doi:10.5937/jaes12-5445 Paper number: 12(2014)3, 295, 217 -226

A CONTRIBUTION TO PASSIVE SHOCK ABSORBER FORCE INVESTIGATION

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Shock absorbers are fundamental part of the vehicle suspension. Suspensions are needed to guarantee vehicle handling and passenger riding comfort. For good handling and braking performance of the vehicle, the tire-road contact forces need to be as stable as possible. Each wheel should always remain in contact with the ground. Comfort means that vibrations, induced by road profile during riding, are of a minimal nuisance to the passengers. When designing a new vehicle, a lot of development effort is focused on the optimal choice of the suspension parameters, stiffness and damping. This paper presents some results of an experimental study conducted on three shock absorbers types that are in current production at Magneti-Marelli. Experimental tests were performed in conditions of random excitation. The aim of this study, based on the experimental data and by use of statistical methods, was to investigate which of the kinematic parameters of excitation has the major impact on damping force of shock absorber (displacement, velocity or acceleration).

Key words: Vehicle, Passive shock absorber, Ride comfort

INTRODUCTION

Automotive shock absorbers are part of the vehicle suspension. Suspension is needed to guarantee vehicle handling and passenger comfort [05, 12-17, 19, 20]. For a good handling and braking performance, the tire-road contact forces need to be as stable as possible. Each wheel should always remain in contact with the ground [12-17, 19, 20]. Comfort means that vibrations, induced by road profiles during riding, are of a minimal nuisance to the passengers.

When designing a new vehicle, a lot of development effort is focused on the optimal choice of suspension parameters, stiffness and damping. A first tuning can be achieved by implementing a full car model and simulating typical road profiles [05, 12-17, 19, 20]. The response from the simulations can give an idea about the quality of the suspension. However, the significance of the results strongly depends on the accuracy of the model.

Shock absorber is one of the most complex parts of the vehicle suspension to model [05,12-17,19,20]. In general, shock absorber behaves in a non-linear and time-variant way. Dampers are typically characterized by a simpler force-velocity diagram, also referred to as the damper characteristic diagram. Some information can also be extracted by plotting forces as a function of displacements resulting in a diagram, that in the automotive industry world is known as work diagram or resistance curve or control diagram [01, 06-08, 11, 18, 21].

The dependency of the shock absorber characteristics on time is due to the progressive rise of the oil temperature during the vehicle operation, which, in turn, is due to the conversion (dissipation) of kinetic energy into heat by viscous losses. Oil viscosity is a determining factor of the shock absorber characteristics and is strongly influenced by the temperature.

This paper presents results of experimental study of three shock absorbers that are in current production at Magneti-Marelli [22]. The experimental tests were performed in conditions of random excitation. The aim of this study, based on the experimental data and by use of statistical methods, was to investigate which of the kinematic parameters of excitation has

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the major impact on damping force of shock absorber (displacement, velocity or acceleration).

EXPERIMENT

Measurements were conducted in Magneti Marelli laboratory with a MTS testing machine, which is able to provide random excitation signals. During the tests the force and displacement values are acquired from the MTS machine transducers and stored inside a PC; moreover also the acceleration of the rod and of the absorber body are measured with two Bruel&Kjaer acceleration sensors. Relative acceleration is then calculated by difference [22].

Relative velocity can be calculated in two ways. Both of these ways are based on the substitution of the differential with the finite difference ratio. The first way consists in the differentiation of the relative displacement with respect to time; the second way consists in the integration of the measured relative acceleration with respect of time.

In both cases it is clear that an estimation of the true value of the velocity is obtained. The random excitation is characterised by a spectrum that has the highest frequency component appropriately lower but of the same order of magnitude of the sampling rate. The integration procedure and the derivation one lead to values of the velocity that are close each to the other but not identical.

Figure 1 shows a diagram to illustrate the relation between the velocity calculated by derivation of displacements and the velocity calculated by integration of the measured relative accelerations. From Figure 1 it is clear that there is a good correlation between two procedures, but none leads to the correct evaluation of the velocity.

Relative acceleration can be calculated both from two accelerations registered during the experimental tests (piston and body) and by means of velocity derivation in respect to time. The relation between them is shown in Figure 2.

From Figure 2 it is clear that there is a good correlation between the two procedures. After some analyses and discussion, bearing in mind that, due to the previously discussed limitations, none of the procedures can lead to the true values of the kinematic quantities but it is highly desirable to unify the methods for calculation of the velocities and accelerations, we decided to calculate relative acceleration by velocity derivation in respect to time.

