Determination of Dynamic Characteristics and Features of Operation of Pneumatic Springs of High-Speed Railway Rolling Stock, while Changing the Pressure Gauge in the Spring

Andrii Kuzyshyn, Vitalii Kovalchuk, Yuriy Royko, Ivan Kravets and Yuliya Sobolevska

Lviv Polytechnic National University
Stepan Bandera St. 12, 79013 Lviv, Ukraine
{andrii.y.kuzyshyn, vitalii.v.kovalchuk, yurii.y.roiko, ivan.b.kravets, yuliia.h.sobolevska}@ lpnu.ua

Abstract: The object of research is the pneumatic spring of the pneumatic spring suspension system for high-speed railway rolling stock. A methodology for experimental tests and theoretical studies of a pneumatic spring has been developed to determine the characteristics of the spring, taking into account changes in the pressure gauge in the spring. Static experimental tests of the pneumatic spring were carried out in the normal mode of its operation, that is, without including the emergency spring in operation. As a result, the "force-strain" dependences are obtained, taking into account the value of the manometric pressure in the pneumatic spring.

It is established that the laws of deformation of the pneumatic spring of high-speed rolling stock are influenced by the value of the manometric pressure in the spring. In the pressure change range from 0.5 to 2.0 atm. the maximum value of the applied force for spring deformation on a steel fork decreases with increasing pressure gauge. At the same time, on the deformation curves when the pressure increases by 0.5 atm. there is a zone of sharp changes in the force value with simultaneous slight deformation of the spring in the vertical plane. Further increase in the pressure gauge from 2.5 to 4.5 atm. leads to smooth operation of the spring in the "force-strain"ratio.

Keywords: high-speed rolling stock; pneumatic spring; rubber-cord shell; stiffness; pressure gauge; force, deformation

1 Introduction

Increasing the speed of railway rolling stock is a key task of increasing the capacity and carrying capacity of Railway Passenger Transportation. This leads to an increased force effects of rolling stock, on the rail track, especially in curved sections of small radius. However, to ensure traffic safety, the dynamic behavior of rolling stock must correspond to the permissible dynamic indicators (coefficient of vertical and horizontal dynamics of the first and second stages of spring suspension, acceleration of the body in vertical and horizontal directions, etc.) and traffic safety indicators (coefficient of stability margin against wheel derailment from the rail) [1] [2].

The main structural elements of the crew part of rolling stock, on the operation of which the level of dynamic indicators and traffic safety indicators depends, are the degrees of spring suspension (Fig. 1).

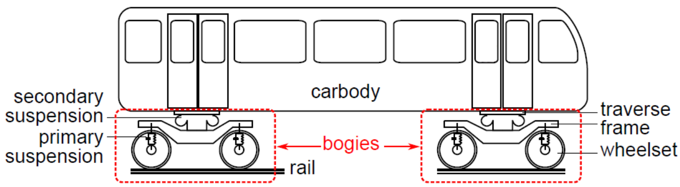

Figure 1
Rolling stock crew [3]

In the first stage of spring suspension, standard twisted cylindrical springs and hydraulic or friction vibration dampers are usually used. The second stage of spring suspension of high-speed rolling stock is more diverse.

For the most part, on high-speed rolling stock, a pneumatic spring suspension system is used in the second stage of spring suspension (between trolleys and the body). It consists of the following main structural elements: a pneumatic spring, an additional tank and a connecting pipeline.

This entire system is filled with compressed air, the pressure gauge of which may change during the operation of rolling stock. The main characteristics of the system are its rigidity and damping coefficient. The damping coefficient mainly depends on the geometric parameters of the connecting pipeline, which was studied in [4]. It should be noted that the dynamic stiffness in a pneumatic spring primarily depends on the state of the air in the pneumatic system and the stiffness of the rubber-cord shell of the pneumatic spring. It is these characteristics that mainly determine the level of dynamic forces in the links between the structural elements of rolling stock, which arise due to the interaction of the wheelset with the rail track.

Therefore, to ensure the safe operation of high-speed rolling stock, even at the design stage, it is necessary to conduct theoretical and experimental studies of the operation of the pneumatic spring suspension system. Such studies should be aimed at determining the stiffness of the pneumatic spring depending on the pressure gauge in the pneumatic spring suspension system.

This will allow us to establish the optimal parameters of the pneumatic spring suspension system, which ensures an acceptable level of dynamic indicators and traffic safety indicators when rolling stock interacts with the rail track.

2 Analysis of Literature Data and Problem Statement

Since the pneumatic spring is the most used structural component of the second stage of spring suspension of high-speed rolling stock, a number of research works are devoted to the issue of studying the pneumatic spring.

The main typical models are the "Nishimura model" [5], "Simpack Model" [6], "Vampire Model" [7], and "Berg model" [8]. Taking into account the development of existing concepts and theories, we will analyse the works that describe the study of the dynamic operation of a pneumatic spring suspension system.

