The Opinion of Impact of Energy Dissipation in a Vehicle Suspension on Mechanical Energy Transfer

Jan WARCZEK
Silesian University of Technology, Faculty of Transport,
Krasińskiego street 8, 40-019 Katowice, jan.warczek@polsl.pl

Abstract

Effective shock absorption of the body in the presence of excitations of a random nature results in the necessity of using vibration dampers. The classic solution is the viscous damper, which works by dissipating mechanical energy in the form of heat generated to the external environment. The characteristics of the flexible components in car suspensions (elastic and damping) affect the transfer of mechanical energy. The purpose of the research was to develop a method for assessing the impact of damping in a suspension system on the transfer of mechanical energy from unsprung masses to the body. The assumed research goal was carried out on a vehicle model with two degrees of freedom, whose parameters corresponded to the real object. The developed method uses energy relationships occurring between selected variables of the vibrating motion of inertial elements of a motor vehicle. The results obtained during simulation tests were empirically verified on a real object. The proposed method allows to assess how the damping in the suspension system affects the energy relationships occurring in the vehicle suspension vibrating motion.

Keywords: energy of mechanical vibrations, vehicle suspension, damping

1. Introduction

Vibration forces acting on the vehicle lead to vibrations in all its structural elements. To ensure comfort and safety of travel, the most measurable are the final effects of energy conversion of vibration excitations on the body (comfort) and on the wheels (safety). The negative impact of vibrations on persons and loads transported by road means of necessity of their minimization. To this end, solutions leading to vibration damping are commonly used in the construction solutions of vehicle suspension. The issue of vibration damping in the most general case boils down to the use of flexible (elastic) elements that act as time buffers in the transfer of mechanical energy [2, 3, 5, 8, 13, 14]. The use of a damping element with high compliance in the suspension leads to a favourable reduction of resonance vibration amplitudes at the expense of worsening of the depreciation in non-resonant areas. The characteristics of the flexible components in car suspensions (elastic and damping) affect the transfer of mechanical energy.

Depending on the adopted scale (size) of the object, vibration analysis can be carried out in relation to selected elements or entire objects. In special cases, vibration analysis can be carried out for individual material points. In motor vehicles, depending on the purpose adopted, the purpose of testing vibration properties is carried out in terms of full vehicle load or within the scope of selected subsystems or structural elements. The vibration properties of construction elements are also optimized, e.g. by using the finite or edge element method [1, 10, 11, 16, 18, 23].

2. The aim of research

Research on real objects is time consuming and very expensive. An alternative, which is simulation research, allows obtaining knowledge about physical phenomena using a computer and appropriate software. In the analysis of energy transformations occurring in a vibrating object, a very popular mathematical modelling tool is to describe such a system using second-order Lagrange equations. In modelling the work of a vehicle suspension, the system often used is to describe this system in the form of a two-mass model with two degrees of freedom [4, 6, 7, 9, 12, 15, 17, 19]. The purpose of the research was to develop a method for assessing the impact of damping in a suspension system on the transfer of mechanical energy from unsprung masses to the body. The assumed research goal was carried out on a vehicle model with two degrees of freedom, whose parameters corresponded to the real object. In the developed method, relationships between selected RMS of variables of vibrating motion of inertial elements of a motor vehicle were used. The results of simulation tests were verified by tests on a real object.

3. The simulation research

The physical model of the suspension has the following on figure 1.

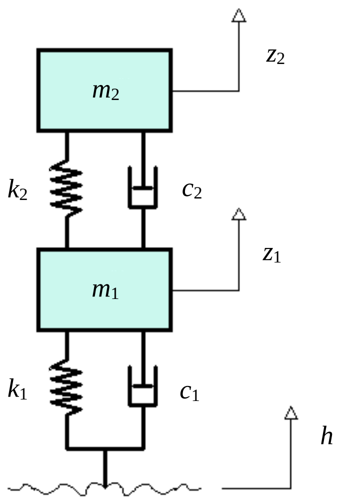

Figure 1. Model of vehicle suspension, m_2 – mass of the sprung part, m_1 – mass of the unsprung part, k_2 – suspension stiffness coefficient, k_1 – wheel stiffness coefficient, c_2 – suspension damping coefficient, c_1 – wheel damping coefficient, h – excitations from road roughness, z_2 – vertical displacements of the sprung mass, z_1 – vertical displacements of the wheel (unsprung masses)

The quarter suspension model can be treated as a good representation of the system dynamics provided that the following assumptions are taken into account:

– the vehicle has a symmetrical structure,
– the vehicle parts (sprung and unsprung parts) are rigid bodies whose masses are concentrated at their centres of gravity,
– mechanical ratios resulting from the design of the suspension linkage system will be taken into account. So the damping characteristics and stiffness of the respective elements are multiplied by these coefficients.