Three types of shock absorbers of Magneti Marelli current production were tested in this experiment (we will refer to them as I, II and III); they were without rubber joints.

For the shock absorber excitation, random signals with 5 and 10 mm RMS amplitude were used. The excitation frequencies were mainly in the interval from 0.1 to 20 Hz.

During experiment we used sampling time step of 0.00195 s, sample size results to be of 30722 points, that ensures the suitability of the acquired results in the interval 1.6929 Exp(-4) to 250 Hz (the Nyquist frequency), that is acceptable when we have in mind the aim of this study [02-04].

For illustration, Figures 3 and 4 show respectively the history of displacement and force in time, for shock absorber "A" and excitation amplitude of 5 mm RMS.

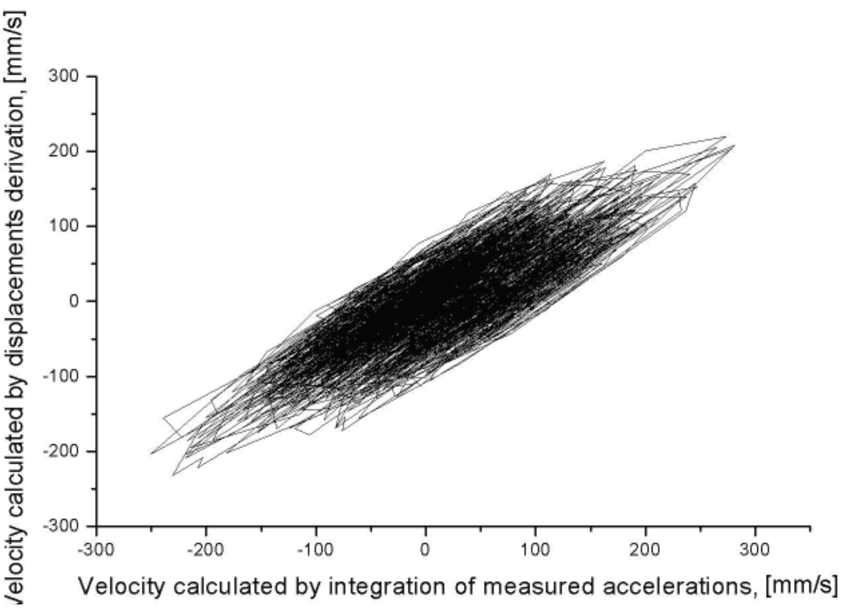

Figure 1: The relation between two procedures of velocity calculation

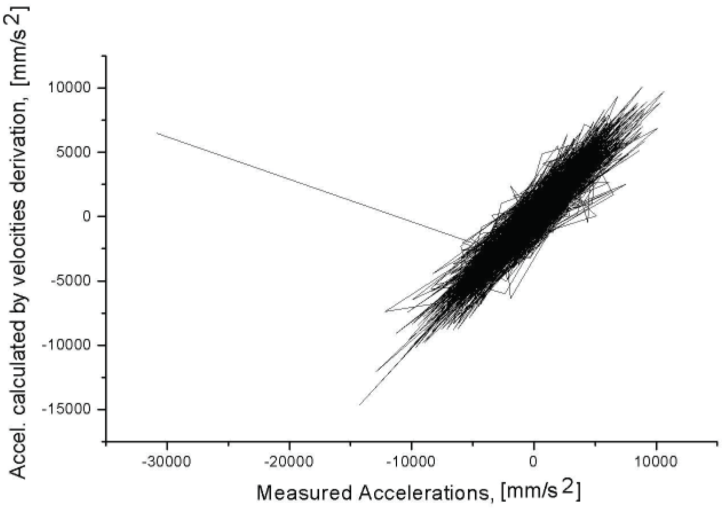

Figure 2: The relation between two calculation procedures for relative acceleration