In [9], the authors investigated the effect of a pneumatic spring suspension system on vertical accelerations of the body, taking into account the unevenness of the rail track and the amount of viscous damping. For this purpose, a test bench was developed and the operation of the pneumatic spring suspension system was analysed depending on the volume of the pneumatic spring and the additional tank, the connecting pipeline and the amplitude of the disturbance. Comparing the experimental results with the theoretical ones, it was proposed to use a hole with a variable cross-section in the connecting pipeline.

In [10], the authors considered some types of pneumatic spring models, such as the spring shock absorber model, Nishimura, Vampire, Simpac, and Gensys. Vertical movements of the trolley, body and wheelset of rolling stock were studied, and frequency characteristics were obtained. It was found that the vertical displacements of the body are approximately 0.05 m, with the introduction of the sinusoidal function as an unevenness on the rail track, which has an amplitude of 0.1 m. however, the vertical displacements of trolleys and wheelsets are the same.

In [11], mathematical modelling and analysis of the vertical stiffness characteristics of a pneumatic spring suspension system are performed. For this purpose, a dynamic model of vertical stiffness based on thermodynamics and hydrodynamics was developed, and the geometric parameters were determined by an approximate analytical method. In addition, the influence of geometric parameters on the characteristics of vertical stiffness is discussed using sensitivity analysis.

In [12], the influence of the volume of an additional tank, the length and diameter of the connecting pipeline on the dynamic behaviour of a pneumatic spring is studied. To achieve this goal, the authors used a nonlinear thermodynamic model of a pneumatic spring, which represented a combination of two different models. The research results showed that by changing the parameters of the pneumatic spring suspension system, it is possible to improve passenger comfort by reducing the Comfort Index by about 10%, which is confirmed by experimental studies [13] [14].

In [15], an innovative method for designing a pneumatic spring based on computer modelling was developed, while simultaneously using devices that can analyse nonlinear changes in the angle of reinforcing fibres of a rubber-cord shell. To verify the reliability of the angular distribution of reinforcing fibres, a non-destructive study of the angular distribution was performed using X-ray Computed Tomography. The vibrating model of the pneumatic spring included components of stiffness and viscosity.

In [16], the design features of rubber-cord elastic elements are considered. Experimental studies have shown that shells with a smaller closing angle have better vibration-proof properties. It is established that the dynamic characteristics of the pneumatic spring suspension system can be formed not only by changing the volume of the additional tank, the geometric parameters of the connecting pipeline, the internal pressure in the system, but also by the design parameters of the pneumatic spring shell itself.

In [17], the efficiency of a pneumatic spring suspension system in comparison with a passive system was investigated. Parametric analysis included the dimensions of the pneumatic spring, internal pressure, additional tank volume, and geometric parameters of the connecting pipeline. Dynamic responses, including acceleration, suspension travel, and dynamic force, were then compared in the form of time and frequency domain analysis compared to passive suspension.

In [18] [19], the authors review existing models describing the dynamic operation of the components of spring suspension of rolling stock. It is established that depending on the type of solving equations, the available models of pneumatic springs can be divided into three groups: mechanical, thermodynamic and finite element.

In [20], the author conducted multidisciplinary modeling of rolling stock with a pneumatic spring suspension system, taking into account both multi-element and pneumatic aspects. The criteria that affect the suspension morphology are presented: the type of trolley, the use of an additional tank, the position of this tank relative to the spring, the type of alignment system, the use of an additional stabilizer bar, the use of an additional hydraulic damper, etc. Thermodynamic models were used to take into account various aspects, which took into account the air flow through connecting pipelines and valves, the pressure in the pneumatic spring, and so on. Several models of connecting pipelines were presented. It is established that an algebraic model is sufficient for short connecting pipelines, and a differential model is required to take into account the dynamics of longer ones.

In [21], mathematical modeling of the pneumatic spring suspension system and its impact on driving safety and comfort are performed. To obtain mathematical models, quasi-static and dynamic characteristics were obtained using the results of the conducted experiments. With this in mind, two different approaches to modeling a pneumatic spring suspension system have been developed: quasi-static and dynamic. In quasi-static mode, the frequency-dependent behavior of the system is ignored, but the relationship between shear stiffness and rotation is taken into account. The dynamic approach involves using a thermodynamic model, taking into account the frequency in the vertical direction and the dependence of the transverse stiffness parameters on the load. It is noted that it is the dynamic model of the pneumatic spring that is necessary for the correct assessment of driving comfort, especially when the pneumatic spring suspension system includes a long connecting pipeline between the pneumatic spring and the tank.

In [22], the authors developed a nonlinear model of a pneumatic spring suspension system taking into account equalization valves and differential pressure, which is later integrated into universal computer algorithms for multi-mass dynamics of rolling stock. The importance of modeling nonlinear valve characteristics for assessing the safety of rolling stock during its operation at low speeds in curved sections of railway track is shown.

In [23], an analytical model of a pneumatic spring suspension system was obtained on the basis of experimental studies. It is shown that the dynamic behavior of a pneumatic spring suspension system can be made more universal by choosing the dimensions of its elements, in particular the volumes of the pneumatic spring and an additional tank. It is established that to reduce vibration, it is necessary to use a control system that is able to make decisions and select the parameters of the connecting pipeline in accordance with a given disturbance frequency.

