

## INVESTIGATION OF DAMPING PROPERTIES OF TELESCOPIC SHOCK ABSORBER AND ITS RUBBER MOUNTS IN FREQUENCY RANGE 50-150 HZ

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**Abstract.** The pneumatic tyre is excited by the road unevenness at a frequency close to its first radial mode for the speeds of 50-60 km·h<sup>-1</sup>. The first resonance of the belt of a radial-type pneumatic tyre has a frequency about 100 Hz. These oscillations are associated with amplitudes of displacement less than 1 mm. The vibrations at the resonance frequency go through the suspension elements and reach the vehicle body without noticeably reducing their amplitude. These high frequency vibrations have a major role in the vibro-acoustic comfort in the passenger compartment of a vehicle. The telescopic shock absorber is one of the suspension components. In the frequency range up to 20 Hz, it has a fundamental role in damping the oscillations generated by the road unevenness. In the frequency range of the natural modes of both sprung and unsprung masses (0.9-12 Hz), the displacements between both ends of a shock absorber are over 5 mm. High-frequency vibrations with an amplitude less than 1 mm do not activate the shock absorber damping properties. The amplitudes are not big enough to create the necessary pressure drop between the two chambers above and under the piston and to provide viscous damping. Other elements that have an influence on damping vibrations with frequencies above 50 Hz are the rubber mounts of the shock absorber. The work explores the behaviour of a telescopic shock absorber and its rubber mounts in the frequency domain determined by the resonances of the pneumatic tyre belt. A method for obtaining the damping coefficient of the rubber mounts for the shock absorber is presented. The purpose of the work is to determine the influence of the damping properties of the shock absorber and its rubber mounts on the vibrational behaviour of the suspension in the frequency range 50-150 Hz. The results of the work can be used to select a proper shock absorber and its rubber mounts, in order to improve the vibro-acoustic comfort of the passenger compartment.

**Keywords:** shock absorber, damper, rubber mount, bushing, insulator.

### Introduction

The role of the shock absorber is to provide better comfort for the vehicle by damping the relative movement between the wheel axle and the body (frame). The shock absorber is active in the frequency range up to 20 Hz. Higher frequency oscillations are primarily damped by the pneumatic tyre and by the rubber mounts in the suspension elements [1]. At the resonance frequency of the tyre belt, the vibrations from the road unevenness are transmitted to the suspension elements without noticeably lowering their levels [2]. The connection between the shock absorber and the body (frame) is a main path through which structure-borne vibrations are transmitted. The frequency range of the structure-borne vibrations, caused by the contact between the tyre and the road surface, is between 50 and 300 Hz [3]. Tyre vibrations may be propagated via the suspension to the windows, body and door panels. The oscillation of the body panels creates noise in the passenger compartment. In [4] the propagation of vibrations through the shock absorber for the frequency range 200-800 Hz is studied.

In [5] it is established that the oscillations measured in the wheel axle pass through the bearings and rigid joints without lowering their levels. Fig. 1 presents a frequency response of vertical acceleration in the wheel axle. It is obtained by radial excitation of the pneumatic tyre with the electrodynamic shaker. The shaker maintains steady harmonic acceleration of 10 g. The maximum values of the vertical acceleration correspond to the first two radial modes of the pneumatic tyre.

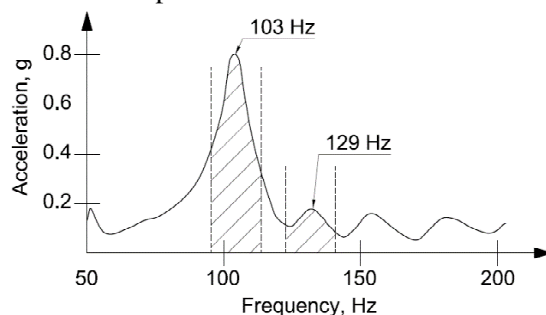


Fig. 1. Frequency response of vertical acceleration in wheel axle

The purpose of this study is to investigate the vibration damping in a shock absorber and its rubber mounts for the frequency range corresponding to the first two resonances of the pneumatic tyre. This frequency domain is 50-150 Hz. The study of the shock absorber behaviour in this frequency domain is necessary for its inclusion as an element in mechanical/mathematical models of the suspension or those built with finite elements.