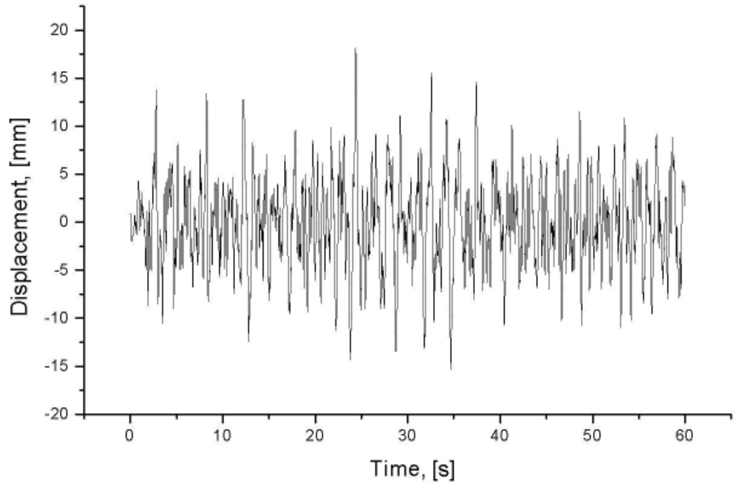

Figure 3: Displacement time history

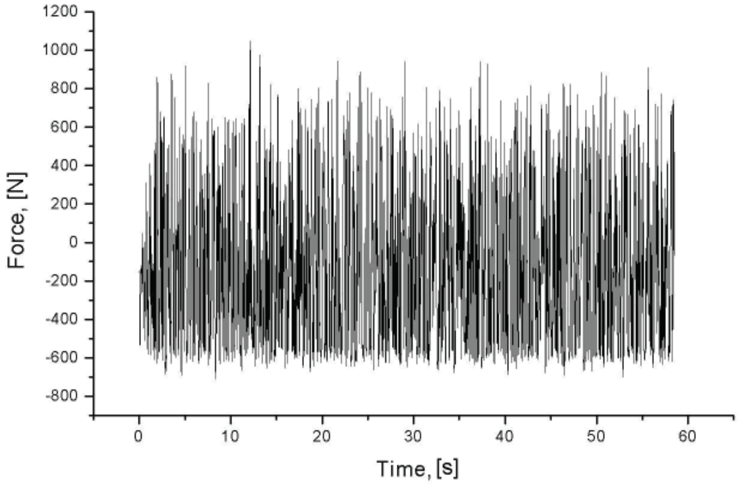

Figure 4: Force time history

The frequency content of the displacement history (see Figure 3) is given in Figure 5 for example. We can see that the excitations are mainly in the interval 0.1 to 20 Hz with random amplitudes.

For illustration in Figures from 6 to 8 the forces as functions of displacement, velocity and acceleration, for shock absorber "I" and excitation with RMS 5 mm amplitude, are given.

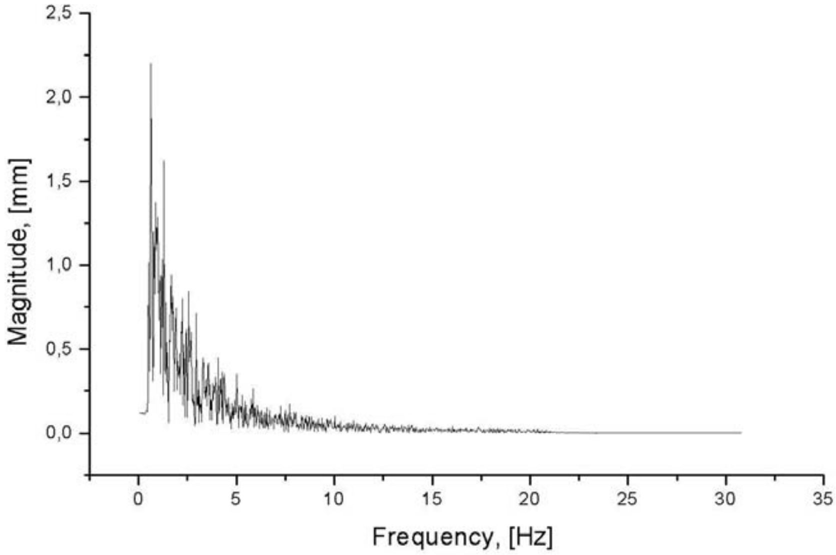

Figure 5: The spectrum of displacements for shock absorber "I" and 5 mm RMS magnitude of excitations