In [24], the authors developed a generalized analytical model for predicting the amplitude - and frequency-dependent behavior of a pneumatic spring as a result of many factors, such as thermodynamics, friction, and damping.

In [4], the authors developed a mathematical model of vibrations of a two-mass system, the elements of which are connected through a pneumatic spring suspension system. The operation of a pneumatic spring suspension system is described using the Boyle-Marriott equations, the ideal gas state, the energy for flow in the connecting pipeline, and the law of conservation of energy. Theoretical studies of the influence of the diameter and length of the connecting element of a pneumatic spring suspension system on energy loss and damping coefficient per cycle of its operation and the stiffness of the pneumatic spring are carried out.

In [25], the authors constructed and investigated diagrams of the "force - strain" operation of a pneumatic spring at different values of the diameter of the connecting pipeline at a speed of 160-200 km/h. and high-speed traffic of 201-250 km/h. Changing the speed value from 160 to 250 km/h. and the diameter of the connecting pipeline from 20 to 35 mm, it was found that the coefficients of operation of the pneumatic spring vary from 0.089 to 0.24, and the dependences have a non-linear nature of change.

In the works [26-36], it is indicated that malfunctions of the rail track: subsidence of the roadbed, switches, upset ballast of the railway track lead to an increase in the dynamic effect on rolling stock. In addition, the change in the stiffness of the pneumatic spring is influenced by the design features of the sub-rail base, such as reinforced concrete sleepers in combination with fiber reinforcement [37], the quality of materials of the upper structure of the track [38], the granulometric composition of crushed stone ballast [39] and the use of geotextile materials under crushed stone ballast [40].

The above analysis of the literature demonstrates a wide range of studies of the dynamic operation of structural components of a pneumatic spring suspension system for high-speed rolling stock. However, most studies are based on separate mechanical or thermodynamic models, and do not take into account the simultaneous behavior of both the state of air in the pneumatic system and the features of the operation of the rubber-cord shell of a pneumatic spring. In addition, the operating conditions of rolling stock tend to change rapidly, which will lead to a change in the pressure gauge in the pneumatic spring suspension system.

Consequently, improving the safety of high-speed rolling stock depends on the level of values of dynamic indicators of its mechanical part, which are affected by the dynamic behavior of the pneumatic spring suspension system.
Therefore, in this work, the dynamic stiffness of a pneumatic spring is studied depending on the value of the manometric pressure and the features of the operation of the rubber-cord shell. For this purpose, theoretical studies and experimental measurements of the force - strain relationship of a pneumatic spring were carried out.

The aim of the work is theoretical and experimental studies of the dynamic stiffness of the pneumatic spring of high-speed railway rolling stock, taking into account the operation of the rubber-cord shell and changes in the pressure gauge in the spring.

To achieve this goal, the following tasks were set:

- To develop a methodology for experimental studies of the stiffness of the rubber-cord shell of a pneumatic spring
- To investigate the stiffness of the pneumatic spring depending on the manometric pressure in the pneumatic spring suspension system

3 Materials and Methods for Studying the Dynamic Stiffness of a Pneumatic Spring of High-Speed Railway Rolling Stock

3.1 Description of the Test Unit for Experimental Determination of Pneumatic Spring Stiffness

Determination and study of the characteristic features of the stiffness of the pneumatic spring of high-speed rolling stock was carried out in laboratory conditions. The test unit consists of the structural components of the installation frame, a pneumatic spring suspension system (pneumatic spring, additional tank, connecting pipeline), power and measuring equipment. The general scheme of the test unit for studying the dynamic stiffness of a pneumatic spring is shown in Fig. 2. The program of experimental tests provided for finding the "force – strain" dependences of a pneumatic spring depending on the value of the manometric pressure. The tests were performed when the pressure gauge changed in the range from 0 atm. up to 5.0 atm. in 0.5 atm. increments.

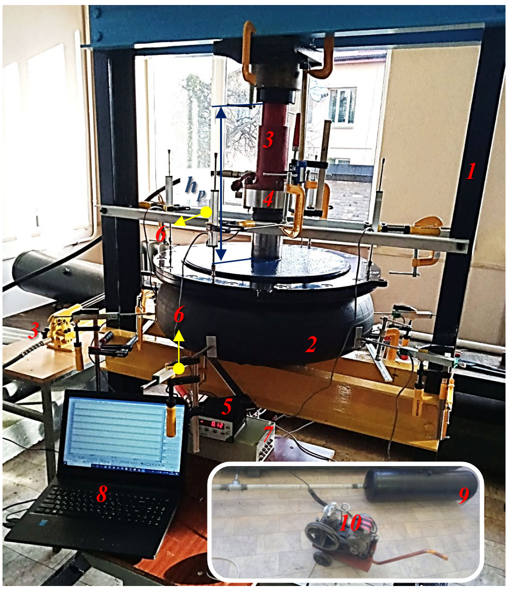

Figure 2
Experimental setup for studying the dynamic stiffness of a pneumatic spring: 1 - supporting structure of the stand; 2 - pneumatic spring; 3-hydraulic jack; 4-strain gauge force sensor; 5-digital reader of the force value; 6-Potentiometric sensors of linear displacements; 7-analog-to-digital converter; 8-personal computer; 9-additional tank; 10-high-pressure compressor

