

STUDY OF VIBRATIONAL BEHAVIOUR OF SHOCK ABSORBER IN HARSHNESS FREQUENCY RANGE 20-100 HZ

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Abstract. The main task of the shock absorber is to provide good comfort to the vehicle by damping the relative movement between the wheel axle and the body. The damping force in the shock absorber is created by the viscous resistance of the working oil as it passes through the valve systems between the various chambers. At low frequencies (below 1 Hz) and large relative displacements of the shock absorber (over 5 mm), the hysteresis of the gas filled in the shock absorber has a slight damping effect also. The fluid and gas forces are the main components of the total force that acts in the damper during its compression and extension in the frequency range up to 20 Hz. These forces counteract the disturbing forces of road unevenness and reduce the level of vibrations that are transmitted to the body at the mounting points. Road disturbances with a frequency above 20 Hz are characterized by small displacements (amplitudes). Small displacements of the piston rod (less than 1 mm) do not provide the necessary pressure in the chambers to overcome the resistance of the valve springs and actuate the fluid damping. In this frequency range, the compressibility of the working fluid also affects the force in the shock absorber. The work examines the vibrational behaviour of a telescopic shock absorber in the frequency range 20-100 Hz – known as the harshness range in NVH (Noise, Vibration and Harshness) studies. For this purpose, a hydrodynamic test bench is used, which is equipped with a force sensor. The pressures in each chamber of the shock absorber are measured. The pressure in the extension chamber is the indicator of the damping force. The results show that the fluid damping of the shock absorber decreases with the increase of the disturbance frequency. Above 50 Hz, the vibrations passing through the shock absorber are damped only by its rubber mounts. The force transmitted through the shock absorber increases and it is linearly proportional to vibration accelerations.

Keywords: shock absorber, harshness, damper, structure-borne noise, high frequency vibrations.

Introduction

Harshness results when the vehicle is unable to absorb vibrations produced by road conditions [1; 2]. It can be considered as the body vertical acceleration in the frequency range over 20 Hz until 100 Hz [3]. In this frequency range, the main elements of the suspension – coil springs and shock absorbers vibrate as solid bodies with distributed parameters [4-6]. The main concern in terms of vehicle comfort is when the natural frequencies of the tyre and the shock absorbers overlap [7]. This leads to increase in the levels of structure-borne noise in the vehicle cabin. The rubber mounts are the only elements of the suspension that in this frequency range have elastic and damping properties [8]. The damping of the shock absorber depends both on the characteristics of the valves in the piston and on the characteristics of the oil with which it is filled [9; 10]. Indicative of the performance of the shock absorber are its operating diagrams. They are obtained on hydraulic and electrodynamic stands equipped with the necessary transducers of force, displacement and velocity [11-13].

The operating diagrams of telescopic shock absorbers represent the force in the shock absorber depending on the displacement and the speed of its piston rod. The influence of various factors on this force can be represented by the following formula [14]:

$$F_{DE} = -F_G + K_D X_{DE} + C_{DE} V_{DE} + (m + C_A) \frac{dV_{DE}}{dt}, \quad (1)$$

where F_g – damper gas compression force, N;
 K_D – damper stiffness (due to gas pressure), $N \cdot m^{-1}$;
 X_{DE} – damper extension displacement, m;
 C_{DE} – damper coefficient in extension, $Ns \cdot m^{-1}$;
 V_{DE} – damper extension velocity, $m \cdot s^{-1}$;
 m – moving damper mass, kg;
 C_A – residual acceleration coefficient, kg.

The static and dynamic gas forces (the first two addends in equation (1)) have a slight influence on the force in the damper in the frequency range below 2 Hz. Viscous damping (the third addend in equation (1)) largely determines the total force in the damper in the frequency range below 20 Hz. As the disturbance frequency increases, the displacements of the piston rod decrease, as does the pressure

difference between the chambers of the shock absorber. There are limit values of the pressure difference below at which the preloaded valves in the piston cannot be activated and in practice viscous damping does not occur. At disturbance frequencies above 20 Hz, the acceleration of the piston and the compressibility of the fluid determine the force in the shock absorber. Compressibility reduces the flow rate through the piston, reducing the pressure drop and thus the viscous damping force. The compressibility is a main component of the residual acceleration coefficient:

$$C_A = -k_{PE}^2 A_{PA}^2 \beta_E \Lambda_{EC}, \quad (2)$$

where k_{PE} – resistance coefficient of piston extension valve, Pa·s·m⁻³;
 A_{PA} – piston annulus area, m²;
 β_E – compressibility of the oil, Pa⁻¹;
 Λ_{EC} – volume in the extension chamber, m³;