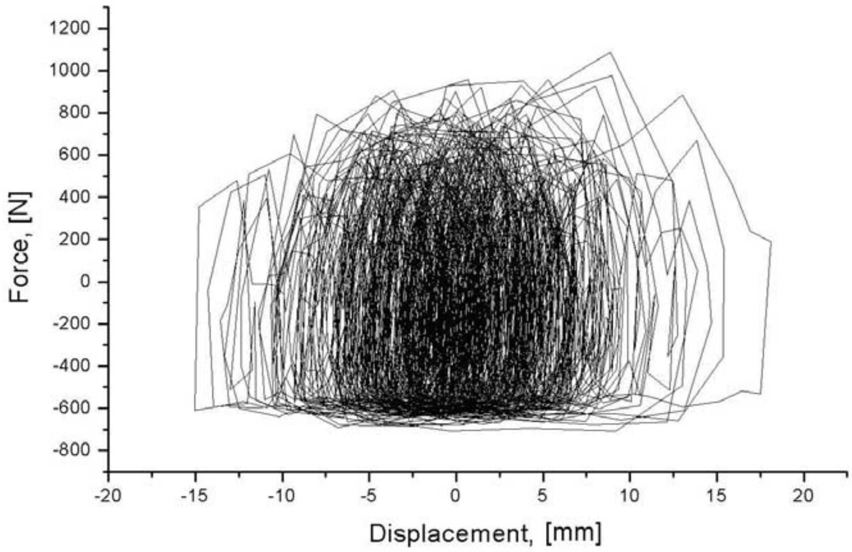

Figure 6: Force as function of displacements

DATA PROCESSING

As already mentioned, the experiment was conducted with a random excitation force. Since it is equal to the force of the damper rod, it means that it depends on the kinematic parameters: relative displacement, relative velocity and relative acceleration [12-17, 19, 20]. Formally, this means that the power can be represented by three components of the excitation force which is dependent on the relative displacement, relative velocity and relative acceleration. As it is known, among the above mentioned kinematic variables there are mathematical relations and the assumed inputs are not mutually independent, but are coupled. It has been found appropriate to investigate the effects of input components on the excitation forces using the method of "black box", where the input values are the components of the input force and the output value is the force on the shock absorber's piston rod.

As known from [02–04, 09], transfer function of dynamic systems with multiple inputs and one output is calculated by dividing the Laplace transform of the output value with the corresponding Laplace transform of the input value in the complex plane. This, of course, applies provided that there is no coupling between the inputs. If this condition is not fulfilled, the above theory can be applied, provided the inputs are decoupled [02-04, 10]. There are several ways of decoupling coupled inputs, but here will be presented the theory from [02–04, 09].