It should be noted that according to the operational parameters of the pneumatic spring suspension system, the operating pressure can be in the range from 3.8 to 7.0 atm. However, under operating conditions, there are cases in which the pressure may decrease below 3.8 atm, which indicates an emergency mode of rolling stock movement. In addition, as part of the laboratory experiment, it was not possible to conduct tests at pressures of 6.0 and 7.0 atm., which is related to the safe operation of measuring equipment and labor protection. Therefore, the pressure range from 0 atm is selected. up to 5.0 atm.

At each stage of the experiment, which was accompanied by an increase in pressure by 0.5 atm. a constant distance was provided from the upper plate of the pneumatic spring to the upper stop of the hydraulic jack.

The pressure gauge change was carried out using a compressor *10*, and the value was controlled by a high-pressure pressure gauge, which was located on the connecting pipeline between the pneumatic spring *2* and the additional tank *9*. A hydraulic jack *3* was used to apply a vertical load to the pneumatic spring *2*, and the amount of the applied load was recorded using a strain gauge force sensor *4* and displayed on the screen of a digital force converter *5*. During vertical loading, the pneumatic spring *2* was deformed, the value of which was recorded using sensors of linear movements *6*, which were read by an analog-to-digital converter 7 and stored in the memory of a personal computer 8.

The equipment used has the following parameters: high – pressure compressor-8 atm. hydraulic jack-maximum load of 30 tons; strain gauge force sensor – operating range of 12 tons; Potentiometric sensors of linear movements-operating range up to 100 mm. The measurement values were read by an analog-to-digital converter with a polling frequency of 1000 Hz.

The vertical load was applied until the vertical deformation of the pneumatic spring along the axis reached a value of 33.0 mm, which corresponds to the normal operation of the pneumatic spring. Exceeding the deformations of 33.0 mm causes the spring to operate in emergency mode.

3.2 Mathematical Model for Studying the Dynamic Stiffness of a Pneumatic Spring

To study the dynamic stiffness of a pneumatic spring, it is necessary to simultaneously take into account the thermodynamic processes that occur in the pneumatic spring suspension system and the features of the operation of the rubber-cord shell of a pneumatic spring.

Considering thermodynamic processes, the component of the force acting on the pneumatic spring when it is deformed by the value $\Delta h(t)$ was found:

$$F_1(t) = P_1(t) \cdot A_1 \tag{1}$$

where $P_1(t)$ is the working fluid pressure in the pneumatic spring, Pa; A_1 is the effective area of the pneumatic spring, m².

Considering the equation of state of an ideal gas, an equation was obtained for determining the internal air pressure in a pneumatic spring:

$$\dot{P}_1(t) = -\dot{h}_1(t)\frac{P_1(t)}{h_1(t)} + \dot{m}_1(t)\frac{R \cdot T_1(t)}{h_1(t) \cdot A_1} + \dot{T}_1(t)\frac{m_1(t) \cdot R}{h_1(t) \cdot A_1} \tag{2}$$

where $h_1(t)$ is the height of the pneumatic spring at a certain point in time, m; $m_1(t)$ is the mass of air in the pneumatic spring, kg; R is the Universal Gas Constant, J/(kg·K); $T_1(t)$ is the temperature of the working fluid of the pneumatic spring, K; $\dot{P}_1$(t) – the first derivative of the internal air pressure in the pneumatic spring in time; $\dot{T}_1$(t) – the first derivative of the air temperature in the pneumatic spring in time; $\dot{m}_1$(t) – the first derivative of the mass fraction of air in the pneumatic spring in time, $\dot{h}_1$ (t) – the first derivative of the height of the pneumatic spring in time.

The air flow rate through a connecting pipeline with certain geometric parameters, using the Bernoulli equation, was found by the formula:

$$V(t) = \sqrt{\frac{2\Delta P(t)}{\rho(t) \cdot \left(f\dfrac{l}{d} + K_S + K_P\right)}} \tag{3}$$

where ρ is the density of the working fluid, kg/m^3; f is the coefficient of friction along the length (Darcy coefficient); l is the length of the connecting pipeline, m; d is the diameter of the connecting pipeline, m; K_s is the coefficient of losses on instantaneous compression; K_p is the coefficient of losses on instantaneous expansion.

