Results of theoretical and experimental research of pneumatic suspension with internal throttling of working fluid

K Chernyshov¹, V Novikov¹, A Pozdeev¹,²,³, I Ryabov¹, A Diakov², K Evseev² and D Chumakov¹

¹Volgograd State Technical University, Volgograd, Russian Federation
²Bauman Moscow State Technical University
³E-mail: pozdeev.vstu@gmail.com

Abstract. The article presents the mathematical model and experimental device of the air suspension installed on the dynamic stand, which imitates the work of the unsupported vehicle suspension. The air suspension is presented in the form of a two-chamber air compressor, the working and auxiliary cavities of which are communicated to each other through a throttle with a check valve open during compression. The design and experimental amplitude-frequency characteristics of scales of vertical movements of the suspended mass are presented.

1. Introduction

The mathematical modeling and bench tests are used to study the vibro-protective properties of the air suspension systems of motor vehicles, which allow to check the adequacy of the mathematical model and to reveal the peculiarities of the work of the real suspension systems [1-21]. Special and universal benches are used for testing, but they can make errors in the work of real systems, for example, in the VolgGTU at the department of “Automatic installations” there is a unique test bench BISS-ITW for the study of unsupported vibrating systems [1-3]. The peculiarity of the stand is the presence of a movable crosshead with weights imitating the suspended mass of the car, moved relatively stationary base of the stand with little friction $F_{fr.add}$, which is absent when the real car is moving along the roughness of the road.

In order to take into account the effect of additional friction on the work of the unsupported air suspension installed on the stand, a calculation scheme (Figure 1) and a mathematical model have been developed:

\[
\begin{align*}
Mz_1 + k(z_1 - z_2) - F_{spr} + Mg + F_{fr} \operatorname{sgn}(z_1 - z_2) + F_{fr.add} \operatorname{sgn} z_1 &= 0, \\
m_1 \ddot{z}_2 + k_1(\dot{z}_2 - \dot{q}) + c_1(z_2 - q) - k(z_1 - \dot{z}_2) + F_{spr} - Mg - F_{fr} \operatorname{sgn}(z_1 - \dot{z}_2) - F_{fr.add} \operatorname{sgn} z_1 &= 0.
\end{align*}
\]

Where $F_{spr} = F_t$ – the elasticity force, which depends on the pressure $p_1$ in the main chamber of the rubber-corded air springs (RCS); $g$ – freefall acceleration; $M$ and $m_1$ – suspended and unsprung mass, $k$ – damping factor, which takes into account the internal losses in RCS; $F_{fr}$ – dry friction force in the air suspension; $F_{fr.add}$ – additional dry friction force between the suspended mass and the base of the bench; $k_1$ and $c_1$ – damping factor and tyre rigidity.
Figure 1. Design model of air suspension with internal throttling of working fluid: a - scheme of oscillating system with air suspension; b - scheme of air spring with internal throttling; 1 - air spring with internal throttling; $M$ - sprung weight; $m_t$ - unsprung weight; $k$ - suspension damping coefficient; $F_{tr}$ - dry friction force in the suspension; $c_t$ - tyre stiffness; $k_t$ - tyre damping coefficient; $l_{10}$ and $l_{20}$ - the initial height of the gas column in the main chamber and the additional chamber under static load when the gas mass flows out of it (into it) $Q$; $l_1(t)$ and $l_2(t)$ - the current height of the gas column in the main and additional chambers under static load; $F_{fr}$ - dry friction force in the suspension; $c_t$ - tyre stiffness; $k_t$ - tyre damping coefficient; $l_{10}$ and $l_{20}$ - the initial height of the gas column in the main chamber and the additional chamber under static load when the gas mass flows out of it (into it) $Q$; $l_1(t)$ and $l_2(t)$ - the current height of the gas column in the main and additional chambers under static load; $p_{10}$ and $p_{20}$ - current pressure in the main and additional chambers; $l_0 = l_{10} + l_{20} = l_1(t) + l_2(t)$ - total height of the gas column; $V_{10} = l_{10} \cdot S$, $V_{20} = l_{20} \cdot S$ - corresponding volumes of main and additional chambers under static load; $S$ - piston area; $V_1(t) = (l_{10} + z - q)S$, $V_2(t) = V_{20} = l_2(t) \cdot S$ - the corresponding volume of cameras at the current load; $m_{10}$ и $m_{20}$ - Initial gas mass in the main and additional chambers; $m_0 = m_{10} + m_{20}$ - gas mass in the spring; $T_0$ - Initial gas temperature in the main and additional chambers; $m_1(t)$, $T_1(t)$ and $m_2(t)$, $T_2(t)$ - current weight and temperature of gas in the main and additional chambers; $p_{10}$ и $p_{20}$ - initial pressure in the main and additional chambers under static load; $q$ - vertical displacement of the road profile; $z_1$ - vertical displacement of the sprung mass; $z_2$ - vertical displacement of unsprung mass.