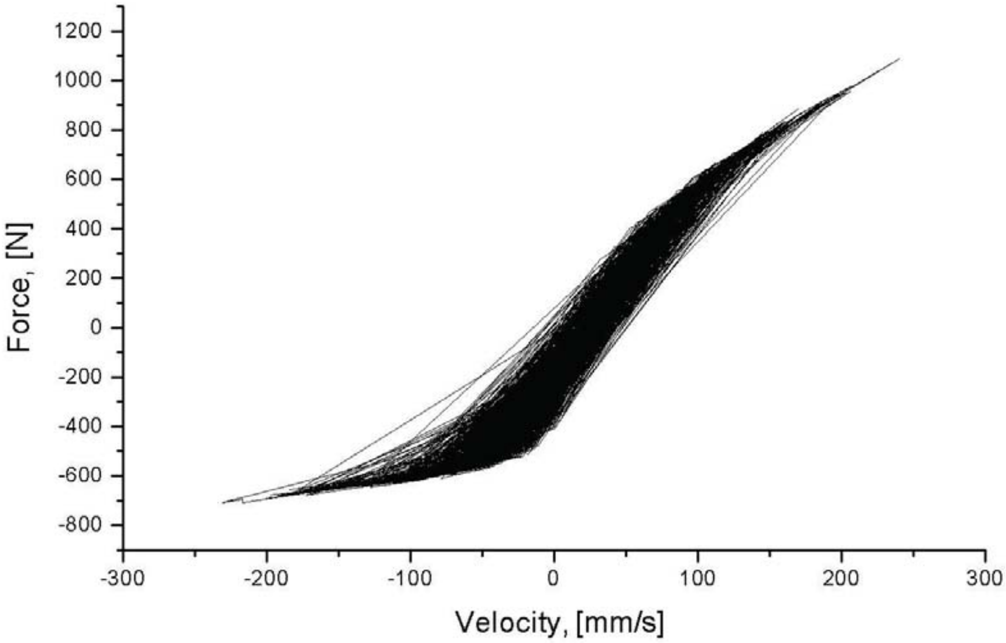

Figure 7: Force as function of velocity

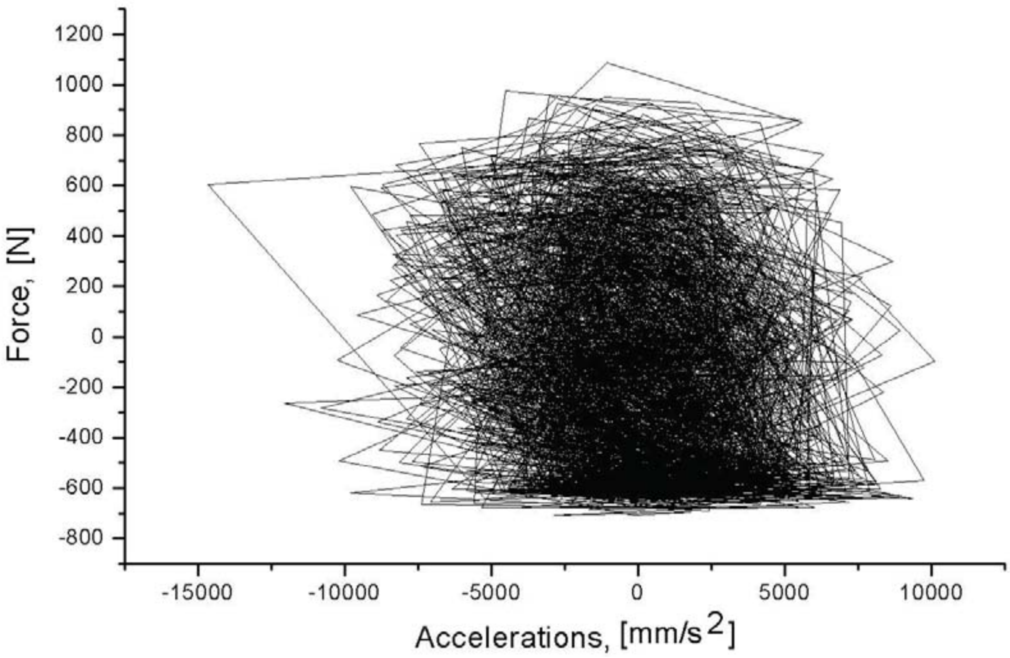

Figure 8: Force as function of acceleration

In our case there are three coupled inputs (relative displacement X1, relative velocity X2 and relative acceleration X3), and their decoupling scheme is displayed in Figure 9, where all three input parameters are mutually coupled. It should be noted that in this case the existence of any noise between the input values and the output value of force is neglected. Based on [02–04, 09] the following relation can be writen:

$$X_1 = X_1$$

$$X_{2.1} = X_2 - L_{12}X_1$$

$$X_{3.2!} = X_3 - L_{13}X_1 - L_{23}L_{2.1}$$

where:

- X_1, X_2 and X_3 - coupled (mutually dependent) excitations (in the complex plane),
- X_1, $X_{2.1}$ and $X_{3.2!}$ - decoupled (mutually independent) excitations (in the complex plane),
- L_{12}, L_{13} and L_{23} – appropriate transfer functions among coupled input signals in the complex plane.

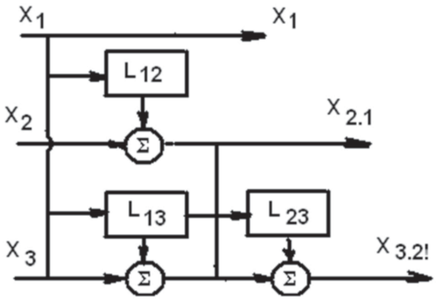

Figure 9: Decoupling of the three input and one output system

The described procedure is used for decoupling the input signals of the relative displacement, relative velocity and relative acceleration. For this purpose, the program [09] is used, based on the theory from [02-04]. The program allowed calculation of partial coherence functions and partial transfer characteristics of the system.