Knowing the value of the average air flow velocity, the cross-sectional area of the flow and the density of the working fluid, the equation for determining the mass flow rate will have the form:

$$\dot{m}(t) = \frac{\left(\dfrac{\pi}{4}\right) d^2 \sqrt{2 \cdot \rho(t) \cdot \left|P_1(t) - P_2(t)\right|}}{\sqrt{f\dfrac{l}{d} + K_S + K_P}} \cdot sign\left(P_1(t) - P_2(t)\right) \tag{4}$$

where $P_2(t)$ is the working fluid pressure in the additional tank, Pa;

To determine the temperature of the working fluid in a pneumatic spring, the law of conservation of energy was used, which takes into account the transfer of energy from the pneumatic spring to the additional tank and vice versa, as well as heat transfer between the pneumatic spring and the environment:

$$\dot{T}_1(t) = -\frac{A_1 \cdot P_1(t) \cdot \dot{h}(t)}{m_1(t) \cdot C_v} - \frac{\dot{m}(t) \cdot C_p \cdot T_1(t)}{m_1(t) \cdot C_v} + \frac{h_T \cdot A_s(t) \cdot (T_s - T_1(t))}{m_1(t) \cdot C_v} \tag{5}$$

where h_T is the heat transfer coefficient, W/(m^2·K); C_v is the specific heat capacity at constant volume, J/(kg·K); C_p is the specific heat capacity at constant pressure, J/(kg·K); A_s is the heat transfer area, m^2.

So, knowing the amount of force acting on the pneumatic spring and, accordingly, its deformation, you can build diagrams of the operation of the pneumatic spring, which will allow you to determine the dynamic stiffness of the pneumatic spring.

The features of the rubber-cord shell, as the second component of the dynamic stiffness of a pneumatic spring, will be investigated experimentally, constructing the "force - strain" relationship and determining its stiffness accordingly.

4 Results of Theoretical and Experimental Studies of Dynamic Stiffness of a Pneumatic Spring of High-Speed Railway Rolling Stock

4.1 Determination of Vertical Stiffness of the Rubber-Cord Shell of a Pneumatic Spring

Determination of the stiffness of the rubber-cord shell of a high-speed rolling stock pneumatic spring was carried out on an experimental installation (Fig. 2) if there is no manometric pressure in the pneumatic spring. Based on the measurements made, the force – strain relationship of the rubber-cord shell was obtained, which is shown in Fig. 3. The study was carried out until the vertical deformation of the pneumatic spring along the axis reached a value of 33.0 mm, which corresponds to the normal operation of the pneumatic spring. Exceeding the deformations of 33.0 mm causes the spring to operate in emergency mode.

The value of the averaged dependence can be described by a polynomial equation:

$$F_2(t) = 1 \cdot 10^7 \cdot \Delta h(t)^2 + 1 \cdot 10^6 \cdot \Delta h(t) - 586,81 \tag{6}$$

where $F_2(t)$ is the deformation force of the rubber – cord shell, N.

Based on the results obtained (Fig. 3) we construct the dependence of the stiffness of the rubber-cord shell in the absence of manometric pressure in the pneumatic spring (Fig. 4).

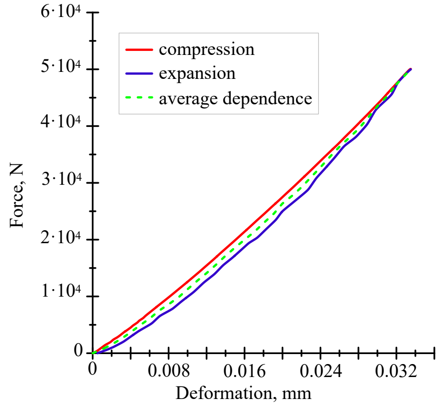

Figure 3

"Force-strain" dependence of the rubber-cord shell of a high-speed rolling stock pneumatic spring in the absence of manometric pressure

Therefore, the obtained stiffness must be taken into account when determining the total stiffness of the pneumatic spring, taking into account the presence of manometric pressure in the pneumatic spring.

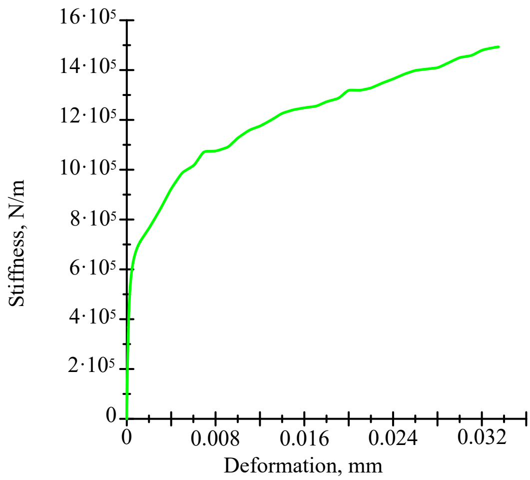

Figure 4

Dependence of the stiffness of the rubber-cord shell of a pneumatic spring

4.2 Determination of the Dynamic Stiffness of the Pneumatic Spring Depending on the Value of Manometric Pressure

The dynamic stiffness of a pneumatic spring was determined by two methods: experimental and theoretical. The theoretical method used a thermodynamic model and the dependence of the stiffness of the rubber-cord shell on the amount of its deformation. The experimental method used an installation, power and measuring equipment.

The force - strain dependencies are constructed on the basis of experimental data (Fig. 5).

