Research of the characteristics of an air spring under cyclic load

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Research of the characteristics of an air spring under cyclic load

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Abstract. The dynamic stiffness, energy dissipation per cycle and the damping coefficient of the air spring of the second stage of spring suspension of high-speed rolling stock are investigated on the basis of a simplified mathematical model. The studies were carried out in the frequency range from 0 to 15 Hz. To determine the parameters of the air spring, its power characteristic was built, the area that characterizes the energy dissipation of the system per cycle. Based on the constructed characteristics, the damping coefficient and air spring stiffness during compression and expansion were determined. The influence of the frequency of the disturbing force and the diameter of the connecting element on the investigated parameters was studied. The obtained ratios of quantities would be made it possible to optimize the parameters of spring suspension of high-speed rolling stock from the point of view of ensuring admissible dynamic performance and traffic safety indicators.

1. Introduction

To ensure smooth running and limit the level of amplitudes of oscillations of high-speed electric trains, it is necessary to have a fairly "soft" spring suspension. For this purpose, air springs are used, which are the main structural element of the spring suspension system on the operation of which the overall dynamic behavior of high-speed rolling stock during its movement on the rail track depends (**Figure 1**). The design of such springs allows providing the standard value of static deflection up to 200 mm for passenger wagons and multiple unit trains, intended for a speed of 200 km/h. In addition, this suspension is characterized by both elastic and dissipative properties, i.e. there is no need to install a special dampener.

At the stage of designing high-speed rolling stock, the question arises regarding the modeling of an air spring and the determination of its main parameters. The dynamic behavior of an air spring can be assessed by the following main parameters: dynamic rigidity, energy dissipation per cycle, damping coefficient, etc. Determining these parameters is important because it allows, at the design stage, for assessing the dynamic indicators of rolling stock and establishing the optimal operational conditions of moving along the rail track [1, 2]. Currently, the verification of the respective indicators of rolling stock is performed mainly experimentally after selecting a definite model of an air spring. However, taking into account a lot of different factors makes such an experiment multifactorial and complex.

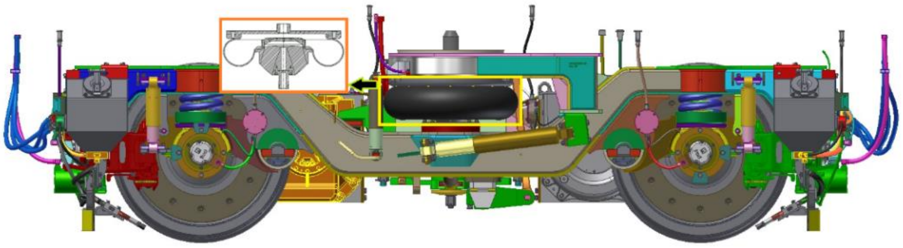

Figure 1. General appearance of an air spring

Thus, the analysis of the relationship between the parameters of an air spring and the factors that characterize its design and the conditions in which it operates is a relevant task and needs further solving.

2. Analysis of literary references and problem statement

Depending on the chosen mathematical model of an air spring, the methods of calculating dynamic rigidity, energy dissipation per cycle and damping coefficient of an air spring are chosen.

In work [3], the rigidity of an air spring was determined as the sum of adiabatic rigidity and rigidity of the air spring due to changes in its effective area:

$$\mathrm{k_s} = \frac{\gamma \cdot \mathrm{A_н} \cdot \mathrm{p}}{\mathrm{H}}, \tag{1}$$

$$\mathrm{k}_l = l \cdot \mathrm{p_g}, \tag{2}$$

where $\mathbf{k_s}$ – adiabatic rigidity of the air spring; $\mathbf{k}_l$ – rigidity of the air spring due to changes in its effective area; $\mathbf{A_н}$ – effective area of the air spring at the design height; $\mathbf{p_g}$ – air pressure in the air spring; $\boldsymbol{\gamma}$ – ratio of specific heats of air; $\mathbf{p}$ – absolute air pressure in the air spring; $\boldsymbol{l}$ – constant change of the effective area of the air spring with change of its height; $\mathbf{H}$ – design height of the air spring.