The assumed physical suspension model was introduced in the form of a system of differential equations to the Matlab / Simulink program.

Motor vehicles are operated on roads with different surfaces. The road profile can theoretically take a variety of deterministic and stochastic forms, but in practice only for special road test sections can it be assumed that the surface can be described by strict mathematical functions. In real conditions, road surfaces on which motor vehicles travel are best characterized by the road profile defined by the power spectral density of inequalities [6, 20, 21, 22]. In accordance with applicable standards, the road profile is described by means of unevenness of the road surface measured along the trace of cooperation of the vehicle wheel with the road surface. Measurements of the height of unevenness of the road surface are recorded with a constant distance of the road length. The irregularity profile registered in this way is the basis for calculating the parameters of longitudinal unevenness. Based on the uneven profile, a set of longitudinal road unevenness parameters is calculated, of which the most important is the IRI (International Roughness Index).

In the conducted simulation tests deterministic extortion was used. The use of explicitly defined enforcement allows a convenient qualitative and quantitative interpretation of the obtained results. Knowing the nature of the functioning of the tested system, it is possible to extend the scope of analyzes to include random excitations that correspond to the actual operating conditions of the vehicle suspension system.

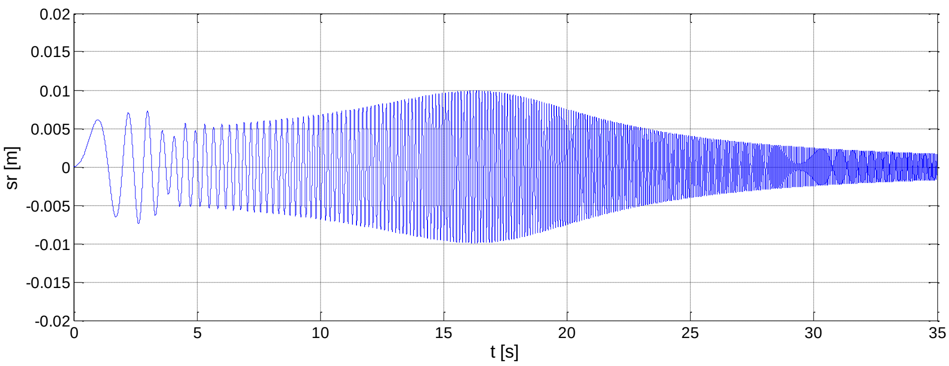

Figure 2. Relative masses displacements (m_2 and m_1) for chirp type excitation (amplitude 6 mm, frequency range 0-20 Hz)

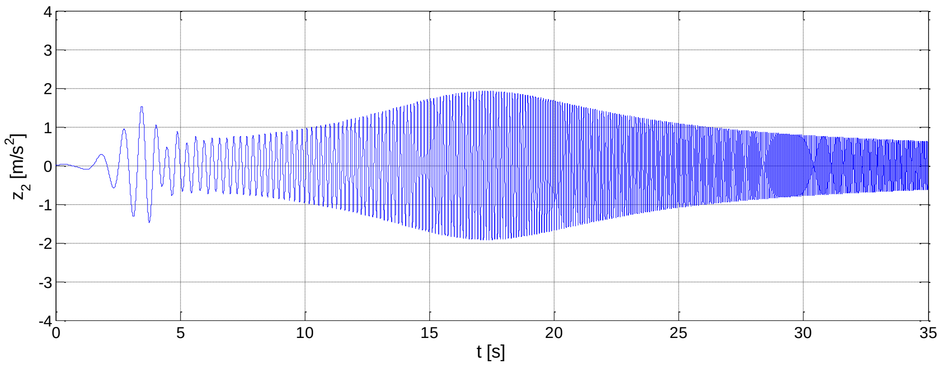

Figure 3. Acceleration of mass m_2 for chirp type excitation
(amplitude 6 mm, frequency range 0-20 Hz)

Figures 2 and 3 show the vibration signals obtained during the simulation tests of body accelerations and suspension deflections under sinusoidal excitation with a linearly increasing frequency (chirp), whose amplitude was 6 mm.