Pressure in the main volume:

$$p_1(t) = \left( \frac{l_1(t)}{z_1 - z_2 + l_{10}} \right)^n \left( \frac{Mg}{S} + p_{atm} \right) - p_{atm}.$$  \hspace{1cm} (2)

Pressure in additional volume:

$$p_2(t) = \left( \frac{l_0 - l_1(t)}{l_0 - l_{10}} \right)^n \left( \frac{Mg}{S} + p_{atm} \right) - p_{atm},$$  \hspace{1cm} (3)

where $p_{atm}$ - atmospheric pressure; $n$ - polytrope index.

The suspension elasticity:

$$F_{spr} = F_1(t) = p_1(t)S.$$  \hspace{1cm} (4)

With $\Delta p = p_1(t) - p_2(t) > 0$ the change in the height of the gas column $l_1$ is determined by the equation:

$$\frac{dl_1(t)}{dt} = -\alpha_g f_0 \cdot \frac{RT_0}{\mu MgS} \left( \frac{p_2(t)}{p_1(t)} \right)^{\frac{1}{n}} \sqrt{\frac{2n}{n-1} \frac{p_1(t) \cdot l_1(t)}{l_{10} + z_1 - z_2} \left( 1 - \left( \frac{p_2(t)}{p_1(t)} \right)^{\frac{n-1}{n}} \right)},$$  \hspace{1cm} (5)
if \( \frac{p_2}{p_1} > \left( \frac{2}{n+1} \right)^{n-1} \) and equation

\[
\frac{dl_1(t)}{dt} = f_0 \cdot \sqrt{RT_0 \mu MgS} \left( \frac{p_1(t)}{p_2(t)} \right)^{n-1} \cdot \frac{2n \cdot p_1(t) \cdot l_1(t)}{l_0 + z - q},
\]

(6)

if \( \frac{p_2}{p_1} \leq \left( \frac{2}{n+1} \right)^{n-1} \).

With \( \Delta p = p_1(t) - p_2(t) < 0 \) the change in the height of the gas column \( l_1 \) is determined by the equation:

\[
\frac{dl_1(t)}{dt} = \alpha g f_0 \cdot \sqrt{RT_0 \mu MgS} \left( \frac{p_1(t)}{p_2(t)} \right)^{n-1} \cdot \frac{2n \cdot p_2(t)(l_0 - l_1(t))}{l_0 - l_{10}} \cdot \sqrt{1 - \left( \frac{p_1(t)}{p_2(t)} \right)^{n-1}},
\]

(7)

if \( \frac{p_1}{p_2} > \left( \frac{2}{n+1} \right)^{n-1} \) and equation

\[
\frac{dl_1(t)}{dt} = f_0 \cdot \sqrt{RT_0 \mu MgS} \left( \frac{2}{n+1} \right)^{n-1} \cdot \frac{2n \cdot p_2(t)(l_0 - l_1(t))}{l_0 - l_{10}},
\]

(8)

if \( \frac{p_2}{p_1} \leq \left( \frac{2}{n+1} \right)^{n-1} \).