In works [4, 5], the vertical dynamic rigidity, which consists of three components, and viscous damping, which involves energy loss due to air flow between the air spring and the additional reservoir, are determined by the formulas:

$$K_1 = n \cdot A_e^2 \cdot \frac{p_0}{V_r};\ K_2 = n \cdot A_e^2 \cdot \frac{p_0}{V_b};\ K_3 = (p_0 - p_a) \cdot \frac{dA_e}{dz}, \tag{3}$$

$$C = R_f \cdot A_e^2 \cdot \rho_0 \cdot \mathrm{g} = \frac{0{,}126}{d_s^3} \cdot A_e^2 \cdot \rho_0 \cdot \mathrm{g}, \tag{4}$$

where $\boldsymbol{K_1}$ – rigidity of the air spring, appearing due to air compression in case of a blocked equalizing pipeline, N/m; $\boldsymbol{K_2}$ – rigidity appearing due to air compression in the additional reservoir, N/m; $\boldsymbol{K_3}$ – rigidity of the surface of the air spring which changes due to changes in its effective area during compression or expansion, N/m; $\boldsymbol{R_f}$ – coefficient of hydraulic resistance; $\boldsymbol{A_e}$ – effective area of the air spring, m^2; $\boldsymbol{\rho_0}$ – air density, kg/m^3; $\mathbf{g}$ – free fall acceleration, m/s^2; $\boldsymbol{d_s}$ – diameter of the connecting pipeline, m.

In paper [6] a method of determining the equivalent rigidity of an air spring was proposed, which additionally takes into account the rigidity of the emergency spring:

$$k_e = k \cdot \frac{\beta_2 \cdot \left(\beta_1 + \frac{\alpha}{1+\alpha}\right)}{\beta_1 + \beta_2 + \frac{\alpha}{1+\alpha}}, \tag{5}$$

$$k = \frac{A_{еф.}^2 \cdot p \cdot n}{V};\ \alpha = \frac{V}{V_{дод.}};\ \beta_1 = \frac{k_1}{k};\ \beta_1 = \frac{k_2}{k},$$

where $\boldsymbol{k}$ – rigidity of the air spring; $\boldsymbol{k_1}$ – rigidity defined by the change in effective area of the air spring; $\boldsymbol{k_2}$ – rigidity of the emergency spring; $\boldsymbol{\alpha}$ – ratio of volumes of the air spring and the additional reservoir.

In work [7], which is a continuation of works [8-10], on the basis of a viscoelastic mechanical model of an air spring, formulas for calculating dynamic rigidity and damping coefficient were obtained:

$$K_{ez} = \left(\frac{1}{\frac{p_0 \cdot A_e^2 \cdot n}{V_{b0} + V_{r0}} + p_g \cdot \frac{dA_e}{dz}} + \frac{1}{K_{\text{дод.}}} \right)^{-1}, \quad (6)$$

$$K_{vz} = \left(\frac{1}{\frac{p_0 \cdot A_e^2 \cdot n}{V_{b0}} + p_g \cdot \frac{dA_e}{dz}} + \frac{1}{K_{\text{дод.}}} \right)^{-1} - K_{ez}, \quad (7)$$

$$C_{z,\beta} = \frac{1}{2} \rho \cdot k_t \cdot A_s \cdot \left(\frac{A_e}{A_s} \cdot \frac{V_{r0}}{V_{b0} + V_{r0}} \right)^{1+\beta}, \beta = 2 \quad (8)$$

where $\boldsymbol{l_s}$ – length of the connecting element; $\boldsymbol{A_s}$ – cross section of the connecting element; $\boldsymbol{\rho}$ – air density at pressure p_0; $\boldsymbol{A_e}$ – effective area of the air spring; $\boldsymbol{V_{r0}}$ – volume of the additional reservoir; $\boldsymbol{V_{b0}}$ – volume of the air spring; $\boldsymbol{p_0}$ – initial absolute pressure of the air spring; $\boldsymbol{p_g}$ – manometric pressure; $\boldsymbol{k_t}$ – loss coefficient in the connecting element; $\boldsymbol{M}$ – mass of air.

In works [11, 12], the laws of thermodynamics were used under the assumption that the process is adiabatic, and the heat exchange between an air spring and the environment was neglected. The dynamic rigidity of the air spring was found by the formula:

$$K_d = (P_s - P_0) \frac{\partial A_e}{\partial z} + \frac{k \cdot P_s \cdot A_e}{m_s} \cdot \frac{G}{\dot{z}} - \frac{k \cdot P_s \cdot A_e}{V_s} \cdot \frac{\partial V_s}{\partial z}, \quad (9)$$

where $\boldsymbol{z}$ – vertical deformation of the air spring, m; $\boldsymbol{A_e}$ – effective area of the air spring, m^2; $\boldsymbol{k}$ – adiabatic index.