The measures of vibration energy used to analyze the vibration energy transfer were the effective values of selected physical quantities describing the object's vibration. The kinetic energy of the vehicle body mass oscillation is dependent on the square of velocity. The effective values of vibration acceleration are an energy measure that describes the energy of the system. Potential energy in oscillating motion depends on the square of the suspension spring deflection. The effective spring deflection value is related to the potential energy transferred by the spring to the body. These are not simply energy values, but rather measures describing the energy relationships occurring during the motion of a vibrating suspension. For certain conditions of excitation of vibrations, the amount of dissipated energy is constant and the suspension system is in a specific dynamic state. Changing the damping in the system with constant excitation leads to changes in the energy transmitted to the vehicle body. To assess the transmission of vibration energy on the body, the relationship between the effective values was used: acceleration of body vibration and relative wheel displacement relative to the body defined as:

$$sr_{\mathrm{RMS}} = \sqrt{\frac{1}{T}\int_0^T sr(t)^2\,dt} \tag{1}$$

$$\ddot{z}_{2\,\mathrm{RMS}} = \sqrt{\frac{1}{T}\int_0^T \ddot{z}_2(t)^2\,dt} \tag{2}$$

where: $sr(t)$ – suspension deflection,

$\ddot{z}_2(t)$ – acceleration of the sprung mass.

The recorded signals were divided into time fragments with a length of 1 second and effective values were determined for each window in accordance with formulas 1 and 2. Then the determined values were marked on a common plane of energy relationships between the analyzed vibration signals. Figure 4 presents the impact of the damping change on the determined relationships.

a)

b)

c)

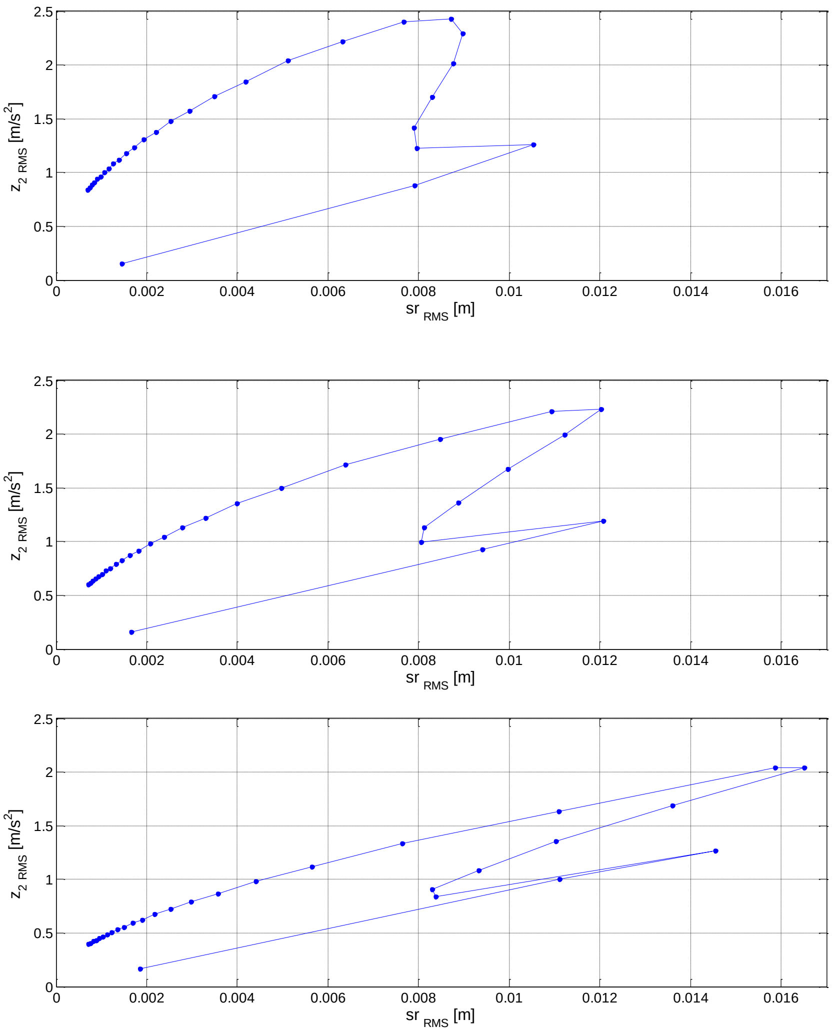

Figure 4. The relations of RMS of body of car accelerations (m_2) and suspensions deflections (*sr*) for excitations with an amplitude of 6 mm in the frequency range from 0 to 30 Hz for damping of: a) 100% nominal, b) 70% nominal, c) 40% nominal