**Materials and methods**

The investigated shock absorber is a gas-charged mono-tube with the gas chamber separated from the compression chamber by a sealing piston. The design of the shock absorber is shown in Figure 2.

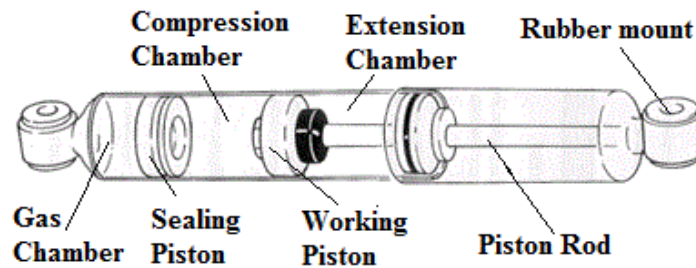


Fig. 2. Design of gas-charged mono-tube shock absorber

Hydraulic oil with a viscosity similar to that of the original oil is used in the shock absorber. In Figure 3 the viscosities of the original and the oil used in the study as a function of the temperature are plotted. The graph is obtained experimentally according to the requirements of ISO 3104 + AC.

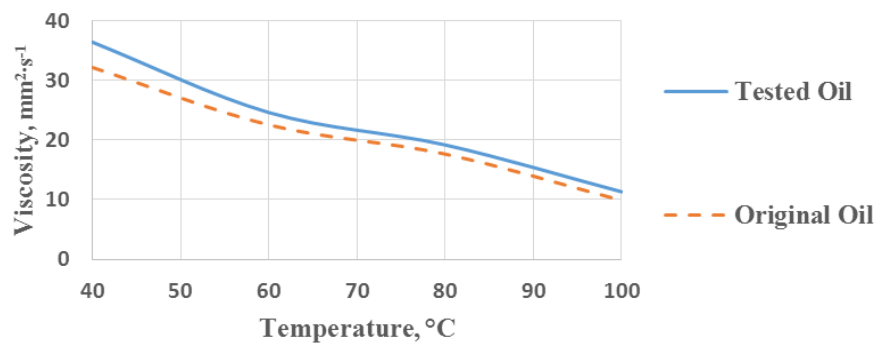


Fig. 3. Viscosity of original oil and that used in experiments

The following conditions are met when performing experiments on the shock absorber: the temperature of the shock absorber's body is in the limits of 50-60 °C; the room temperature is 25 °C; the oil used in the shock absorber provides damper characteristics similar to the original ones for the temperature range 50-60 °C (Fig. 4); the original rubber mounts are kept in; accuracy of the measuring equipment less than 1 %.

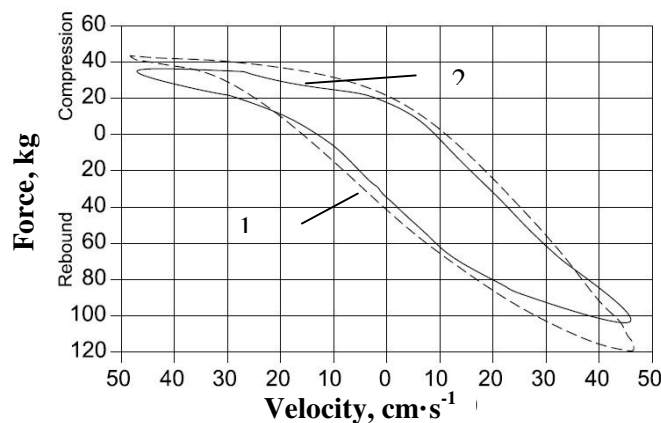


Fig. 4. Force in shock absorber as function of speed  $F(v)$ : 1 –  $F(v)$  for the initial shock absorber; 2 –  $F(v)$  for the tested shock absorber