In paper [13], the formula of dissipation coefficient was obtained using a dimensionless model of an air spring with an additional reservoir connected by a hydraulic support:

$$\xi = R_F \frac{V_{20}}{nRT_0} \cdot \frac{nP_{10}\nu A - \alpha V_{10}(P_{10} - P_A)}{nP_{10}\nu A - \alpha (V_{10} + V_{20})(P_{10} - P_A)} \sqrt{\frac{\frac{nP_{10}\nu A}{V_{10}} - \alpha (P_{10} - P_A)}{m}}, \quad (10)$$

where $\boldsymbol{R_F}$ – orifice resistance, $\frac{P_a}{kg/s}$; $\boldsymbol{P}$ – absolute pressure, Pa; $\boldsymbol{\alpha}$ – derivative of the effective area, m; $\boldsymbol{A}$ – effective area of the air spring, m^2; $\boldsymbol{C_2}$ – pneumatic capacity of the additional reservoir, kg/Pa; $\boldsymbol{m}$ – sprung mass, kg; subscripts: **1** – spring; **2** – reservoir; **0** – static conditions.

In work [14], the dynamic vertical rigidity of the spring was determined taking into account the change in its effective area and volume due to the deformation of the spring:

$$K_z = n(P + P_a) \frac{S_e^2}{V} + 2\pi P R_e A_z^R. \quad (11)$$

Additional consideration of the rigidity of the air spring shell, performed in work [15], allowed obtaining the following formula for determining the vertical rigidity of the spring at a fixed height:

$$K = k(P_{z0} + p_a)\frac{A_{z0}}{V_{z0}}\frac{dV_z}{dh}\bigg|_{h=h_0} - P_{z0}\frac{dA_z}{dh}\bigg|_{h=h_0} + K_B, \tag{12}$$

where $\frac{dV_z}{dh}$ – rate of change of the effective volume; $\frac{dA_z}{dh}$ – rate of change of the effective area; K_B – rigidity of the air spring shell; k – polytrope indicator.

Thus, the analysis of the carried out works showed a significant number of approaches to determining the characteristics of rigidity and dissipation of air springs. However, the issue of discrepancy between the values of spring rigidity during its compression and expansion was not analyzed. Also, the considered models did not take into account the phenomenon of heat exchange between the spring and the environment.

3. The purpose and tasks of the study

The purpose of this work is to study the dynamic rigidity of an air spring at different modes of its operation and energy dissipation depending on the design and parameters of the connecting element and the frequency spectrum of system oscillations.

Tasks of the study:

– analysis of the developed models and approaches to determining the characteristics of an air spring;

– obtaining the dependences of the dynamic rigidity of an air spring on the deformation at different modes of its operation (compression and expansion);

– research of the relationships between dynamic rigidity, energy dissipation per cycle, damping coefficient and perturbation force frequency;

– assessment of the influence of the diameter of the connecting element on the characteristics of the spring.

4. Determination of the parameters of an air spring

Determination of the main parameters of an air spring, namely dynamic rigidity, energy dissipation per cycle and damping coefficient, previously involves obtaining the dependence of force on deformation (**Figure 2**).

From the obtained dependence «force-deformation», rigidity and damping coefficient are determined by the formulas:

$$S = \frac{F_0}{x_0}, \tag{13}$$

$$D = \frac{E}{F_0 \cdot x_0}, \tag{14}$$

where F_0 – amplitude of force; x_0 – amplitude of deformation; E – energy dissipation per cycle.

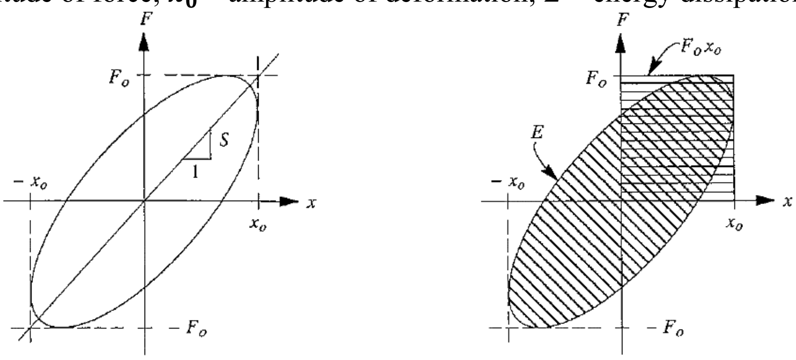

Figure 2. Dependence of force on deformation of an air spring for determining rigidity and damping

When constructing the «force-deformation» dependence, the ideal behavior of gas was considered, where the force was determined by the internal pressure of the air spring:

$$\dot{P}_1(t) = -\dot{h}(t)\frac{P(t)}{h(t)} + \dot{m}(t)\frac{R \cdot T(t)}{h(t) \cdot A_{a.s.}} + \dot{T}(t)\frac{m(t) \cdot R}{h(t) \cdot A_{a.s.}}, \quad (15)$$

where $\boldsymbol{P}, \boldsymbol{V}, \boldsymbol{T}$ – pressure, volume and temperature of the working medium of the air spring, respectively; $\boldsymbol{m}$ – mass of air; $\boldsymbol{R}$ – universal gas constant; $\boldsymbol{h(t)}$ – height of the air spring in the modeling process; $\boldsymbol{A_{a.s.}}$ – cross-sectional area of the air spring.