If the piston is moving at constant speed, then the pressures will be constant. In this case, the density of the oil will be constant, so the compressibility will have no influence. The effects of compressibility will only be observed in transient processes, and in particular related to acceleration of the piston. The compressibility of the oil is defined as:

$$\beta_E = -\frac{1}{\Lambda} \frac{d\Lambda}{dP} = -\frac{1}{\Lambda} \frac{d\Lambda}{dt} \frac{dt}{dP}, \quad (3)$$

where P – pressure of the oil, Pa;
 t – time, s;

Therefore, the rate of change in the volume in the pressure chamber due to the compressibility and the pressure change will be:

$$\frac{d\Lambda}{dt} = -\Lambda \beta_E \frac{dP}{dt}, \quad (4)$$

The pressure is related to the piston speed through the valve characteristics. This means that the rate of the pressure change is related to acceleration of the piston dV/dt . For a linear valve:

$$P = -kAV, \quad (5)$$

$$\frac{dP}{dt} = kA \frac{dV}{dt}, \quad (6)$$

The rate of change in the volume will therefore be:

$$\frac{d\Lambda}{dt} = -\Lambda \beta_E kA \frac{dV}{dt}, \quad (7)$$

Hence acceleration creates a compressible volume change which causes a flow deficit through the valves, which reduces the force compared with the incompressible case. In short, the compressibility of the fluid creates a relationship between the force in the damper and acceleration of the damper piston.

The purpose of the article is to establish experimentally the nature of the relationship between the force in the shock absorber and acceleration of the piston for the harshness frequency range of 20-100 Hz.

Materials and methods

The studied shock absorber is a single tube with a gas chamber separated from the compression chamber by a solid (plastic) floating piston. The construction of the shock absorber is presented in Figure 1. To assemble the test object, the hydraulic oil in the shock absorber is initially removed. Next, the necessary fittings to connect the pressure sensors are installed. The compression and rebound (extension) chambers are refilled with hydraulic oil before the pressure sensors are installed.

The experiments are carried out in the following condition: the shock absorber body temperature is kept within 50-60 °C; room temperature is 25 °C; to refill the shock absorber is used an oil with close to the original viscosity; the (accuracy) error of the measuring equipment is less than 1%.

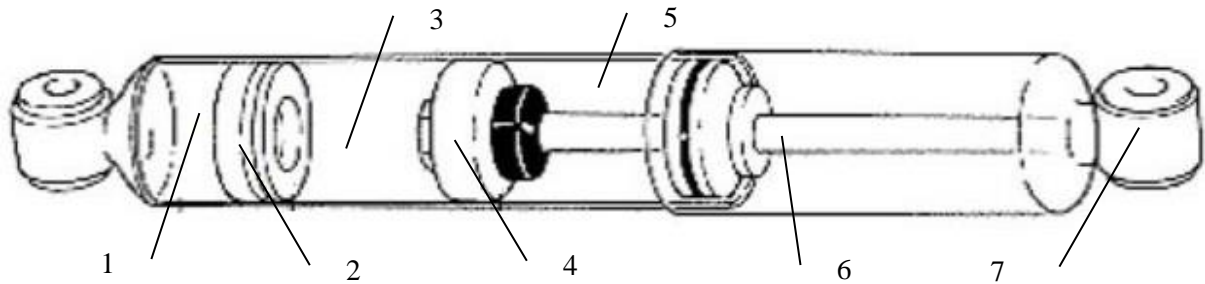


Fig. 1. **Design of gas-charged mono-tube shock absorber:** 1 – gas chamber; 2 – sealing piston; 3 – compression chamber; 4 – working piston; 5 – extension chamber; 6 – piston rod; 7 – rubber mount [14]

A hydrodynamic bench shown in Figure 2 is used to study the behaviour of the damper in the frequency range 20-100 Hz. The hydrodynamic bench can change the amplitude and frequency of the disturbance within 1-100 Hz range. Considering that the hydrodynamic bench has limited power, the increase in frequency can lead to a decrease in the amplitude of displacements. The elements in Figures 2 and 3 have the same numbering.