Taking into account the operation of the non-return valve in equations (5), (6), (7) and (8) of area \( f_0 \), the following conditions apply:

\[
f_0 = \begin{cases} 
  f_1 + f_2 & \text{if } z_1 \geq z_2 \text{ and } p_1 > p_2 \\
  f_1 & \text{if } z_1 \leq z_2 \text{ and } p_1 \leq p_2 
\end{cases}
\]

(9)

where \( f_1 \) – throttle area; \( f_2 \)– check valve area.

2. Description of the experimental installation

The experimental installation of a double-chamber pneumatic compressor with a rubber-corded casing on the basis of a dynamic stand with a servo-hydraulic drive of the Indian company BISS-ITW (Fig. 2) was assembled to check the adequacy of the mathematical model.

The test air compressor 1 is installed on the test bench between the support plate 3 of the hydraulic impulse 2 and the installation channels of the suspended mass 5. Test pneumatic compressor contains a rubber-cord case of hose type 1 of grade VL 260-340, hollow piston 4 and the top cover with a filling connection. The cavity A of the rubber cord sheath 1 is communicated with the cavity B of the piston 4 through a pneumatic damper, the body 6 of which is mounted on the axis on the top end of the piston 4. Inside housing 6 there is a throttle that constantly communicates the cavities A and B with each other, two rows of radial holes and an elastic check valve that covers these holes in the course of the rebound and communicates the cavity A with the cavity B of the piston 4 only in the course of compression. The upper end of piston 4 is equipped with a buffer of maximum compression stroke 8.
Figure 2. Experimental installation of air suspension on the bench: a - scheme; b - general view; 1 - rubber-cord cover of pneumo compressors; 2 - hydraulic impulse with support plate 3; 4 - hollow piston of pneumo compressors with a buffer of the maximum course of compression 8; 5 - suspended weight; 6 - pneumatic damper with a throttle and a check valve; 7 - directing lever with a swinging axis 9; 10 - unsprung weight; 11 - Spring with rigidity imitating wheel tire; 12 - Compressor unit; 13 - Pressure gauge; 14 - Force sensor; 15 - Movement sensor and speed of the suspended mass; 16 - Position sensor; 17 - Microcontroller; A - Rubber-cord shell cavity Pneumatic springs (main chamber); B - Pneumatic springs piston cavity (additional chamber)

The volume of the air springs above the piston in static position \( V_{st} = 12 \) l, the volume of the extended hollow piston \( V_p = 7.6 \) l. Height of the air springs in static position \( H_{st} = 450 \) mm. Static pressure in the air compressor \( \rho_{st} = 1.2 \) kG/cm\(^2\) (0.12 MPa). Throttle diameter \( d = 6.7 \) mm, \( F_{fr,add} = 10 \) N.

3. Bench testing methodology
The test methodology was as follows.
During the tests, the air compressor had a working volume of 19.6 liters. This volume of communication with the hollow piston cavity, the volume of 7.6 liters, was carried out through a throttle or through a throttle with a check valve open during compression.

During the internal throttling study, a damping unit is installed in the piston of the air springs. After the pneumo compressor is set in a static position, corresponding to its height of 450 mm, by supplying the pneumatic compressor with air from the external compressor. At the same time, the working pressure inside the air compressor is controlled by a pressure gauge, and the height of the air compressor is fixed visually by a measuring scale.

Next, the control panel of the stand is used to set the displacement of the hydraulic impulse rod according to the harmonic law with the kinematic disturbance amplitude \( q_0 = 12 \) mm at frequencies, Hz: 0.5; 0.75; 0.8; 0.9; 1.0; 1.25; 1.5; 1.75; 1.8; 1.9; 2.0; 2.2; 2.3; 2.5; 3.0; 4.0; 5.0; 6.0; 7.0; 8.0. The selected pitch and frequency range allow to determine the resonant frequencies of the vibrations of the suspended and unsprung masses. Harmonic profile is chosen as the heaviest mode of suspension loading, causing steady forced oscillations, and its parameters are close to the profile of a heavily worn and broken road.

