Studying the diagrams "force - deformation" of a pneumatic spring of a modern rolling stock at increased speeds

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Abstract. The research object is a pneumatic spring of a modern rolling stock. Using a two - mass calculation scheme, the diagrams "force-deformation" of a pneumatic spring are designed at driving speed of 160-200 km/h and high-speed driving of 201-250 km/h. It is determined that when the diameter of the connecting pipeline is changed from 20 to 35 mm, the inclination angle of the diagram "force - deformation" of the pneumatic spring is changed at speed driving from 32 degrees up to 57 deg.; at high-speed driving from 30 deg. up to 48 deg. It is proposed to determine a pneumatic spring operation coefficient using the forms of the designed diagrams "force - deformation." Studying the specified coefficient in the speed range from 160 to 250 km/h, and the connecting pipeline diameter from 20 to 35 mm showed that the values of the pneumatic spring operation coefficient vary from 0.089 to 0.24, and the dependencies are nonlinear. Using the pneumatic spring operation coefficient will make it possible to further evaluate the pneumatic spring suspension system operation depending on the operating conditions of the railway vehicle.

1 INTRODUCTION

For the safe operation of a rolling stock in conditions of speed and high-speed driving, the main task is to observe its dynamic safety indicators within acceptable limits. For this purpose, a pneumatic system is used as the second stage of spring suspension of a modern rolling stock, which includes pneumatic springs connected through a connecting pipeline to an additional tank (fig. 1).

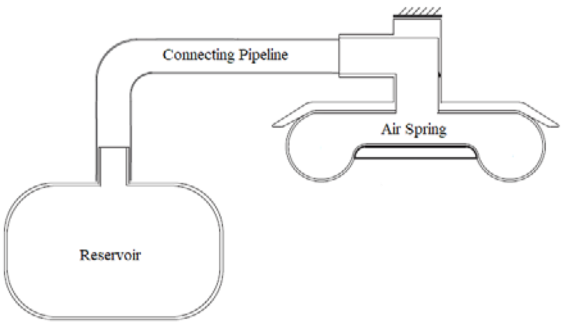

Fig. 1. Pneumatic spring suspension system

In Ukraine, there are several types of a rolling stock that have pneumatic spring suspension systems, namely: Hyundai Rotem and ЕКр-1 "Tarpan" electric trains, as well as ДПКр-2 and ДПКр-3 diesel trains (fig. 2).

The main characteristics of a pneumatic spring suspension system are its rigidity and damping coefficient. They are these characteristics that mainly determine the level of dynamic forces in the links between the structural elements of a rolling stock, which arise due to the interaction of the wheel set with the rail track.

Fig. 2. General view of the ДПКр-3 diesel train

Taking into account the prospects for introducing the speed and high-speed driving in the railway transport of Ukraine, the question of studying the characteristics of the pneumatic spring suspension system at the design stage of rolling stock is relevant. This allows analyzing and establishing safe operating conditions for rolling stock at high speeds, as well as various operating conditions, including the technical condition of both structural elements of rolling stock and railway tracks.

2 Analysis of references

While staying on a moving train, passengers and staff are exposed to vibrations and forces that occur when a rolling stock interacts with the track in straight and curved sections [1-5].

The pneumatic spring suspension system used on a modern rolling stock demonstrates high shock-absorbing

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and damping properties that allow providing comfortable and safe conditions for passenger transportation [6-7].

In [8], the authors investigated the stiffness and hysteresis of a pneumatic spring. First, a thermodynamic model was presented, which included heat transfer between the air inside the spring and the environment. The mathematical model included nonlinear differential equations with time-dependent volume and pressure functions. It is noted that the equation can be simplified to the law of mechanical force-displacement. As a result of linearizing the equation, a mechanical model is obtained that can be described by elastic and damping elements in a certain location (fig. 3).

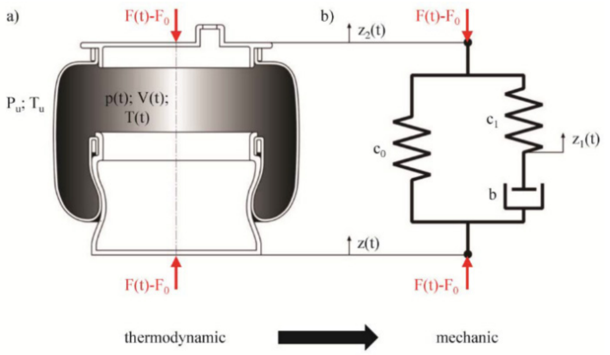

Fig. 3. Comparison of thermodynamic (A) and mechanical (B) models for an air spring [8]

However, the work did not take into account the design features of the pneumatic spring suspension system, namely the presence of a connecting pipeline with its geometric parameters and an additional tank.