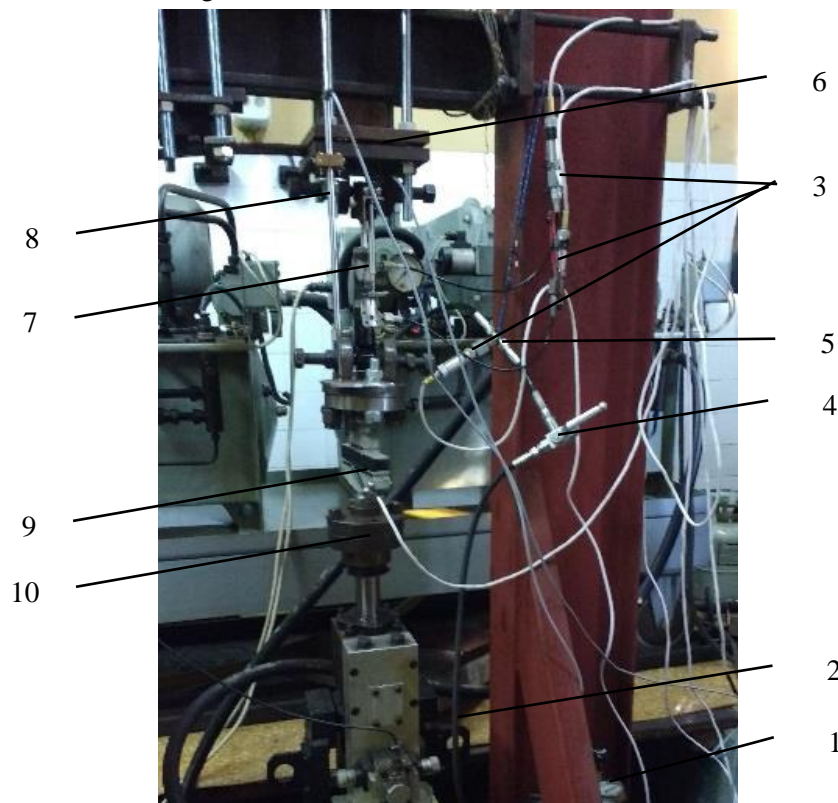


Fig. 2. **Hydrodynamic shaker**

The lower support of the shock absorber is attached to the vibrating head 10 of the hydrodynamic shaker (Figure 3) and the upper one is fixed to the beam 6. On the vibrating head, a strain gauge 9 is attached, measuring the active force in the shock absorber. Pressure transducers 3 measure the pressures in the shock absorber chambers – in the rebound chamber P_{ec} , in the compression chamber P_{cc} and in the gas chamber P_{gc} . The inductive displacement sensor 7 is measuring the stroke of the piston Z_a – relative to the shock absorber body. The inductive displacement sensor 8 is measuring the full stroke of the shock absorber Z_b between its mounts. The system is completed by a gas bottle 1, a shut-off valve 4, a tee 5 and hoses of different lengths. In this way it is possible to supply different pressure to the gas chamber of the shock absorber.

