and equal to 620 J and 0.87, respectively. The maximum difference in the damping coefficient when changing the length of the connecting pipeline from 0.6 m to 1.4 m is no more than 15 % of its maximum value.

3. The results of comparing the characteristics of the pneumatic spring suspension system during compression and expansion showed that with a connecting pipeline diameter of 20 mm and its length of 1 m, the difference is 34.55 %. Changing the length of the connecting pipeline from 0.4 m to 2 m leads to a difference in stiffness between compression and expansion of not more than 19 %. Effective operation of the pneumatic spring suspension system occurs when the dynamic rigidity changes in the range from 377 kN/m to 870 kN/m and the damping coefficient from 0.66 to 0.87.

Conflicts of interest

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

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Data availability

All data are available in the main text of the manuscript.

References

1. Proekt planu vidnovlennia Ukrainy. Materialy robochoi hrupy «Vidnovlennia ta rozbudova infrastruktury» *(2022)*. Natsionalna rada z vidnovlennia Ukrainy vid naslidkiv viyny, 178. Available at: https://www.kmu.gov.ua/storage/app/sites/1/recoveryrada/ua/restoration-and-development-of-infrastructure.pdf
2. Kuzyshyn, A., Batig, A., Kostritsa, S., Sobolevska, J., Kovalchuk, V., Dovhanyuk, S., Voznyak, O. (2018). Research of safety indicators of diesel train movement with two-stage spring suspension. MATEC Web of Conferences, 234, 05003. doi: https://doi.org/10.1051/matecconf/201823405003
3. Kuzyshyn, A., Batig, A., Kostritsa, S., Sobolevska, J., Dovhaniuk, S., Dzhus, V. (2020). Study of the dynamic behavior of rolling stock using a computer experiment. IOP Conference Series: Materials Science and Engineering, 985 (1), 012002. doi: https://doi.org/10.1088/1757-899x/985/1/012002
4. Kovalchuk, V., Kuzyshyn, A., Kostritsya, S., Sobolevska, Y., Batig, A., Dovganyuk, S. (2018). Improving a methodology of theoretical determination of the frame and directing forces in modern diesel trains. Eastern-European Journal of Enterprise Technologies, 6 (7 (96)), 19–26. doi: https://doi.org/10.15587/1729-4061.2018.149838

5. Kuzyshyn, A., Kostritsa, S., Ursulyak, L., Batig, A., Sobolevska, J., Voznyak, O. (2019). Research of the impact of geometric unevenness of the railway track on the dynamic parameters of the railway rolling stock with two-stage spring suspension. IOP Conference Series: Materials Science and Engineering, 664 (1), 012024. doi: https://doi.org/10.1088/1757-899x/664/1/012024
6. Moheyeldein, M. M., Abd-El-Tawwab, A. M., Abd El-gwwad, K. A., Salem, M. M. M. (2018). An analytical study of the performance indices of air spring suspensions over the passive suspension. Beni-Suef University Journal of Basic and Applied Sciences, 7 (4), 525–534. doi: https://doi.org/10.1016/j.bjbas.2018.06.004
7. Alonso, A., Giménez, J. G., Nieto, J., Vinolas, J. (2010). Air suspension characterisation and effectiveness of a variable area orifice. Vehicle System Dynamics, 48, 271–286. doi: https://doi.org/10.1080/00423111003731258
8. Matsumiya, S., Nishioka, K., Nishimura, S., Suzuki, M. (1969). On the diaphragm air spring sumride. The Sumitomo Search, 2, 86–92.
9. Berg, M. (1999). A Three–Dimensional Airspring Model with Friction and Orifice Damping. Vehicle System Dynamics, 33, 528–539. doi: https://doi.org/10.1080/00423114.1999.12063109
10. Sayyaadi, H., Shokouhi, N. (2010). Effects of air reservoir volume and connecting pipes' length and diameter on the air spring behavior in rail-vehicles. Iranian Journal of Science and Technology Transaction B: Engineering, 34 (B5), 499–508.
11. Xu, L. (2020). Mathematical Modeling and Characteristic Analysis of the Vertical Stiffness for Railway Vehicle Air Spring System. Mathematical Problems in Engineering, 2020, 1–12. doi: https://doi.org/10.1155/2020/2036563
12. Nieto, A. J., Morales, A. L., González, A., Chicharro, J. M., Pintado, P. (2008). An analytical model of pneumatic suspensions based on an experimental characterization. Journal of Sound and Vibration, 313 (1-2), 290–307. doi: https://doi.org/10.1016/j.jsv.2007.11.027
13. Sowayan, A. (2020). Stiffness of a Nonlinear Adiabatic Polytropic Air Spring Model: Quantitative and Conductive Investigation. International Review on Modelling and Simulations (IREMOS), 13 (3), 177. doi: https://doi.org/10.15866/iremos.v13i3.18761
14. Zhu, H., Yang, J., Zhang, Y., Feng, X. (2017). A novel air spring dynamic model with pneumatic thermodynamics, effective friction and viscoelastic damping. Journal of Sound and Vibration, 408, 87–104. doi: https://doi.org/10.1016/j.jsv.2017.07.015
15. Chen, J.-J., Yin, Z.-H., Rakheja, S., He, J.-H., Guo, K.-H. (2017). Theoretical modelling and experimental analysis of the vertical stiffness of a convoluted air spring including the effect of the stiffness of the bellows. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 232 (4), 547–561. doi: https://doi.org/10.1177/0954407017704589
16. Gao, H. X., Chi, M. R., Zhu, M. H., Wu, P. B. (2013). Study on Different Connection Types of Air Spring. Applied Mechanics and Materials, 423-426, 2026–2034. doi: https://doi.org/10.4028/www.scientific.net/amm.423-426.2026
17. Reidemeister, O. H., Kivisheva, A. V. (2016). Dependence of air spring parameters on throttle resistance. Science and Transport Progress. Bulletin of Dnipropetrovsk National University of Railway Transport, 2 (62), 157–164. doi: https://doi.org/10.15802/stp2016/67339
18. Reidemeister, O. H., Laguza, A. V. (2018). Determination of characteristics of throttling device for pneumatic spring. Science and Transport Progress. Bulletin of Dnipropetrovsk National University of Railway Transport, 1 (73), 66–76. doi: https://doi.org/10.15802/stp2018/123394