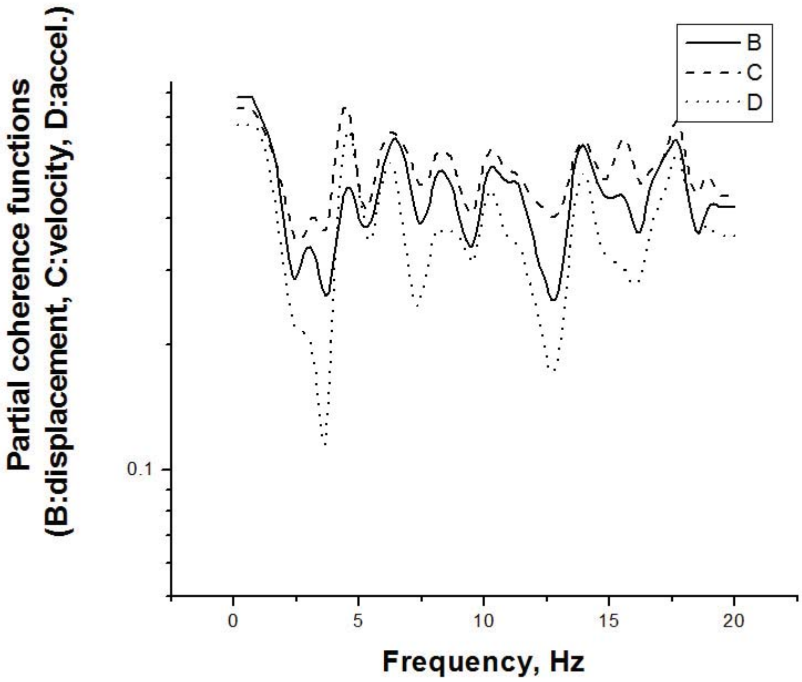

Figure 10: Functions of the partial coherences, shock absorber I, excitation rms 5, mm

ANALYSIS OF THE RESULTS

Partial coherence functions

It is well known from [02-04] that partial coherence functions provide statistical information on the impact of certain inputs (in this case relative displacement, relative velocity and relative acceleration) on the output value (in this case the force on the piston rod of a shock absorber). The application of software [09] enabled a calculation of functions of partial coherences, which, for the purpose of illustration, are partially shown in Figures 10-12. Note that these functions do not provide quantitative information about the impact of some of the observed inputs on the force in the shock absorber, because their values are relative and their maximum is equal to one. In practice, as a lower limit of the existence of couplings between the input and output values, the value of partial coherence function of about 0.30 is required [02-04].

The analysis of all data, partially given in Figures 10-12, shows that the design of shock absorbers, RMS of the applied excitation and frequency, have the impact on the observed functions of partial coherence. It is evident that relative velocity, relative acceleration and relative displacement have the influence on the analyzed partial coherence functions, because the average values of functions of partial coherence is about 0.5.

This points out a need to perform additional analysis of partial transfer characteristics, as will hereinafter be discussed.

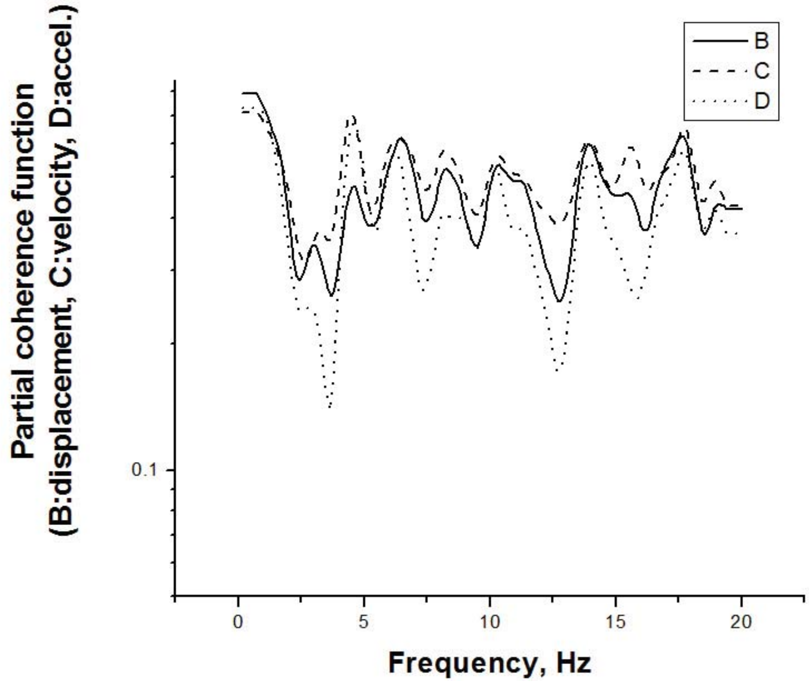

Figure 11: Functions of the partial coherences, shock absorber II, excitation rms 5, mm