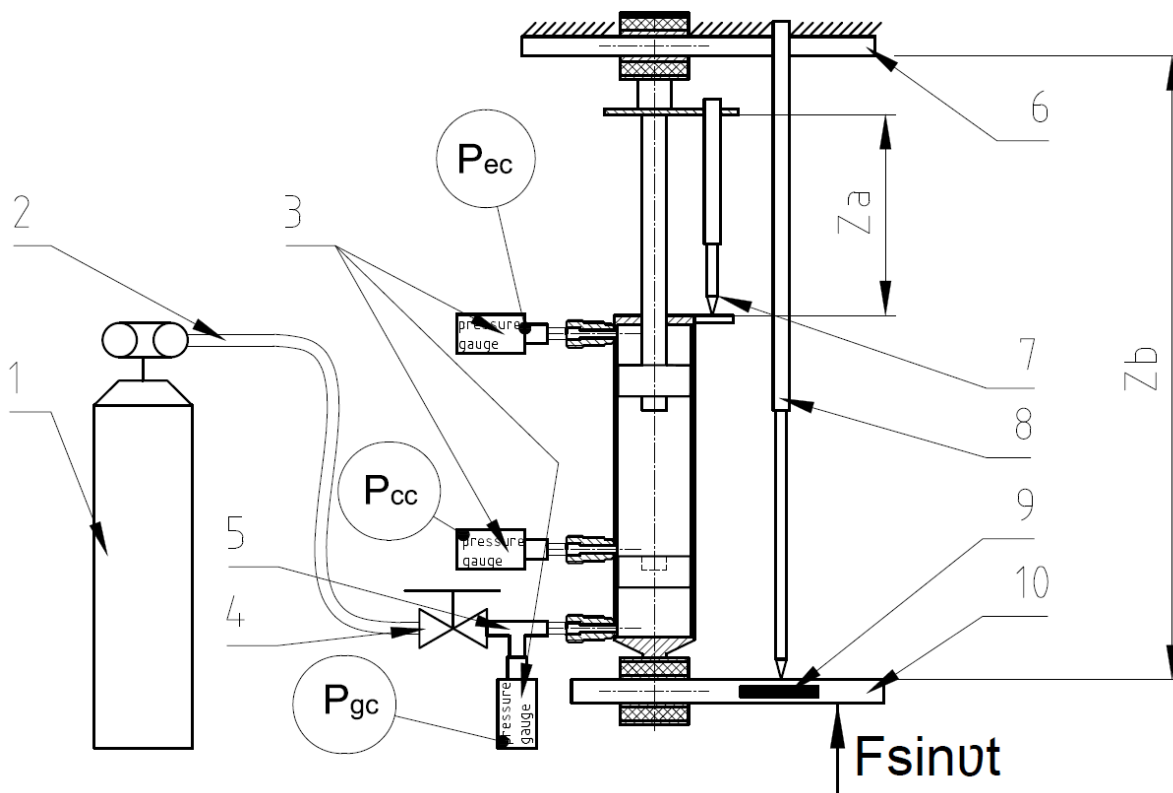


Fig. 3. **Layout of the testing setting:** 1 – gas bottle; 2 – pipeline; 3 – pressure transducer; 4 – stopcock; 5 – three-ways connector; 6 – fixed support; 7, 8 – displacement transducer; 9 – strain transducer; 10 – shaker plate

The sequence of measurements is as follows:

- The hydrodynamic bench acts on the shock absorber with a disturbance of different frequency and amplitude – the experiment plan includes tests at frequencies of 20 Hz, 50 Hz, 70 Hz and 100 Hz.
- For each frequency, the pressures in the three damper chambers P_{ec} , P_{cc} , P_{gc} , displacements Z_a , Z_b , as well as the force in the damper are measured.

Using the measured data, the hydraulic force on the piston due to the pressure drop between the compression and rebound chambers is obtained [14]:

$$F_{pc} = P_{cc} A_p - P_{ec} A_{pa}, \tag{8}$$

$$F_{pe} = P_{ec} A_{pa} - P_{cc} A_p, \tag{9}$$

where F_{pc} , F_{pe} – hydraulic forces on the piston due the pressure drop, respectively in compression and extension of the shock absorber, N;
 P_{cc} , P_{ec} – pressures in the compression and extension chambers respectively, Pa;
 A_p , A_{pa} – piston and piston annulus areas, m²;

A comparison is made between the displacements Z_a and Z_b , which gives an estimation of the damping in the shock absorber and in its rubber mounts. Piston displacement Z_a is differentiated once to obtain the velocity and a second time to obtain the acceleration. Velocity and acceleration give additional information about the influence of the individual components in the total force in the damper (equation (1)). For data processing the maximum values of each measured parameter are used.

Results and discussion

Figure 4 shows an example record of the total force measured in the damper at a set disturbance of 50 Hz. This is the total force of all the components presented in equation (1), acting on the shock absorber itself and its rubber mounts.