In [9], the authors conducted an analytical study of the performance indicators of air suspensions. For this purpose, two different models of pneumatic springs are considered: classical and dynamic (fig. 4).

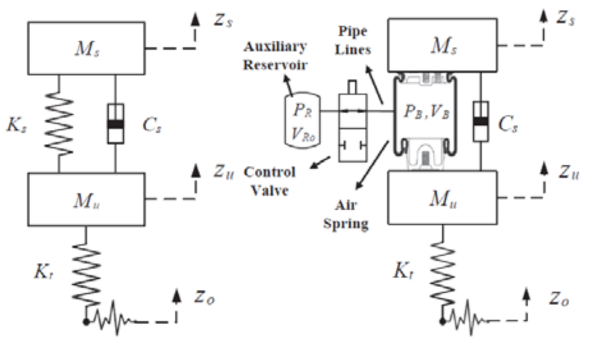

Fig. 4. Classic and dynamic models of the pneumatic spring suspension system [9]

In [10], the author conducted a study of the transverse rigidity of a pneumatic spring suspension system. The model for studying lateral vibrations consists of three parallel elements: the stiffness of the rubber diaphragm; the stiffness of the emergency spring and the damping coefficient (fig. 5), which are based on the laws of thermodynamics, hydrodynamics, and Newton's law.

However, the studies are conducted for driving speeds from 20 to 200 km/h.

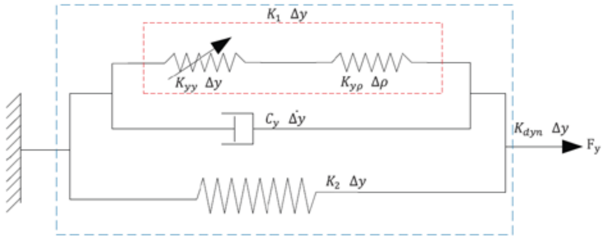

Fig. 5. The nonlinear model for studying the transverse stiffness of a pneumatic spring system [10]

In [11], the author proposed a nonlinear model of a pneumatic spring for general three-dimensional movements (fig. 6).

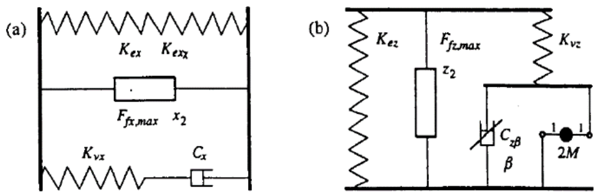

Fig. 6. The pneumatic spring model and model parameters. (a) – horizontal driving, six parameters; (b) - vertical driving, seven parameters [11]

The nonlinearity of the model is primarily due to the airflow in the connecting pipeline between the pneumatic spring and the additional tank. The model has 6 and 7 parameters for horizontal and vertical displacements, respectively. In this case, the stiffness of the pneumatic spring is defined as the amplitude of the dynamic force divided by the displacement amplitude. Damping is defined as the energy dissipation per cycle divided by the product of the dynamic force amplitude and the displacement amplitude. Based on the studies, the dependencies "force-deformation" are found in the frequency range from 0 to 16 Hz.

In [12], the authors presented an analytical model of a pneumatic spring suspension system based on the experimental characteristics. Based on the conducted research, it was shown that the dynamic behaviour of an air suspension can be made more versatile by conveniently choosing the dimensions of its elements, in particular the volumes of the air spring and tank. On the one hand, reducing the volume of the pneumatic spring increases the stiffness as well as the highest natural frequency. On the other hand, increasing the tank volume reduces the stiffness and the lowest eigenfrequency.

Therefore, by implementing these changes in the suspension elements, you can increase the difference between these two eigenfrequencies.

In [13], the authors characterized the mechanical behaviour of a particular rubber spring for high-speed railway vehicles. For this purpose, the mathematical model consisted of three parallel elements: a nonlinear elastic spring, a friction element, and a viscous Maxwell component.

However, these studies are relevant for determining only a separate component of the stiffness of a pneumatic spring.

In [14], a dynamic model of a pneumatic spring suspension system is developed, taking into account the thermodynamic processes in the pneumatic spring-pipeline-reservoir system, effective friction, and viscoelastic rubber damping (fig. 7). Note that the parameters in the friction model depend on the standard deviation using the statistical method, and not on the constant values in the classical Berg friction model. The property of viscoelastic rubber is modelled with only two parameters.