The dissipation of energy per cycle during the operation of the air spring is characterized by the area of the figure limited by the graph of the dependence «force-deformation» (**Figure 2**).

5. Analysis of the obtained results

Study of the parameters of an air spring were carried out with the following initial data: perturbing force frequency f = 0…15 Hz; diameter of the connecting element d = 0…40 mm and length l = 0,5 m, respectively; initial pressure of the air spring p_0 = 0,7 MPa; amplitude value of the sinusoidal irregularity of the rail track H = 10 mm; universal gas constant R = 287 J/(kg·K); specific heat for the process of constant volume c_v = 718 J/(kg·K) and pressure c_p = 1005 J/(kg·K), respectively. The solving of differential equations of the mathematical model of an air spring was performed in the system of computer algebra «Mathcad». Based on the produced results, the "force-deformation" characteristics of the air spring were built and the dependences of dynamic rigidity during compression and expansion, energy dissipation per cycle, damping coefficient on the perturbing force frequency and the diameter of the connecting element were obtained (**Figure 3-8**).

From the obtained dependence «force-deformation», rigidity and damping coefficient are determined by the formulas:

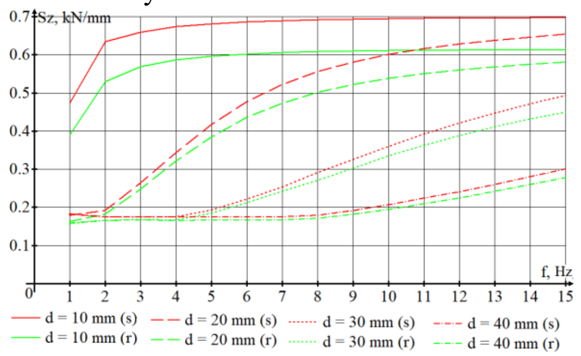

Figure 3. Dependence of rigidity of the air spring on frequency of perturbing force

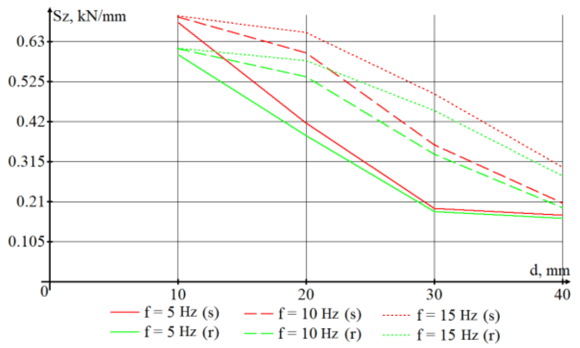

Figure 4. Dependence of rigidity of the air spring on diameter of the connecting element

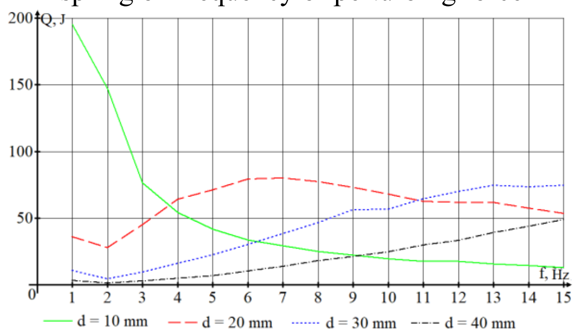

Figure 5. Dependence of energy dissipation of the air spring on frequency of perturbing force

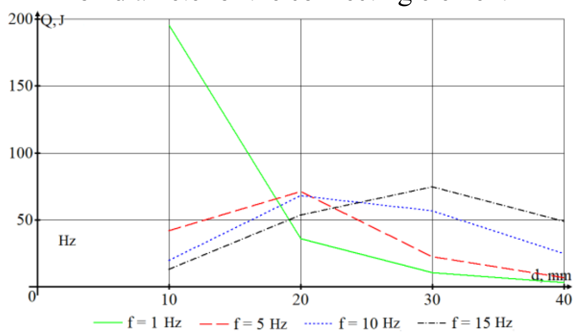

Figure 6. Dependence of energy dissipation of the air spring on the diameter of the connecting element