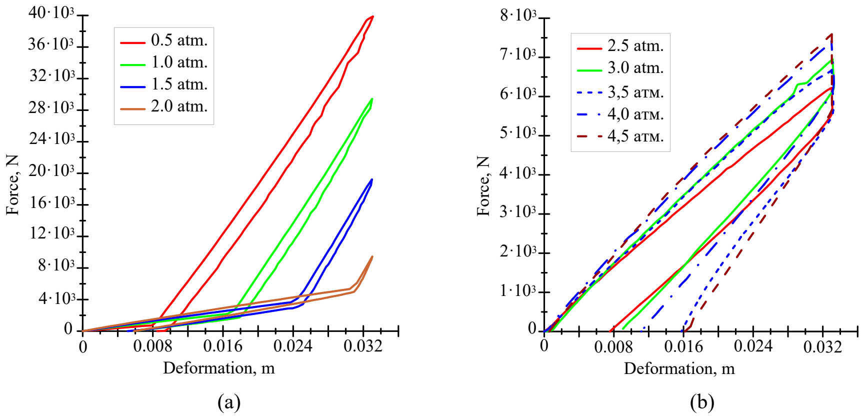

Figure 5

(a) Experimental "force-strain" dependencies of a pneumatic spring of high-speed rolling stock at a manometric pressure of 0.5 atm. up to 2.0 atm (b) at a manometric pressure of 2.5 atm. up to 4.5 atm.

Analysis of experimental dependencies (Fig. 5a) shows that at a pressure gauge of up to 2.0 atm. there are certain features of the operation of the pneumatic spring. At the initial stage of its deformation, the key place is occupied by the work of air in the middle of the pneumatic spring suspension system and the rubber-cord shell. However, there comes a time when the main work is performed by the rubber-cord shell of the pneumatic spring. At a manometric pressure of 0.5 atm. such a feature is observed at avertical deformation of 8 mm, at 1.0 atm. – 16.6 mm, at 1.5 atm. – 24 mm, at 2.0 atm. – 30.4 mm. Consequently, to the values of these displacements, the stiffness of the pneumatic spring is formed by the work of air and the rubber-cord shell, and with large deformations – only the rubber-cord shell. At the same time, the maximum value of the applied force for deforming the pneumatic spring by its maximum value is: at 0.5 amt. – $39.880 \cdot 10^3$ N, at 1.0 atm. – $29.335 \cdot 10^3$ N, at 1.5 atm. – $19.073 \cdot 10^3$ N, at 2.0 atm – $9.413 \cdot 10^3$ N. As can be seen from Fig. 5a, when the value of manometric pressure increases to 2.0 atm., the maximum force value decreases.

At a manometric pressure of 2.0 atm. up to 5.0 atm. the dynamic stiffness of the pneumatic spring is simultaneously formed by the work of the air in the middle of the pneumatic spring suspension system and the work of the rubber-cord shell of the pneumatic spring, over the entire range of its vertical deformation (Fig. 5b). The maximum values of the force required to deform the pneumatic spring are in the range from $6.211 \cdot 10^3$ N to $7.592 \cdot 10^3$ N.

The resulting dependencies (Fig. 5a) non-linear operation of the pneumatic spring in the pressure range of 0.5-2.0 atm. they emphasize the need in practice to provide a minimum pressure of 2.5 atm in the pneumatic spring suspension system. and higher. This is due to the close linear force-strain dependencies shown in Fig. 5b. For safety reasons and optimal operation of the pneumatic spring suspension system for high-speed rolling stock, this is necessary for its normal operation in high-speed traffic conditions. Also, when operating rolling stock, this can be the basis for predicting the need for maintenance of high-speed rolling stock.

Taking into account the simultaneous operation of thermodynamic processes occurring in the pneumatic spring suspension system and the operation of the rubber-cord shell of the pneumatic spring, we will perform a theoretical determination of the rigidity of the pneumatic spring of high-speed railway rolling stock and compare it with full-scale results (Table. 1).

