ASSESSMENT OF DAMPING EFFICIENCY FOR RAIL VEHICLE SHOCK ABSORBERS BASED ON ACCELERATION MEASUREMENTS

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Abstract: *In this paper is developed a method for assessment of the damping efficiency of vertical shock absorbers used in secondary suspensions of railway vehicles. The method is based on the analysis of vertical accelerations recorded simultaneously in operating conditions on axle, bogie and car body. Using a Two Degree of Freedom (2DOF) linear system, it is shown that the average phase-difference between the band-pass filtered accelerations of bogy and car body depends only on the damping condition of secondary suspension dampers, being practically independent of that displayed by bogie dampers. An analytical relation is assessed between the values of this parameter and the values of equivalent damping ratio of secondary suspension. The method is illustrated using the experimental results obtained by vibration measurements conducted on a passenger railway car.*
Keywords: *rail vehicle suspensions, vertical dampers condition, phase difference of vibrations*

1. INTRODUCTION

Vibration-based condition monitoring techniques are applied for many engineering structures due to the advancement of measuring devices and of signal processing methods. However, the condition monitoring for components of complex structures, as are the rail vehicle suspensions, the condition assessment for different components is still a difficult task due to the dynamic interactions associated with their operation [1]. In order to detect the faults of specific suspension components, as are the dampers or springs, appropriate models and measuring systems should be developed such as to highlight their effects in system dynamic interactions by vibration measurements in operating conditions. Ideally, the developed method shouldn't assume the perfect knowledge of track condition or repeatability of conducted tests. These features are essential for its applicability and cost. There are two types of condition monitoring techniques [2]:

- knowledge-based (data-driven) techniques, where no analytical system model is employed and only qualitative or empirical system knowledge are used for fault diagnosis [3-6];
- model-based techniques, where the system's available measurements are compared with a priori information represented by the system's mathematical model [7,8].Both model-based and data-driven approaches are studied in [9] for the suspension fault detection problem. The associated signal processing methods used to analyses the outputs of railway vehicles must be chosen such as to provide the most indicative features as fault indicators [2].

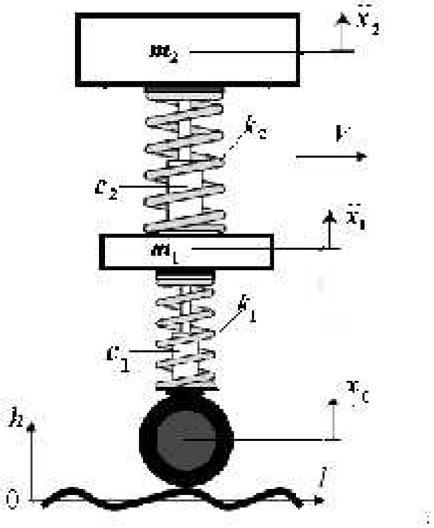
In this paper a simple model of a railway vehicle suspension is used to assess the condition of secondary suspension dampers by determining the average phase- difference between vertical accelerations measured for normal operating conditions on bogie frame and body floor. The values of this coefficient are independent of the condition of primary suspension dampers. By appropriate band-pass filtering of the recorded signals, an analytical relation is obtained between the values of measured cross correlation coefficients and the values of equivalent damping ratio of secondary suspension. This relation can be used to assess the damping capacity of secondary suspension dampers placed on the same vertical axis as the employed accelerometers.

2. ANALYTICAL MODEL

The schematic of mechanical model utilized in this paper to assess the damping capacity of the shock absorbers used in railway vehicle suspensions is shown in Fig.1.

The system equations of motion are

$$\begin{cases} m_1\ddot{x}_1 = -c_1\left(\dot{x}_1-\dot{x}_0\right)-k_1\left(x_1-x_0\right)+c_2\left(\dot{x}_2-\dot{x}_1\right)+k_2\left(x_2-x_1\right) \\ m_2\ddot{x}_2 = -c_2\left(\dot{x}_2-\dot{x}_1\right)-k_2\left(x_2-x_1\right) \end{cases} \tag{1}$$



m_1, m_2 - bogie and body masses distributed on one wheelset;
k_1, c_1 - stiffness and damping coefficients of primary suspension;
k_2, c_2 - stiffness and damping coefficients of secondary primary suspension;
$h(l)$ - profile of rail vertical unevenness;
V - constant vehicle speed ;
$x_0(t) = h(Vt), x_1(t), x_2(t)$ - vertical absolute displacements of wheel, bogie and body;
$\ddot{x}_0(t), \ddot{x}_1(t), \ddot{x}_2(t)$ - absolute accelerations of wheel, bogie and body, measured simultaneously on the same vertical axis.

Figure 1: 2DOF mechanical model of suspension

Using the standard notations

$$\omega_1 = \sqrt{\frac{k_1}{m_1}}, \ \zeta_1 = \frac{c_1}{2\sqrt{k_1 m_1}} = \frac{c_1}{2m_1\omega_1}, \ \omega_2 = \sqrt{\frac{k_2}{m_2}}, \ \zeta_2 = \frac{c_2}{2\sqrt{k_2 m_2}} = \frac{c_2}{2m_2\omega_2}$$
$$\mu = \frac{m_2}{m_1}, \ \frac{k_2}{m_1} = \mu\omega_2^2, \ \frac{c_2}{m_1} = 2\mu\zeta_2\omega_2 \qquad (2)$$

The equations of motion (1) can be rewritten as:

$$\begin{cases} \ddot{x}_1 = -2\zeta_1\omega_1(\dot{x}_1 - \dot{x}_0) - \omega_1^2(x_1 - x_0) + 2\mu\zeta_2\omega_2(\dot{x}_2 - \dot{x}_1) - \mu\omega_2^2(x_2 - x_1) \\ \ddot{x}_2 = -2\zeta_2\omega_2(\dot{x}_2 - \dot{x}_1) - \omega_2^2(x_2 - x_1) \end{cases} \qquad (3)$$