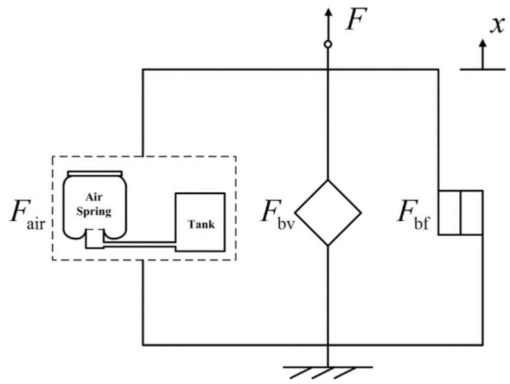

Fig. 7. Mechanical model of pneumatic spring suspension system [14]

At the same time, the proposed model parameters are determined and additionally checked by bench tests of the pneumatic spring, pipeline system and additional tank separately.

Thus, in the considered works, studies of the pneumatic spring suspension system operation have not been carried out, taking into account the properties of bindings between the structural elements of rolling stock, as well as the influence of geometric parameters of the connecting pipeline on the dynamic rigidity of the pneumatic spring in conditions of speed and high-speed driving.

The obtained diagrams of the operation of the pneumatic spring allow designers, even at the design stage, to study various operating conditions of rolling stock and determine its dynamic indicators and driving safety indicators in conditions of speed and high-speed driving.

3 Purpose and objectives of the study

The purpose of the work is to build and study diagrams of the operation of a pneumatic spring with an additional tank in conditions of increased driving speeds. To achieve this goal, you must complete the following tasks:

– construct the diagrams "force - deformation" of a pneumatic spring with different diameters of the connecting pipeline in speed and high-speed driving conditions;

– set the value of the angle of inclination of the diagrams "force - deformation" of the pneumatic spring when changing the diameter of the connecting pipeline in conditions of increased rolling stock speeds;

– set the coefficients of operation of the pneumatic spring under different operating conditions of a rolling stock according to the parameters of the diagram "force - deformation".

4 Research Methodology

To study the operation of a pneumatic spring suspension system, taking into account the peculiarities of the interaction of elements of the rolling stock-track system, we will design a simplified calculation scheme of rolling stock as a system with two degrees of freedom (fig. 8) [15].

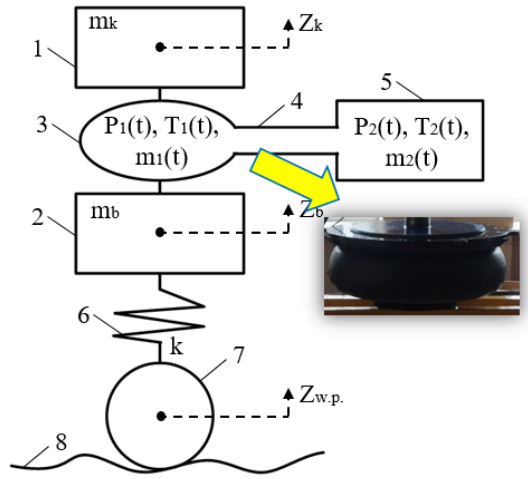

Fig. 8. Calculation scheme of a dynamic model with two degrees of freedom: 1 - body; 2 - trolley; 3 - pneumatic spring; 4 - connecting pipeline; 5 - additional tank; 6 - spring of the box stage of spring suspension; 7 - wheelset wheel; 8 - vertical irregularities of the rail track [15]

Using the D'alembert principle, we write down the oscillation equations of the 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 - body weight reduced to one spring, kg; m_b - trolley weight reduced to one spring, kg; P_1 - pressure in the pneumatic spring, Pa; A_1 - effective area of the pneumatic spring, m^2; k - equivalent stiffness of the spring suspension axle box stage, H/m; g - acceleration of gravity, m/s^2.

The pneumatic spring and additional tank in the working position are filled with compressed air, which provides the necessary rigidity and damping. Since the relative pressure and temperature are within the limits where the ideal behaviour of the gas can be assumed, the equation of state of the ideal gas is used to find the internal pressure of the air spring:

$$PV = mRT \tag{2}$$

where P, V, T - respectively, the pressure, volume and temperature of the working fluid of the pneumatic spring, Pa; m^3; K; m - Air weight, kg; R - universal gas steel, J/(kg·K).