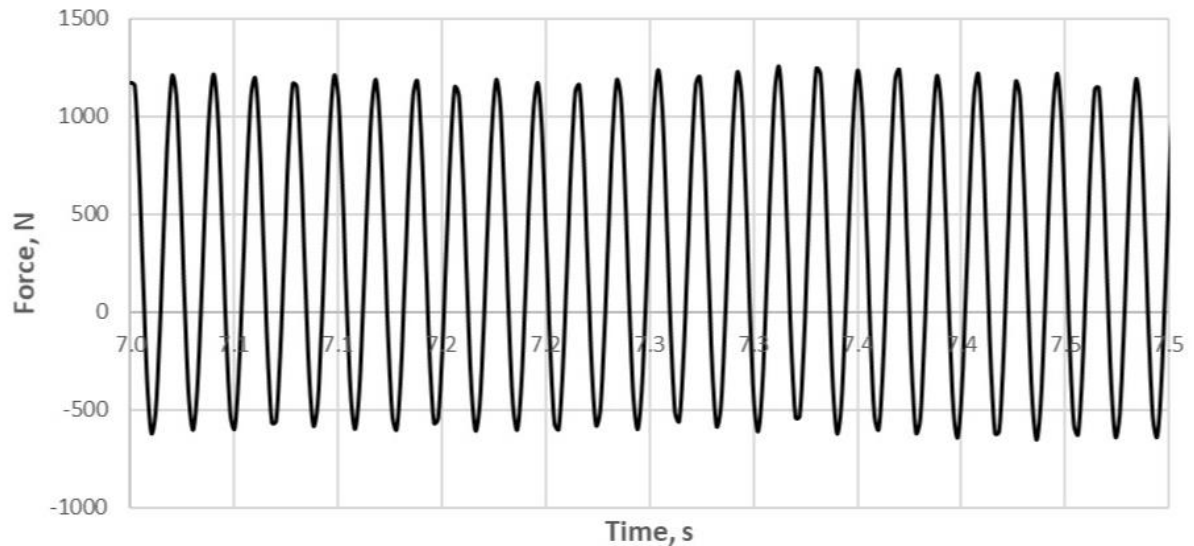


Fig. 4. Time-domain record of the total force in the shock absorber at disturbance of 50 Hz

In Figure 5, both the total forces in the damper during compression and extension, as well as the viscous damping forces due to the pressure drop are plotted as a function of frequency. The maximum values of the specific forces are plotted on the graph. It is observed that the viscous damping forces (both for compression and extension) account for the majority of the total force up to a frequency of 50 Hz. While the hydraulic damping forces continue to decrease with increasing frequency, the total forces measured in the damper increase. At a 20 Hz disturbance, the total compression force is approximately three times less than the extension force. At frequencies above 70 Hz, they have relatively close values. At this point we can assume that the valves of the piston are shut, and the total force is a sum of the elastic force in the rubber mount and the force due to the acceleration of the piston and the residual acceleration coefficient. The total forces are plotted with solid lines and the forces of viscous damping with dashed ones. The values of the total forces are obtained by the strain transducer (number 9 in Figures 2 and 3) and the values of the viscous damping forces are calculated by equations (8) and (9) through the measured pressure drop from the pressure transducers (number 3 in Figures 2 and 3).