### Assessment of the condition of the tested shock absorber

The performance of the initial and tested shock absorber is compared for the frequency range up to 5 Hz. The shock absorber is tested on a “Shock Dyno” (the Intercomp Variable Speed Shock Dynamometer, position 1 in Figure 5) for speeds from 1 to 50  $\text{cm}\cdot\text{s}^{-1}$  with 5 cm stroke. In Figure 4 the damping force of the two shock absorbers for the speed 50  $\text{cm}\cdot\text{s}^{-1}$  is compared. The hydraulic pressures in the compression and extension chambers are measured for a static pressure of 5 Bar in the gas chamber.

### Method for experimental investigation of the shock absorber in the frequency range above 20 Hz

The hydrodynamic shaker, shown in Figure 5 (position 2), is used to study the behaviour of the shock absorber at vibrations above 20 Hz.

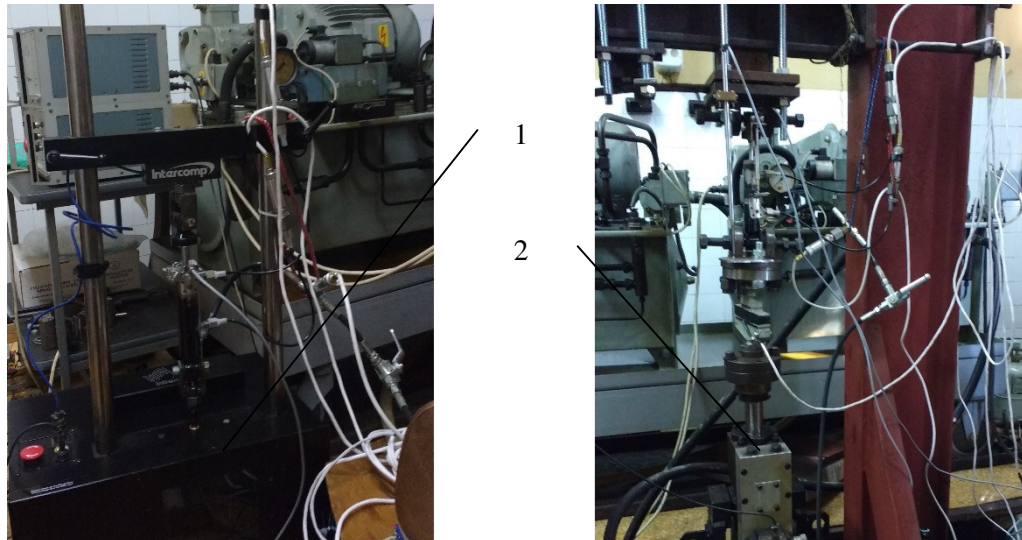


Fig. 5. **Equipment for testing shock absorbers:** 1 – electrodynamic shaker “Shock Dyno”;  
2 – hydrodynamic shaker

The shock absorber is mounted on a hydrodynamic bench with potential to control the frequency and amplitude of excitation. The lower shock absorber rubber mount is attached to the vibrating plate 10 of the shaker (Fig. 6).