4. Analysis of the results of theoretical and experimental research of air suspension with internal throttling
The results of calculation and experiment have been obtained the amplitude-frequency characteristics (AFC) for the suspended mass installed on a pneumatic compressor with internal throttling. The throttling system was made in the form of a throttle with a hole diameter of 6.7 mm and
a check valve, which is made in the form of 12 holes with a diameter of 2 mm and operates on the course of compression.

Figure 3 shows the dependence of the amplitudes of the resonant vibrations of the suspended mass on the ratio of the volumes of additional and main chambers at a constant diameter of the throttle hole. As can be seen from the figure, with the increase in the ratio of the vibration amplitude is reduced in both the first and second resonances.

Figure 4 shows the calculated dependencies of the amplitudes of the resonant vibrations of the suspended mass on the diameter of the throttle hole with a constant ratio of the additional volume to the main volume. The figure shows that for each resonance there is an optimal diameter of the throttle opening, at which the amplitude of the vibrations of the suspended mass is minimal.

Figure 3. Dependence of the vibration amplitudes of the suspended mass at the first (1) and second (2) resonances on the ratio of the volumes of additional and main chambers with $d_{thr} = 4$ mm

Figure 4. Dependence of the vibration amplitudes of the suspended mass at the first (1) and second (2) resonances on the throttle diameter with $V_{rel}= 5/1$

Figures 5 and 6 show the experimental and calculated frequency response of the absolute displacements of the suspended mass on the air suspension with the throttling system, made in the form of a throttle hole and a check valve opened on the course of compression, obtained at the following parameters: $d_{thr} = 6.7$ mm and $V_{RCS} = 12 l$; $V_o = 7.6 l$; $H_a = 450$ mm; $m_t = 128$ kg; $c_t = 300$ kN/m; $k = 1.3$; $T_0 = 323$ K; $p_{atm} = 0.1$ MPa; $g = 9.81$ m/s$^2$; $D_p = 0.25$ m; $R = 297$, vibration amplitudes $q = 3$ and 5 mm.

It can be seen from the graphs that dry friction between the suspended mass and the base of the stand leads to a decrease in the amplitude of low-frequency resonance by 13...47% for kinematic perturbation $q = 3$ and 5 mm.

5. Conclusions
   - The analysis of the calculated and experimental results shows that the mathematical model quite adequately describes the investigated oscillating system with air suspension and air damping system in the form of a throttle with a check valve.
   - The presence of even a small dry friction leads to a decrease in the amplitude of low-frequency resonance by 13...47%.
   - The amplitude of oscillations decreases in both the first and second resonances as the ratio of additional and basic volumes increases.
   - For each resonance, there is an optimum throttle diameter at which the amplitude of the suspended mass is minimized.
Figure 5. AFC of the vertical displacements of the suspended mass $M = 0.5 \text{ t}$ on the air suspension with internal air throttling at the kinematic perturbation amplitude $q = 3 \text{ mm}$, 1 - experiment; 2 - calculation taking into account additional friction; 3 - calculation without taking into account additional friction; $F_r = 68 \text{ N}$; $F_{r,\text{add}} = 10 \text{ N}$; $k_d = 590 \text{ N} \cdot \text{s/m}$

Figure 6. AFC of the vertical displacements of the suspended mass $M = 0.5 \text{ t}$ on the air suspension with internal air throttling at the kinematic perturbation amplitude $q = 5 \text{ mm}$, 1 - experiment; 2 - calculation taking into account additional friction; 3 - calculation without taking into account additional friction; $F_r = 107 \text{ N}$; $F_{r,\text{add}} = 20 \text{ N}$; $k_d = 300 \text{ N} \cdot \text{s/m}$

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