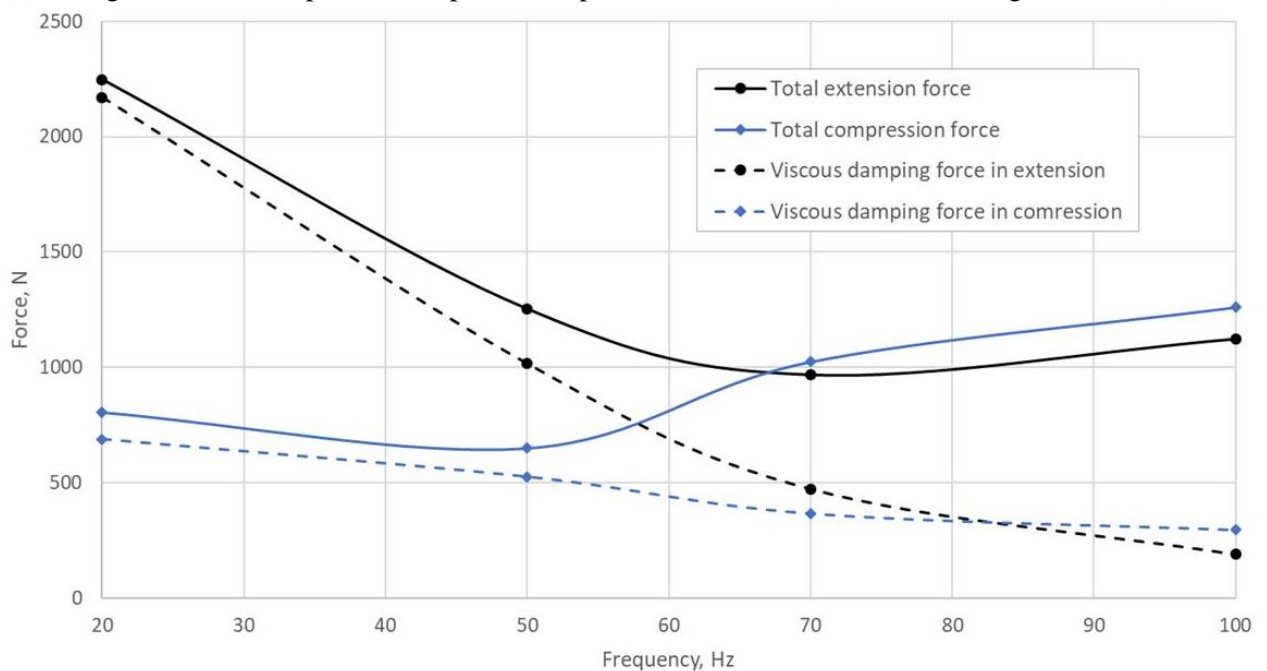


Fig. 5. Frequency-domain of the forces in the shock absorber

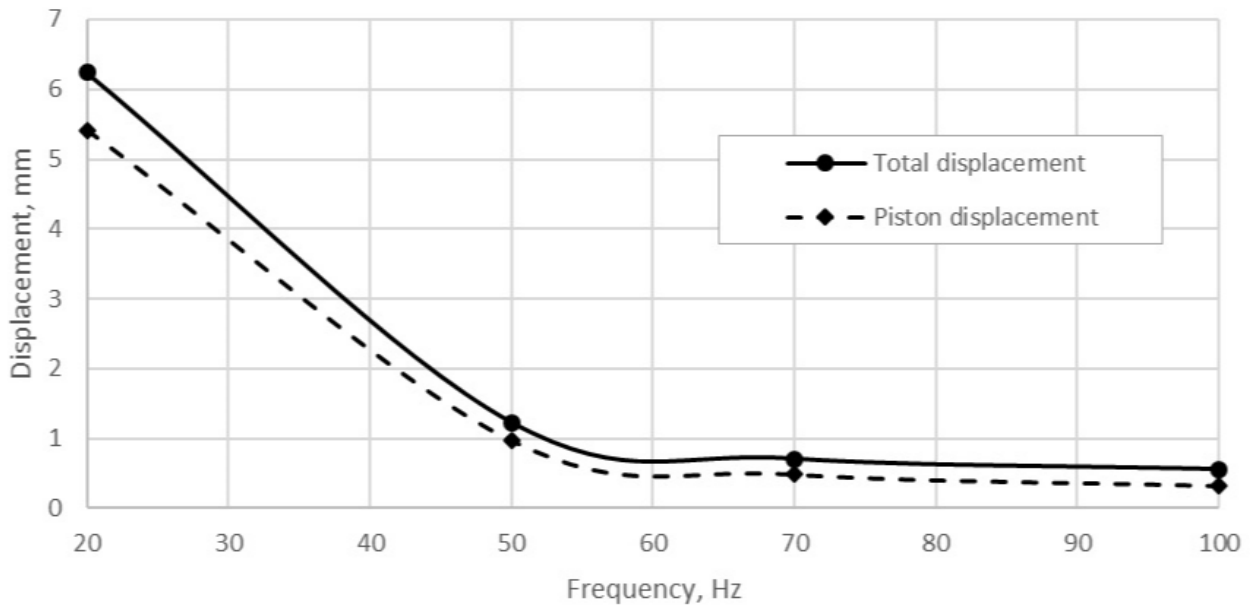


Fig. 6. Displacements of the piston between the two mounting points of the shock absorber as a function of the disturbing frequency