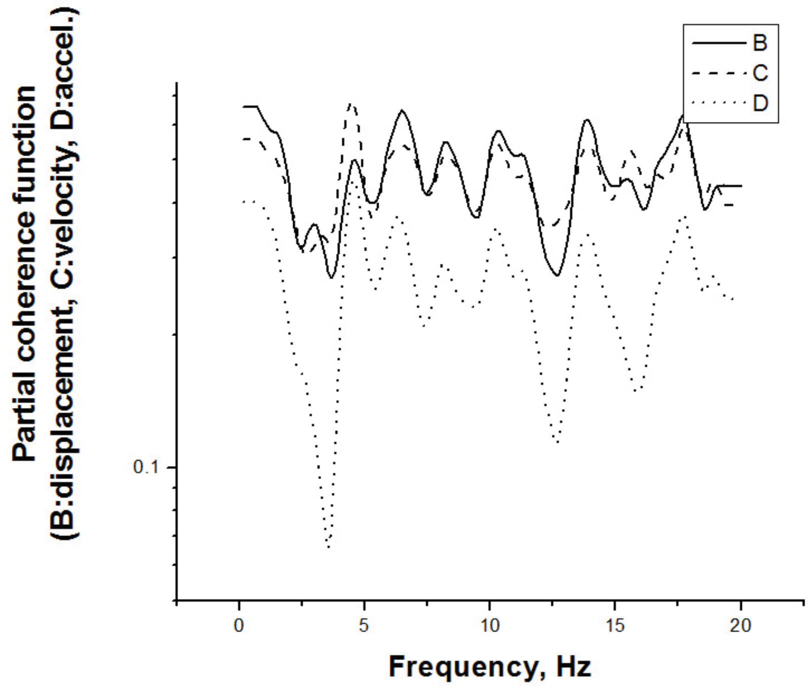

Figure 12: Functions of the partial coherences, shock absorber III, excitation rms 10, mm

Partial transfer characteristics

Applying the developed software [9], partial functions of transfer characteristics have been calculated, which are partially shown in Figures 13-15, for the sake of illustration.

The analysis of all the data, partially shown in Figures 13-15, leads to a conclusion that the design of passive shock absorbers, RMS of the excitation and frequency have the influence on the transfer characteristics of passive shock absorbers.

Limit values of partial transfer characteristics of the tested shock absorbers and RMS of the excitation of 5 and 10 mm, are shown in Table 1.

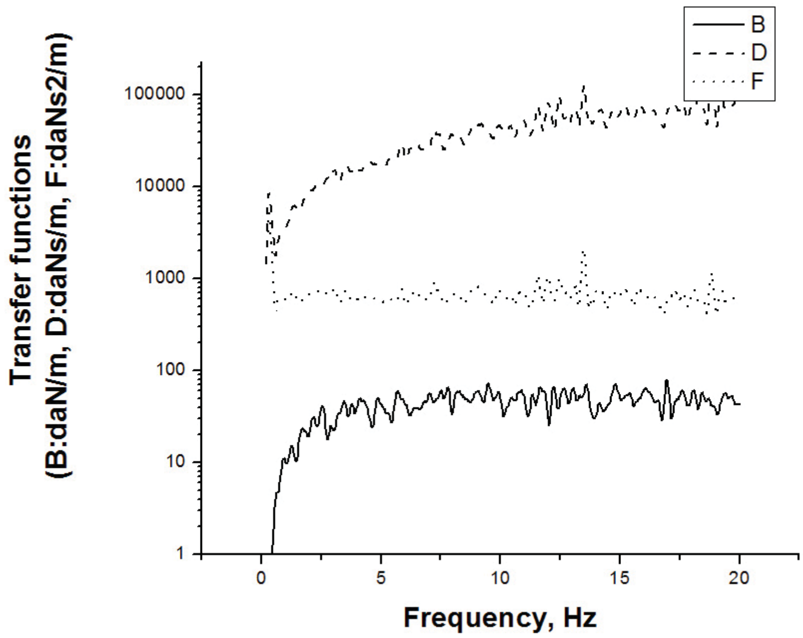

Figure 13: Transfer functions for the shock absorber I, excitation rms 5, mm (B: displacement, D: velocity, F: acceleration)