The determined curves show similarity to the attractors used in the analysis of chaotic vibrations. Along with the progressing wear of the shock absorber and thus the damping in the suspension, which decreases in this way, the designated curve gradually turns in the direction of the abscissa with a clear narrowing (Fig. 4c).

4. Research on a real object

Experimental confirmation of the correctness of the proposed method was carried out during the bench tests of the hydropneumatic suspension of a passenger car, the basic element of which is a hydropneumatic column. The damping characteristics of the hydropneumatic column were determined during the tests at the indicator stand. In this way, work charts for different sinusoidal excitation frequencies were obtained. On the basis of the intersection points of the work charts with the abscissa, where the maximum damping force occurs for the highest relative speed of the piston movement, points describing the damping characteristics have been determined (Fig. 5). The choice of the hydropneumatic column as the research object resulted from the fact that the length of the excitation stroke was negligible on the result obtained in the form of damping characteristics.

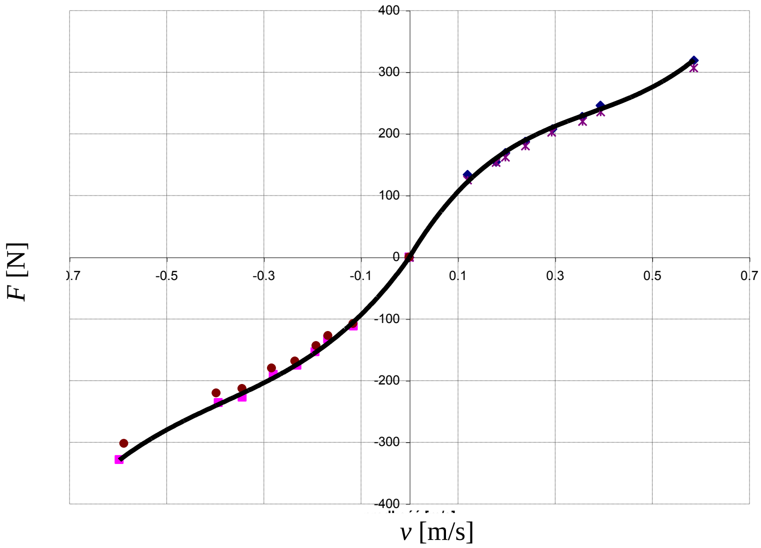

Figure 5. The damping characteristics of hydropneumatic column determined for a sphere of nominal pressure p_0 = 5.5 MPa

The column of identified technical condition was built into the suspension, which was then excited by vibration using a high power exciter analogous to the solutions used in suspension testing stands at PTI (Periodical Technical Inspection) station. The amplitude of the exciter plate vibration displacements was constant and amounted to 6 mm, while the use of a frequency converter in the control system enabled the adjustment of the length of forcing cycles. During the tests, the following signals were recorded: acceleration of body vibrations at the point of upper attachment of the suspension column and relative displacements of the wheel and the body. Examples of real time object acceleration vibration accelerations and suspension deflections recorded during real object tests are shown in Figures 6 and 7.

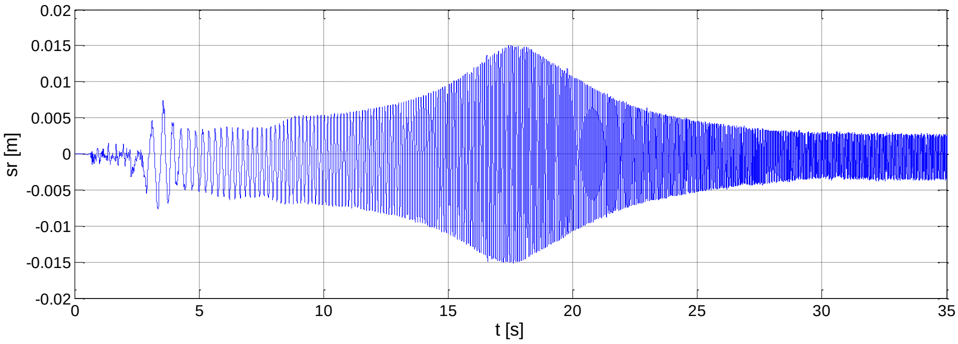

Figure 6. Recorded signal of wheel displacements relative to body of car (suspension deflections) in the vertical direction