Differentiating Equation (2) over time, we obtain:

$$\frac{dP(t)}{dt}V(t) + \frac{dV(t)}{dt}P(t) = \frac{dm(t)}{dt}RT(t) + \frac{dT}{dt}m(t)R \tag{3}$$

Knowing the average speed of the working body and performing analytical transformations, we obtain a formula for finding the mass flow rate:

$$\dot{m}(t) = \rho(t) A_{per} \sqrt{\frac{2\Delta P(t)}{\rho(t)\left(f\frac{l}{d} + K_s + K_p\right)}} \qquad (4)$$

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 - expansion loss ratio.

To find the temperature of the working body, we use the law of energy conservation, taking into account the heat transfer between the pneumatic spring and the environment, as well as the energy transfer between the pneumatic spring and the additional tank:

$$U(t) = Q(t) - W(t) + E(t) \qquad (5)$$

$$\Delta U = mC_v \Delta T(t) \qquad (6)$$

$$\Delta Q(t) = h_T A_s(t)(T_s - T(t))\Delta t \qquad (7)$$

$$\Delta W(t) = F(t)\Delta h(t) = A_p P(t)\Delta h(t) \qquad (8)$$

where U - internal energy of the air spring, J; W - work done, J; Q - heat transfer, W; E - Energy Transfer between the pneumatic spring and the additional tank, J; Δh - deviation of the height of the pneumatic spring from the equilibrium state, m; h_T - heat transfer coefficient, W/(m^2·K); A_s - heat transfer area, m^2; C_v - specific heat capacity at constant volume, J / (kg·K).

The equation for an additional tank is formed on a similar principle. Heat transfer with the environment and volume changes are considered absent.

5 The obtained results

Construction and research of diagrams of the pneumatic spring suspension system operation are carried out with the initial data indicated in Table. 1.

The MATLAB mathematical environment is used to calculate the mathematical model.

When performing research diagrams of the "force – deformation" of a pneumatic spring at a speed of 160-200 km/h and high-driving speed 201-250 km/h are designed (fig. 9-10).

Analysis of the diagram in high-speed driving conditions showed that with a connecting pipeline length of 4 m and its diameter of 30 mm, the deformation of the pneumatic spring is in the range from 8 mm to 28 mm. In conditions of high driving speed, the deformation of the pneumatic spring is in the range from 4 mm to 8 mm.

According to the designed diagrams of "force-deformation" of the pneumatic spring, the angles of their inclination, which determine the dynamic stiffness of the spring are found (table. 2).

Table 1. Initial data for evaluating the pneumatic suspension system operation.

Name	Designation	Value
Concentrated mass, kg: - body - trolley	 m_k m_b	 17262 3350
Equivalent stiffness of the spring suspension axle box stage, H/m	k	1136000
Effective area of the pneumatic spring, m^2	A_1	0,2419
Geometric parameters of the connecting pipeline: - length, m - diameter, mm	 l d	 3,0…6,0 20,0…35,0
Compression loss ratio	K_s	0,5
Expansion loss ratio	K_p	1,0
System speed, km/h	ν	160-250
Amplitude of vertical inequality, m	H	0,015
Length of vertical unevenness, m	L	10

It is determined when the diameter of the connecting pipeline changes from 20 to 35 mm, the angle of inclination of the diagram "force-deformation" of the pneumatic spring will change: at high driving speed from 32 deg to 57 deg.; at high driving speed from 30 deg. up to 48 deg.

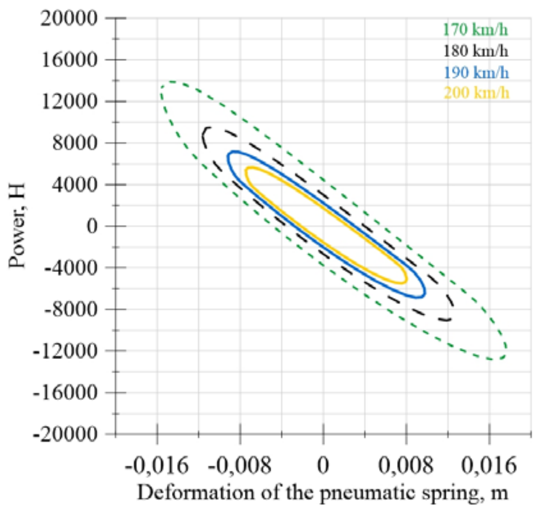

Fig. 9. Diagrams" force-deformation" of a pneumatic spring at l = 4 m, d = 30 mm in high driving speed conditions