Table 1
Dynamic stiffness of the pneumatic spring

<table>
<tr><th rowspan="2">Pressure value</th><th rowspan="2">Deformation range</th><th colspan="2">Pneumatic spring stiffness, N / m</th></tr>
<tr><th>Theoretical method</th><th>Experimental method</th></tr>
<tr><td rowspan="2">0.5</td><td>up to 8 mm</td><td>$1.041 \cdot 10^5$</td><td>$9.52 \cdot 10^4$</td></tr>
<tr><td>from 8 mm to 33 mm</td><td colspan="2">$k = 9 \cdot 10^{10} \cdot x^3 - 7 \cdot 10^9 \cdot x^2 + 2 \cdot 10^8 \cdot x - 1 \cdot 10^6$</td></tr>
<tr><td rowspan="2">1.0</td><td>up to 16.6 mm</td><td>$1.247 \cdot 10^5$ N/m</td><td>$1.328 \cdot 10^5$ N/m</td></tr>
<tr><td>from 16.6 mm to 33 mm</td><td colspan="2">$k = -1 \cdot 10^9 \cdot x^2 + 1 \cdot 10^8 \cdot x - 1 \cdot 10^6$</td></tr>
<tr><td rowspan="2">1.5</td><td>up to 24 mm</td><td>$1.531 \cdot 10^5$ N/m</td><td>$1.547 \cdot 10^5$ N/m</td></tr>
<tr><td>from 24 mm to 33 mm</td><td colspan="2">$k = 5 \cdot 10^7 \cdot x - 1 \cdot 10^6$</td></tr>
<tr><td rowspan="2">2.0</td><td>up to 30.4 mm</td><td>$1.801 \cdot 10^5$</td><td>$1.806 \cdot 10^5$</td></tr>
<tr><td>from 30.4 mm to 33 mm</td><td colspan="2">$k = 4 \cdot 10^7 \cdot x - 1 \cdot 10^6$</td></tr>
<tr><td rowspan="2">2.5</td><td>up to 12 mm</td><td colspan="2">$2.029 \cdot 10^5$</td></tr>
<tr><td>from 12 to 33 mm</td><td>$k = \left(\frac{k_1 \cdot k_2}{k_1 + k_2} \right) \cdot \xi$
where $\xi = 0,95$</td><td>$1.956 \cdot 10^5$</td></tr>
<tr><td>3.0</td><td>from 0 mm to 33 mm</td><td>$k = \left(\frac{k_1 \cdot k_2}{k_1 + k_2} \right) \cdot \xi$
where $\xi = 0,95$</td><td>$2.161 \cdot 10^5$</td></tr>
</table>

3.5		$k=\left(\dfrac{k_1\cdot k_2}{k_1+k_2}\right)\cdot\xi$ where $\xi=0,85$	$2.126\cdot10^5$
4.0		$k=\left(\dfrac{k_1\cdot k_2}{k_1+k_2}\right)\cdot\xi$ where $\xi=0,85$	$2.341\cdot10^5$
4.5		$k=\left(\dfrac{k_1\cdot k_2}{k_1+k_2}\right)\cdot\xi$ where $\xi=0,85$	$2.478\cdot10^5$
5.0		$k=\left(\dfrac{k_1\cdot k_2}{k_1+k_2}\right)\cdot\xi$ where $\xi=0.95$	$2.96\cdot10^5$

In the Table 1 k_1 is the stiffness component of the pneumatic spring, which is responsible for the operation of air, and k_2 is the component responsible for the operation of the rubber – cord shell.

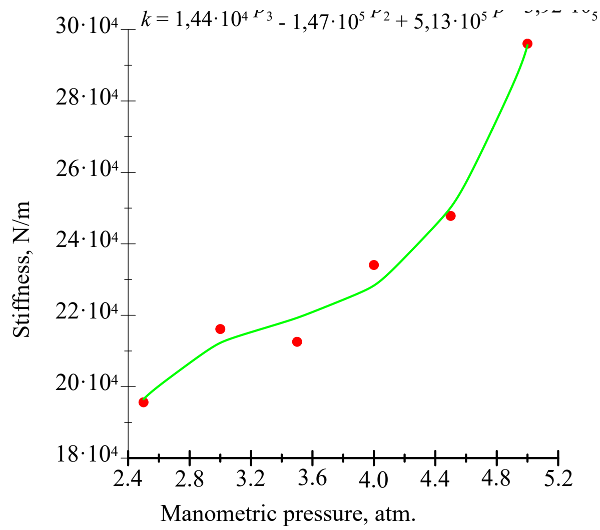

Figure 6

Dependence of the dynamic stiffness of the pneumatic spring on the value of manometric pressure

Analyzing the Table 1 and comparing the theoretical and experimental values of the stiffness of the pneumatic spring, in the range of manometric pressure from 2.5 atm. up to 5.0 atm. the correction factor ξ is introduced. This coefficient is introduced in order to equalize the conditions of experimental and theoretical studies. The coefficient ξ ranges from 0.85 to 0.95.

It is established that when the manometric pressure value increases in the range of 2.5÷5.0 atm. the stiffness of the pneumatic spring increases from $1.956 \cdot 10^5$ N/m to $2.96 \cdot 10^5$ N/m and has a non-linear change, which is shown in Fig. 6.

Analysis of experimental data on the stiffness of a pneumatic spring allowed us to establish the theoretical dependence of the dynamic stiffness of a pneumatic spring as a function of the manometric pressure value. This will allow us to study the dynamic indicators and traffic safety indicators of modern high-speed rolling stock even at the design stage.

5 The Results of the Discussion of the Dynamic Behavior of the Pneumatic Spring when the Manometric Pressure Changes

To find the dynamic stiffness of a pneumatic spring, it is proposed to use an experimental installation that includes the structural components of the frame, power and measuring equipment. The test was carried out with a change in manometric pressure in the range from 0 atm. up to 5.0 atm. with a step of 0.5 atm.

During an experimental study of the operation of a pneumatic spring, ranges of vertical deformation values were revealed at which air performs useful work, and there are ranges when the key work is performed by a rubber-cord shell. At a manometric pressure of 0.5 atm. such a feature is observed at a vertical deformation of 8 mm, at 1.0 amt. – 16.6 mm, at 1.5 atm. – 24 mm, at 2.0 atm. – 30.4 mm (Fig. 5a).