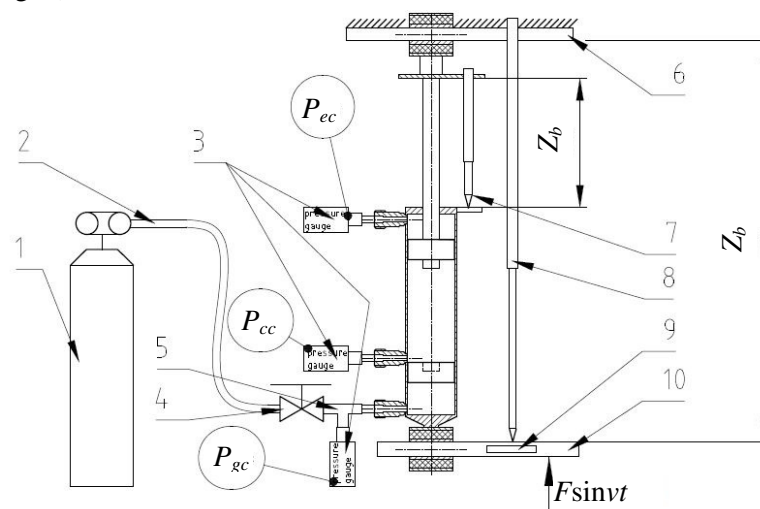


Fig. 6. **Layout of testing system:** 1 – gas bottle; 2 – pipeline; 3 – pressure transducer; 4 – stopcock;  
5 – three-ways connector; 6 – fixed support; 7, 8 – displacement transducer;  
9 – strain transducer; 10 – shaker plate

The upper one is fixed to the support 6. A full bridge strain transducer 9 is mounted on the vibrating plate on the bench, measuring the force applied to the shock absorber. Pressure transducers 3

measure the pressures in the three chambers of the shock absorber – in the extension chamber  $P_{ec}$ , in the compression chamber  $P_{cc}$  and in the gas chamber  $P_{gc}$ . The relative movement between the piston rod and the cylinder tube is measured by the displacement transducer 7 ( $Z_a$  in Fig. 6). The full stroke of the shock absorber between its mounting points ( $Z_b$  in Fig. 6) is measured by the displacement transducer 8. The system is complemented by a gas bottle 1, a stopcock 4, a three-ways connector 5 and different lengths of pipelines. They allow different pressure to be applied to the gas chamber of the shock absorber.

The sequence of performing the measurements is as follows:

- The hydraulic shaker applies a different frequency and amplitude excitation to the shock absorber’s lower mount – the experimental plan includes tests at frequencies 20 Hz, 50 Hz, 70 Hz and 100 Hz.
- For each frequency, the pressures  $P_{ec}$ ,  $P_{cc}$ ,  $P_{gc}$  in the three chambers of the shock absorber and the displacements  $Z_a$ ,  $Z_b$  are measured.

The experimentally obtained data are processed and the piston force, due to the pressure difference below and above the piston, is calculated [6]:

$$F_{pc} = P_{cc} A_p - P_{ec} A_{pa} \tag{1}$$

$$F_{pe} = P_{ec} A_{pa} - P_{cc} A_p, \tag{2}$$

where  $F_{pc}$ ,  $F_{pe}$  – respectively the piston force in compression and extension, N;  
 $P_{cc}$ ,  $P_{ec}$  – respectively the pressure in the compression and extension chamber, Pa;  
 $A_p$ ,  $A_{pa}$  – respectively the piston and piston annulus area, m<sup>2</sup>.

A comparison between the displacements  $Z_a$ ,  $Z_b$  is made. Also an estimation of the damping in the shock absorber and that in its rubber mounts is done.

*Method for experimentally determining the coefficients of stiffness and damping of rubber insulators*

The method for experimentally determining the coefficients of stiffness and damping of rubber insulators is based on a comparative method of two dynamic systems presented in Figure 7. In the first dynamic system (A), the mass  $m_b$  and coefficients of stiffness  $c_b$  and damping  $\beta_b$  are known. To create the second system (B), three new parameters are added, from which the mass  $m_v$  is known and the coefficients of damping  $\beta_v$  and stiffness  $c_v$  are unknown. The first system (A), which is called “spring – mass”, is a simplified model of the free end of a cantilever beam. The second system (B), which is called “spring – mass – insulator”, is created by installing a fixed rubber insulator to the free end of the beam. The unknown coefficients  $\beta_v$  and  $c_v$  are respectively the dynamic coefficients of damping and stiffness of the rubber insulator.