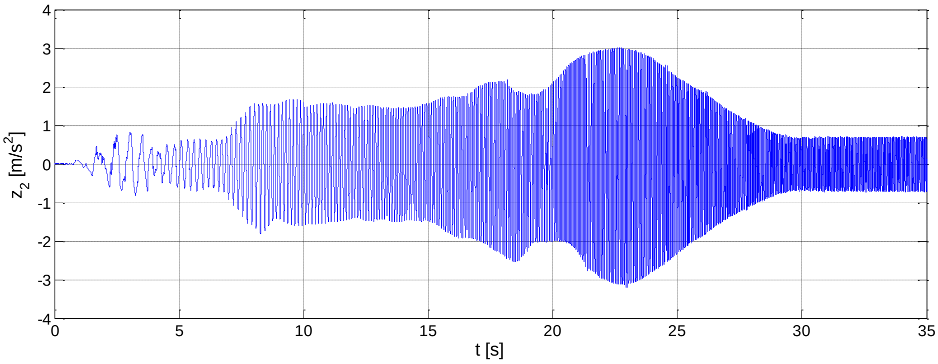

Figure 7. Recorded signal of acceleration of body vibration in the vertical direction

The results obtained were analyzed and the energy relationships between the analyzed signals were determined, as shown in Figure 8.

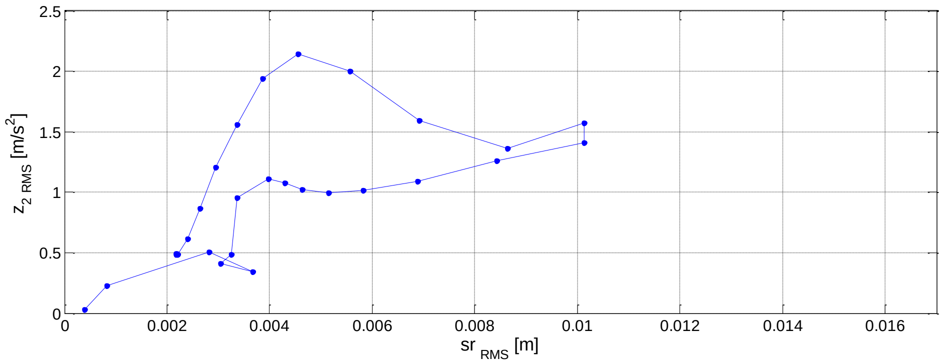

Figure 8. The relations between of accelerations of the body (RMS) and relative displacements of the wheel and the body (RMS) determined for the real object

The analysis was carried out for 1 second time windows that can be treated as signal fragments recorded at a quasi constant frequency of excitation. Using the point markers, the determined energy relations between the studied motion parameters were marked on the graph.

The relationship between the analyzed signals is clearly visible, which results from the operating parameters of the tested suspension system. The obtained results confirm the calculated energy relations between the analyzed signals.

5. Conclusions

The effective value of body vibration acceleration is a measure of vibration energy. The effective values of vibration accelerations are used to assess the criteria for daily vibration exposure, which are regulated by law. The test results show that the same RMS acceleration of body of car vibrations occurs for different frequencies of excitations. The monitored suspension deflection is related to the mechanical energy in spring converted vehicle suspension. Only the combined presentation of body accelerations and suspension deflections in the form of RMS gives the opportunity to assess the impact of energy dissipation in the shock absorber on the vibroactivity of the entire system. The obtained results in the form of occurring between the analyzed parameters of the vibrating motion are presented in the form of graphs containing characteristic curves describing the studied phenomenon. The results obtained during simulation tests were empirically verified on a real object. Depending on the damping properties of the suspension, the vibrations of the sprung mass analyzed on the energy dependence plane of selected parameters of the vibrating motion are specific and allow the assessment of the way of mechanical energy transfer in the system. The proposed method allows to assess how the damping in the suspension system affects the energy relationships occurring in the vehicle suspension vibrating motion.

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