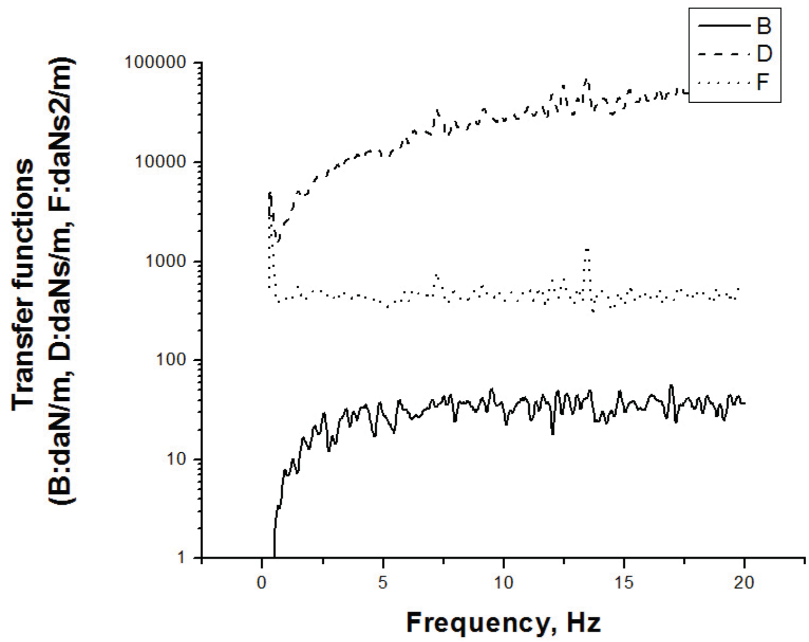

Figure 14. Transfer functions for the shock absorber II, excitation rms 5, mm (B: displacement, D: velocity, F: acceleration)

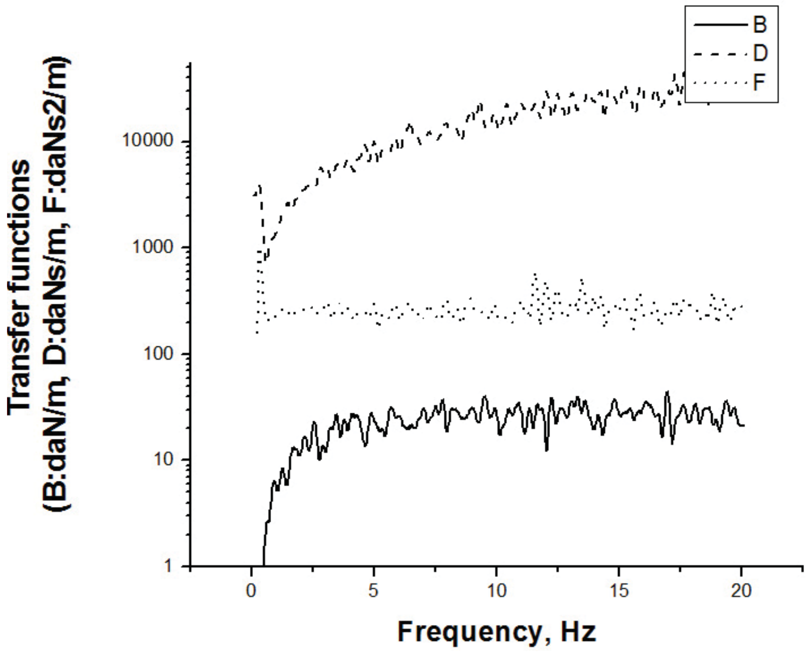

Figure 15. Transfer functions for the shock absorber III, excitation rms 10, mm (B: displacement, D: velocity, F: acceleration)

Table 1: Limit values of transfer characteristics of shock absorber

Design of shock absorber	Parameter	Minimum value	Maximum value
I	Relative velocity Relative acceleration Relative displacement	1000 500 1	200000 2000 100
II	Relative velocity Relative acceleration Relative displacement	10000 300 1	100000 1000 80
III	Relative velocity Relative acceleration Relative displacement	1000 200 1	30000 500 70

Note: Dimensions of the shown values correspond to ones given in Figures 7-12.

Analysis of the data presented in Table 1 shows that the greatest impact on the force in a passive shock absorber has relative velocity, than acceleration and the lowest has relative displacement. This fact, however, should be taken into consideration when modeling passive shock absorber.

CONCLUSION

This paper presents the analysis aimed to determine which of kinematic parameters has the major influence on the force in a passive shock absorber. The obtained results clearly indicate that the force in a shock absorber depends on relative displacement, relative velocity and relative acceleration. In addition, relative velocity has the greatest, and relative displacement of the piston with the reference to the body of the shock absorber has the least impact, what should be taken into consideration during their modeling.

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Paper sent to revision: 04.02.2014.

Paper ready for publication: 08.09.2014.