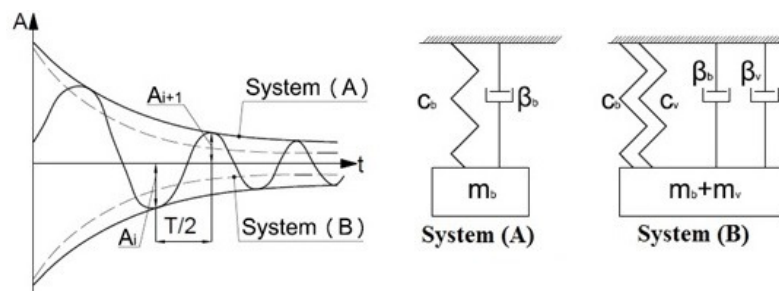


Fig. 7. Time-domain responses of dynamic systems (A) and (B)

By using the relationship between the described parameters in the case of free damped oscillations, the unknown coefficients  $\beta_v$  and  $c_v$  can be determined. For free damped oscillations, the following can be written [7]:

$$\tau = \frac{2\pi}{\Omega} = \frac{2\pi}{\sqrt{\omega^2 - n^2}} \tag{3}$$

where  $\tau$  – period of the free damped oscillations, s;

$$\begin{aligned} \Omega & - \text{frequency of the free damped oscillations, rad}\cdot\text{s}^{-1}; \\ \omega & = \sqrt{\frac{c}{m}} - \text{natural frequency of the dynamic system, rad}\cdot\text{s}^{-1}; \\ n & = \frac{\beta}{2m} - \text{coefficient that parameterizes the strength of the damping, rad}\cdot\text{s}^{-1}. \end{aligned}$$

For the specific dynamic systems (A) and (B)  $n < \omega$ , so in this case the damped oscillations are periodic. Position 3 in Figure 7 is the time-domain response of the dynamic systems after initial perturbation. The continuous line record (position 3 in Figure 7) corresponds to the free damped oscillations of system (A) and the dashed line record corresponds to the free damped oscillations of system (B). Changing the masses of the systems, also their frequency of oscillation changes.

For each system, the logarithmic decrement is determined. The logarithmic decrement is defined as the natural logarithm of the ratio of any two successive amplitudes:

$$\delta = \ln \frac{A_i}{A_{i+1}} = \frac{n\tau}{2} = \frac{\pi n}{\sqrt{\omega^2 - n^2}} \quad (4)$$

The logarithmic decrement characterizes damping in the dynamic systems. Possessing the time-domain responses of the two systems (A) and (B), the dynamic coefficients of damping and stiffness of the rubber insulator can be determined from equations (3) and (4). The coefficients of damping  $\beta_v$  and stiffness  $c_v$  are respectively:

$$\beta_v = \frac{4m_\Sigma \delta_\Sigma}{\tau_\Sigma} - \frac{4m_b \delta_b}{\tau_b} \quad (5)$$

$$c_v = \frac{4m_\Sigma}{\tau_\Sigma^2} (\delta_\Sigma^2 + \pi^2) - \frac{4m_b}{\tau_b^2} (\delta_b^2 + \pi^2) \quad (6)$$

where  $m_b, m_\Sigma$  – respectively the mass of the system (A) “spring – mass” and the mass of the system (B) “spring – mass – insulator” (Fig. 7), kg;  
 $\tau_b, \tau_\Sigma$  – respectively the periods of oscillation of system (A) and system (B), s;  
 $\delta_b$  and  $\delta_\Sigma$  – respectively the logarithmic decrement of system (A) and system (B);  
 $\beta_v$  – dynamic damping coefficient of the rubber insulator, kg·s<sup>-1</sup>;  
 $c_v$  – dynamic stiffness coefficient of the rubber insulator, kg·s<sup>-2</sup>.