Figure 6 shows the total displacement of the shock absorber between its mounts and the piston displacement only. The difference between them represents the deformation of the rubber mounts. Above 50 Hz the deformation of the rubber mounts is relatively constant. For this frequency range the stiffness of the rubber mounts has no significant changes [15], which means that the elastic force in the shock absorber should be a constant too. Due to the limited power of the hydrodynamic bench, the increase in frequency leads to a decrease in the amplitude of displacements, but for the measurements at 70 and 100 Hz, the amplitude is relatively constant.

As the disturbing frequency increases, the influence of the compressibility of the working fluid and that of the resistance of the valve system in the piston on the force in the shock absorber increases also. Figure 7 shows the acceleration of the piston, which increases for frequencies above 50 Hz. It is observed that for frequencies above 70 Hz, the nature of the total forces in the damper is similar to that of the acceleration. This means that the force due to the acceleration and the coefficient of residual acceleration is the main component in the total force.

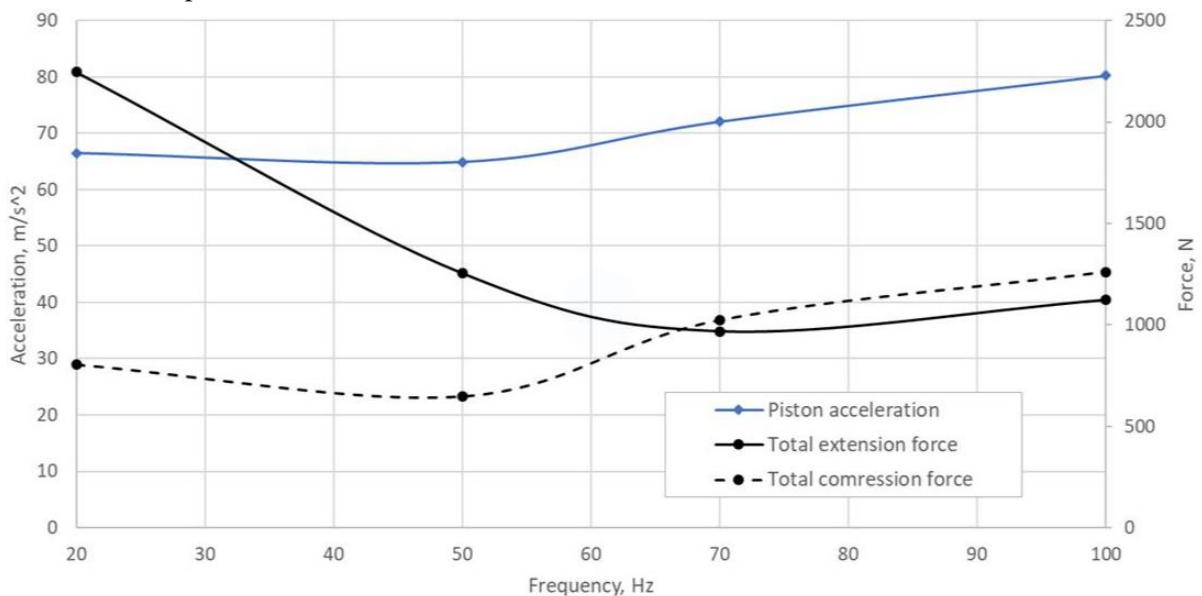


Fig. 7. Acceleration of the piston and the total forces in the shock absorber depending on the disturbing frequency

Conclusions

Based on the conducted experiments, it is established that:

1. In the frequency range above 50 Hz, the valve system in the piston of the shock absorber is not the main damping element. The valves are shut and there is no viscous damping.
2. The rubber mounts remain active, and they are the only elements of the shock absorber damping high-frequency oscillations.
3. In the frequency range above 70 Hz, the total force in the damper is a result mainly of the vibration acceleration of the piston and the residual acceleration coefficient.

Acknowledgements

This research is supported by the Bulgarian Ministry of Education and Science under the National Program “Young Scientist and Postdoctoral Students-2”.

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