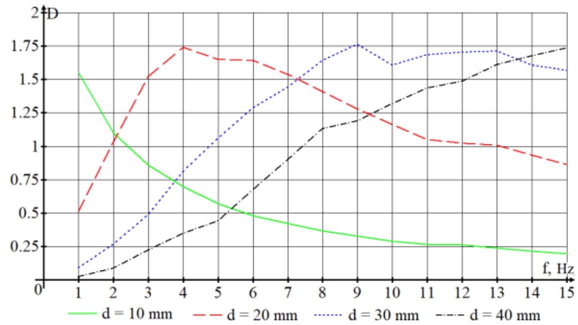

Figure 7. Dependence of damping coefficient of the air spring on frequency of perturbing force

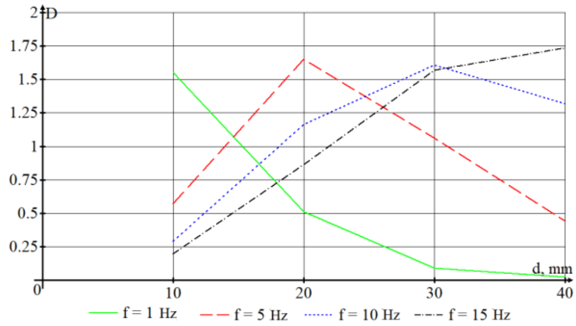

Figure 8. Dependence of damping coefficient of the air spring on the diameter of the connecting element

From the analysis of (**Figure 3**), it is established that the dynamic rigidity of an air spring during its compression is bigger than during expansion in the considered range of frequencies. As the diameter of the connecting element increases, the frequency to which the spring rigidity corresponds to static one (at d = 20 mm - to the frequency of 2 Hz, at d = 30 mm - to the frequency of 4 Hz, at d = 40 mm - to the frequency of 8 Hz) rises. Also, the rigidity of the air spring decreases. With the constant diameter of the connecting element, the increase in frequency f leads to a greater difference between the rigidity of the air spring when it is compressed and expanded.

The graphs in (**Figure 4**) confirm the previously made conclusion that as the diameter of the connecting element increases, the rigidity of the air spring decreases. As the diameter increases, the difference between rigidity at compression and expansion decreases.

Analysis of the correlations in (**Figure 5-6**) shows that with increasing frequency of the perturbing force, the energy dissipation of the air spring increases to a certain maximum value and decreases with further rise of f. The obtained maximum energy dissipation depends on the diameter of the connecting element. The smaller the diameter, the lower the frequencies at which the maximum energy dissipation is observed. If the diameter of the connecting element is small enough (up to 10 mm), the air spring will effectively dissipate energy only at low oscillation frequencies (up to 5 Hz). At higher frequencies $f > 5$ Hz, the flow of air into the additional reservoir is virtually stopped and energy dissipation is significantly reduced $Q < 40$ J. At the frequency of oscillations $f = 4$ Hz, the most effective is an air spring with a diameter of the connecting element $d = 10 \div 20$ mm, which corresponds to the results of work [16], and a spring with d = 40 mm virtually does not provide the damping of oscillations. In the frequency range of the perturbing force from 5 to 15 Hz, the change in diameter in the range from 20 to 30 mm has almost no effect on the maximum dissipation value.

The dependences of damping coefficient on f and d (**Figure 7-8**) are similar in nature to the dependences obtained for energy dissipation. They also have a marked maximum of damping coefficient, shifted in frequency, depending on the diameter of the connecting element. But in contrast to energy dissipation, the value of the maximum damping coefficient is almost the same for all calculated cases.

6. Conclusions

Approaches to determining the characteristics of an air spring were considered in the work. On the basis of the obtained dependences of the force on the deformation, the dynamic rigidity of the spring during compression and expansion, the dissipation of energy by the spring and the damping coefficient were determined. The dependences of the listed parameters on the frequency of the perturbing force and the diameter of the connecting element of the spring with an additional reservoir were investigated.

It was shown that the dynamic rigidity of the spring during compression and expansion is different, and in the considered frequency range the rigidity at compression is greater. A decrease in the diameter or a growth in the frequency of the perturbing force leads to an increase in the discrepancy between the rigidity at compression and the one at expansion. As the diameter of the connecting element increases, the frequency to which the rigidity of the spring corresponds to static one rises. The dependences of the dissipation energy and the damping coefficient on the frequency of the perturbing force have maxima that occur at lower frequencies for smaller diameters of the connecting element.

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