#### *Method for numerical investigation of the shock absorber in the frequency range 50-150 Hz*

A model of the shock absorber based on the Finite Element Method (FEM) is built in SolidWorks Simulation software. It is constructed of three main elements: a rigidly connected piston and piston rod, a housing and a sealing piston, separating the compression from the gas chamber. The compressibility of the oil and gas is represented by springs between the individual elements. The upper mounting point is attached with radially distributed springs, without their own mass, to a fixed axis. The stiffness of the springs has a value corresponding to the stiffness coefficient of the rubber mount. A harmonic vertical excitation is applied to the lower mount of the shock absorber within the 50-150 Hz frequency range. The excitation is an acceleration of 0.35 g amplitude value. This value is experimentally obtained at the lower mounting point of the shock absorber during radial harmonic disturbance of 10 gin the contact patch of a pneumatic tyre. The frequency response of acceleration in a point at the top of the piston rod is obtained. For the purpose of the study, the frequency response analysis called Harmonic study in the SolidWorks Simulation software has been used.

## **Results and discussion**

### *Experimental results*

In Figure 8 the forces on the piston in compression  $F_{pc}$  and extension  $F_{pe}$  as a function of frequency are shown. The dashed line represents the maximum force obtained in extension, and the

continuous line – the maximum force in compression. It is noted that after 70 Hz the force in extension has negligible values, so it can be considered that only the gas-filled chamber remains active and the fluid acts on the sealing piston as a solid body.

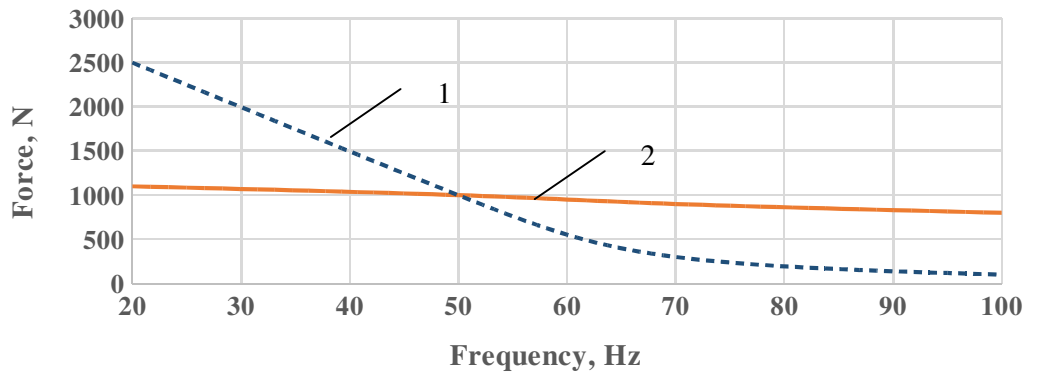


Fig. 8. **Piston force as function of frequency:** 1 – piston force in extension; 2 – piston force in compression

In Figure 9 the two measured displacements  $Z_a$  and  $Z_b$  at the 20 Hz frequency are presented. The dashed line represents the relative displacement  $Z_b$  of the two mounting points of the shock absorber. The continuous line represents the relative displacement  $Z_a$  between the piston rod and the cylinder tube. It is noted that the oscillation absorption is realized also in the rubber insulators at both mounting points, besides that in the shock absorber. In Figure 10 the two measured displacements are presented as a function of the frequency. Given that the system has limited power, the increase in frequency itself leads to a reduction in the displacements.