Table 2. Inclination angles of the diagrams "force - deformation" of a pneumatic spring in driving speed and high-speed conditions

V, km/h	Diameter of the connecting pipeline, mm			
	20	25	30	35
160	57	54	46	38
180	50	46	39	34
200	48	43	37	32
220	47	42	36	31
240	46	41	35	30
250	46	40	34	30

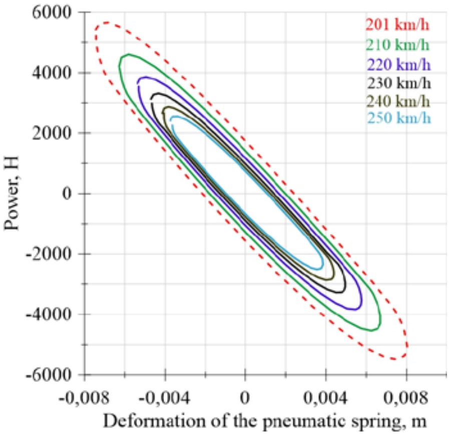

Fig. 10. Diagrams" force-deformation" of a pneumatic spring at $l = 4$ m, $d = 30$ mm in high driving speed conditions

Using the forms of the designed diagrams "force-deformation" of a pneumatic spring, the authors of the paper have proposed to determine the coefficient of a pneumatic spring operation during its compression and expansion (fig. 11), which is found by the formula:

$$k_{a.s.}^{\text{com-exp}} = \frac{a}{b} \tag{9}$$

The dependencies of a pneumatic spring operation coefficient on the diameter of the connecting pipeline in conditions of speed driving and high-speed driving are studied (fig. 12).

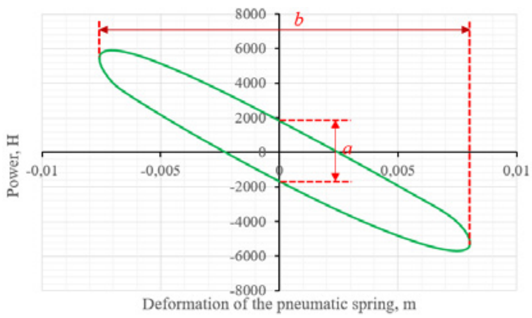

Fig. 11. Graphical display of the components of the pneumatic spring operation coefficient

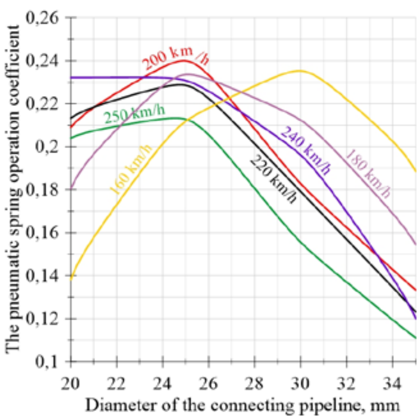

Fig. 12. Dependence of a pneumatic spring operation coefficient on the diameter of the connecting pipeline in conditions of speed driving and high-speed driving

Dependency analysis shows that for driving speeds starting from 180 km/h and more, the pneumatic spring operation coefficient at its compression and expansion is extreme, which is seen when the diameter of the connecting pipeline is 25 mm. The highest values of the coefficient are seen at a speed of 200 km/h. With an increase in the diameter of the connecting pipeline, the values of the pneumatic spring operation coefficient sharply decrease.

Thus, changing the speed value from 160 to 250 km/h and the diameter of the connecting pipeline from 20 to 35 mm, it is found that the pneumatic spring operation coefficients vary from 0.089 to 0.24.

The phenomenon of nonlinearity of the pneumatic spring operation coefficient and the influence of a set of operational factors will be investigated in the following works.

6 Conclusions

1. Using the design scheme of a dynamic model of rolling stock with two degrees of freedom, the diagrams "force - deformation" of a pneumatic spring operation are designed at different values of the connecting pipeline diameter in conditions of speed 160-200 km/h and high-speed driving of 201-250 km/h.

2. It is determined that when the diameter of the connecting pipeline changes from 20 to 35 mm, the inclination angle of the diagram "force-deformation" of the pneumatic spring will change: at speed driving from 32 degrees. up to 57 deg.; at high-speed driving from 30 deg. up to 48 deg.

3. Changing the speed value from 160 to 250 km/h. and the diameter of the connecting pipeline from 20 to 35 mm, it is found that the pneumatic spring operation coefficients vary from 0.089 to 0.24, and the dependences have a non-linear nature of change.

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