It should be noted that at a manometric pressure of 2.0 atm. up to 5.0 atm. the dynamic stiffness of the pneumatic spring is simultaneously formed by the operation of the air inside the pneumatic spring suspension system and the operation of the rubber-cord shell of the pneumatic spring (Fig. 5b).

Due to the rubber-cord shell of a pneumatic spring, the "force-strain" relationship was obtained experimentally, the value of which is described by a polynomial equation (Fig. 3). Using the "force-strain" relationship, the stiffness of the rubber-cord shell in the absence of manometric pressure in the pneumatic spring is found.

Using a thermodynamic model and an experimental dependence of the stiffness of the rubber-cord shell, the vertical stiffness of a pneumatic spring was found at different values of manometric pressure (Table 1). After comparing the obtained values with the values obtained by the experimental method for the manometric pressure range from 2.5 atm. up to 5.0 atm. the correction factor ξ is introduced, which is in the range of 0.85÷0.95, which is necessary to equalize the conditions of experimental and theoretical studies.

An increase in the manometric pressure from 2.5 atm. to 5.0 atm. increases the stiffness of the pneumatic spring from $1.956 \cdot 10^5$ N/m to $2.96 \cdot 10^5$ N/m, the dependence of which is shown in Fig. 6. At the same time, changing these values allows you to change the forces in the spring suspension system, and influence the level of dynamic indicators and traffic safety indicators during the interaction of rolling stock with the rail track.

These studies will be the basis for setting the input parameters of a pneumatic spring when developing a dynamic model "rolling stock - track" in conditions of high-speed movement of rolling stock. This will allow us to determine the dynamic indicators and safety indicators of rolling stock movement, depending on the operating conditions of the pneumatic spring suspension system.

It should be noted that the accumulation of big data by monitoring will allow using machine learning to study the influence of rolling stock operating conditions on its dynamic performance, including the stiffness of the pneumatic spring. The collected data for a certain period of time can be used in the study of dynamic indicators and indicators of rolling stock safety using the theoretical model "rolling stock - track". This will allow us to evaluate the operation of the pneumatic spring for a certain period of its life cycle at the design stage, taking into account changes in pressure and the influence of structural and technical factors of the rail track and undercarriage of rolling stock. On the basis of this, it is possible to further establish criteria for safe operation of rolling stock in high-speed traffic conditions, as well as predict the urgency of its maintenance.

One of the disadvantages of the study is the failure to take into account changes in the geometric shape of the rubber-cord shell of a pneumatic spring due to its deformation. It should be noted that taking into account the change in the geometric shape of the rubber-cord shell will increase the accuracy of determining the effective area of the pneumatic spring, which will affect the value of the force and stiffness of the pneumatic spring.

Conclusions

An experimental unit has been developed to determine the stiffness of the pneumatic spring of high-speed railway rolling stock, which consists of counter-structural elements of the frame, power and measuring equipment. It allows you to measure the forces and deformations of the pneumatic spring when the pressure gauge changes.

Based on the results of static experimental tests, the dependence of the stiffness of the rubber-cord shell with the absence of manometric pressure in the pneumatic spring is obtained. The maximum stiffness value is $14.94 \cdot 10^5$ N/m, which corresponds to the maximum deformation value of the pneumatic spring when the spring is operating in normal mode.

It was established herein that, at a manometric pressure of 0.5 atm., 1.0 atm., 1.5 atm.

and 2.0 atm. when the vertical deformation of the pneumatic spring is up to 8 mm, 16.6 mm, 24 mm, 30.4 mm, respectively, the work is performed by the manometric pressure in the middle of the pneumatic spring suspension system and the rubber cord shell. When the amount of deformation increases, the work is performed by the rubber-cord sheath. At a manometric pressure of 2.0 atm. up to 5.0 atm. the stiffness of the pneumatic spring simultaneously forms the manometric pressure inside the pneumatic spring suspension system and the operation of the rubber-cord shell.

Based on the theoretical thermodynamic model and the experimental dependence of the stiffness of the rubber-cord shell, theoretical values of the stiffness of the pneumatic spring are obtained. Comparing the obtained theoretical values with full-scale ones, a correction factor ξ in the range from 0.85 to 0.95 is introduced into the formula for determining stiffness, which makes it possible to equalize the conditions of experimental and theoretical studies.

It was established that with an increase in the manometric pressure in the range of 2.5÷5.0 atm. the stiffness of the pneumatic spring increases from $1.956 \cdot 10^5$ N/m to $2.96 \cdot 10^5$ N/m. Based on experimental data, a trend line is constructed that describes the dependence of changes in the stiffness of a pneumatic spring on the value of manometric pressure.

Acknowledgement

We express our gratitude to the design bureau of the Kryukovsky Railway Car Building Works, for assistance in experimental studies of the characteristics and features of the operation of the pneumatic springs in high-speed rolling stock.

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