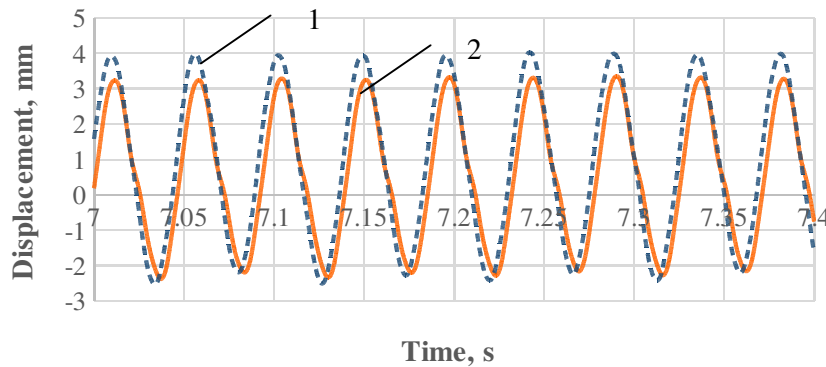


Fig. 9. **Time record of displacements:** 1 – relative displacement  $Z_b$  between the two mounting points of the shock absorber; 2 – relative displacement  $Z_a$  between the piston rod and the cylinder tube

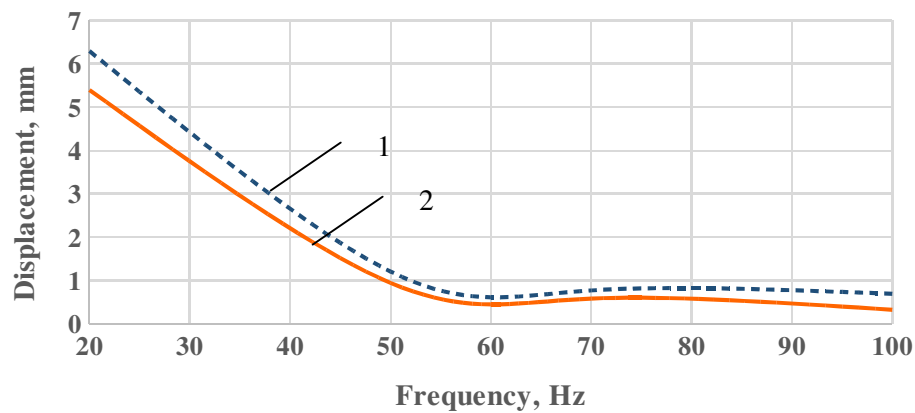


Fig. 10. **Displacements as function of frequency:** 1 – relative displacement  $Z_b$  between the two mounting points of the shock absorber; 2 – relative displacement  $Z_a$  between the piston rod and the cylinder tube



The damping and the stiffness coefficients of the rubber mounts have non-linear behaviour and they depend on the frequency of vibration and the preload on them. In Figure 11 the damping and the stiffness coefficients for the rubber insulators in the shock absorber mounts are shown. It is noted that the stiffness coefficient increases with increasing the frequency and the preload. The damping coefficient decreases as the frequency increases.

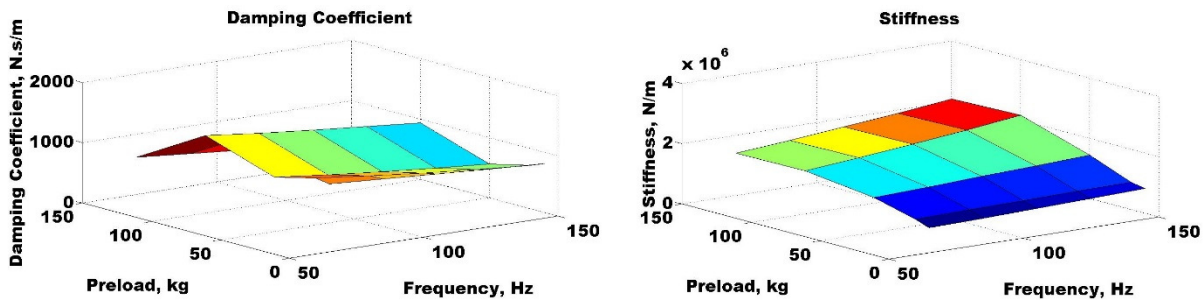


Fig. 11. Coefficients of damping and stiffness for rubber insulator in shock absorber mount

Results of numerical study

The experimental results show that the displacements above 50 Hz are not big enough to create the necessary pressure drop between the two chambers above and under the piston and to provide viscous damping. In that case the shock absorber can be modelled as a rod (dynamic) system composed of elements with distributed parameters (mass, stiffness). In Figure 12 is shown the frequency response function of a resultant acceleration for a point from the upper end of the piston rod. For the purpose of the study the Frequency response analysis called Harmonic study in the SolidWorks Simulation software has been used. The harmonic excitation is applied at the bottom mount of the shock absorber (acceleration 0.35 g in the vertical direction). The metal elements are made of Plain Carbon Steel. The Elastic modulus for this steel is  $E = 2.1 \cdot 10^{11} \text{ N}\cdot\text{m}^{-2}$ , the Poisson's ratio is  $\mu = 0.28$  and the mass density is  $\rho = 7800 \text{ kg}\cdot\text{m}^{-3}$ . The compressibility of the oil is represented by springs with total normal stiffness of  $c_o = 7 \cdot 10^6 \text{ N}\cdot\text{m}^{-1}$  between the individual elements (Spring Connection). The compressibility of the gas is represented by springs with total normal stiffness of  $c_g = 200 \text{ N}\cdot\text{m}^{-1}$  between the housing and the sealing piston. The upper mount has total radial stiffness of  $c_m = 7 \cdot 10^5 \text{ N}\cdot\text{m}^{-1}$ . The mesh is constructed by linear tetrahedral solid elements with a total size of 5 mm and tolerance of 0.25 mm. There is a peak in the resultant accelerations at a frequency of 95-100 Hz, which corresponds to a resonance with a natural mode of the shock absorber (Fig. 12). This peak is close to the first resonance of the pneumatic tyre.

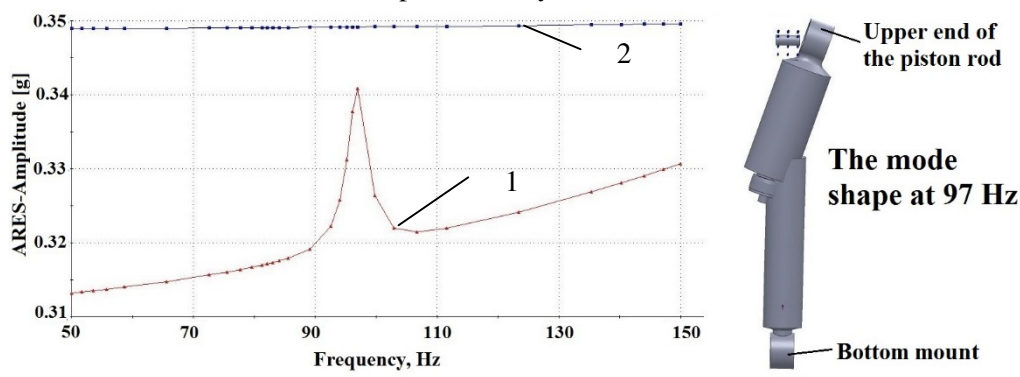


Fig. 12. Frequency response function: 1 – frequency response of acceleration for a point from the upper end of the piston rod; 2 – harmonic excitation at the bottom mount

Conclusions

1. In the frequency range above 50 Hz, the valve system in the shock absorber piston is not active. The damping of the vibration is done by compressing air in the gas chamber and by the rubber insulators in the upper and bottom mount.

2. In the frequency range above 50 Hz, the shock absorber can be presented as a rod system in mechanical/mathematical models and those with finite elements. The rod system is composed of three separate parts (cylinder tube, rigidly connected piston and piston rod, sealing piston) connected by springs. The connecting springs have parameters determined by the stiffness (compressibility) of the fluid between the elements and the stiffness of the disc springs in the piston.
3. Rubber insulators in the shock absorber mounts play a major role in damping high frequency oscillations. Their damping and stiffness coefficients are non-linear and depend on the frequency of vibration.
4. The modeling of the shock absorber as a rod system shows that its own natural modes can be with similar frequency to the natural modes of the pneumatic tyre.

### Acknowledgements

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