In order to use the measured absolute acceleration $\ddot{x}_0(t)$ as the input of system (1), the following variables are introduced:

$$y_1 = x_1 - x_0, \ \dot{y}_1 = \dot{x}_1 - \dot{x}_0, \ \ddot{x}_1 = \ddot{y}_1 + \ddot{x}_0, y_2 = x_2 - x_1,$$
$$\dot{y}_2 = \dot{x}_2 - \dot{x}_1, \ \ddot{x}_2 = \ddot{y}_2 + \ddot{x}_1 = \ddot{y}_2 + \ddot{y}_1 + \ddot{x}_0 \qquad (4)$$

The equations of motion (3) become

$$\begin{cases} \ddot{y}_1 = -2\zeta_1\omega_1\dot{y}_1 - \omega_1^2 y_1 + 2\mu\zeta_2\omega_2\dot{y}_2 + \mu\omega_2^2 y_2 - \ddot{x}_0 \\ \ddot{y}_2 = 2\zeta_1\omega_1\dot{y}_1 + \omega_1^2 y_1 - 2(1+\mu)\zeta_2\omega_2\dot{y}_2 - (1+\mu)\omega_2^2 y_2 \end{cases} \qquad (5)$$

The equivalent damping ratios ζ_1, ζ_2 assess the damping capacity of shock absorbers from primary and secondary suspensions. Usually, these coefficients are determined by equivalent linearization of the nonlinear damping characteristics recorded for an imposed cyclic relative motion between the mounting ends of the shock absorber. The coefficient c_e of linear equivalent damping characteristic $F_e(\dot{y}) = c_e\dot{y}$ is determined by imposing the condition to provide same dissipated power P_d as the measured damping characteristic for $-\dot{y}_{max} \leq \dot{y}(t) \leq \dot{y}_{max}$.

$$P_d = \int_{-\dot{y}_{max}}^{\dot{y}_{max}} |F(\dot{y})| d\dot{y} = \int_{-\dot{y}_{max}}^{\dot{y}_{max}} |F_e(\dot{y})| d\dot{y} = c_e\dot{y}_{max}^2, \ c_e = \frac{P_d}{\dot{y}_{max}^2} \qquad (6)$$

The application of equivalent linearization method is illustrated for the railway vehicle shock absorbers T69 used for secondary suspensions [10].

In Fig.2 are plotted the measured damping characteristic $F(\dot{y})$ and the linear equivalent damping characteristic $F_e(\dot{y}) = c_e \dot{y}$. The value of linear equivalent damping coefficient $c_e = 66559 N/ms^{-1}$ was obtained from (6), for $\dot{y}_{max} = 0.275 ms^{-1}$, $P_d = 5033.5W$.

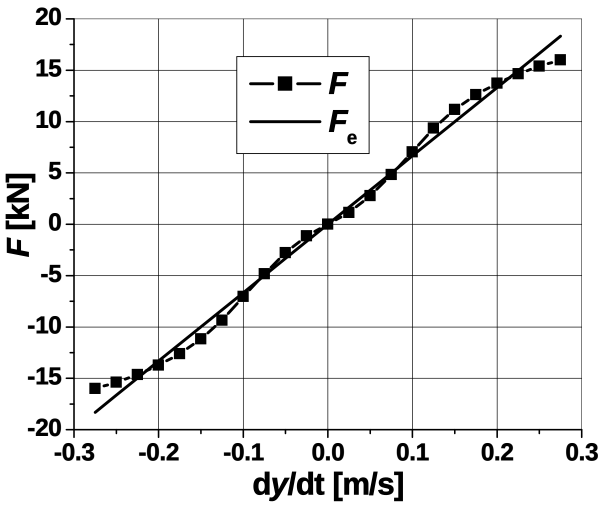


Figure 2. Nonlinear and linear equivalent damping characteristics of T69

3. APPLICATION OF PHASE-FREQUENCY CHARACTERISTIC FOR ASSESSING OF DAMPERS CONDITION

Let $H_{u_2 u_1}(\mu)$ be the frequency response function corresponding to any chosen input and output variables $u_1(t)$ and $u_2(t)$ of suspension system model shown in Fig.1. Then the expressions of frequency response functions of interest for the present study can be expressed in terms of $H_{\ddot{y}_1 \ddot{x}_0}(\mu), H_{\ddot{y}_2 \ddot{x}_0}(\mu)$ by the following relations:

$$H_{\ddot{x}_1 \ddot{x}_0}(\mu) = H_{\ddot{y}_1 \ddot{x}_0}(\mu) + 1,\ H_{\ddot{x}_2 \ddot{x}_0}(\mu) = H_{\ddot{y}_2 \ddot{x}_0}(\mu) + H_{\ddot{x}_1 \ddot{x}_0}(\mu),\ H_{\ddot{x}_2 \ddot{x}_1}(\mu) = \frac{H_{\ddot{x}_2 \ddot{x}_0}(\mu)}{H_{\ddot{x}_1 \ddot{x}_0}(\mu)} \qquad (7)$$

The frequency response functions of relative accelerations $H_{\ddot{y}_1 \ddot{x}_0}(\omega), H_{\ddot{y}_2 \ddot{x}_0}(\omega)$ are determined from equations (5) by taking $\ddot{x}_0(t) = \ddot{X}_0 \, exp(i\mu t)$ and $\ddot{y}_k(t) = H_{\ddot{y}_k \ddot{x}_0}(\mu) \, \ddot{X}_0 \, exp(i\mu t), k = 1,2$. The following algebraic linear system for these frequency response functions is obtained:

$$\begin{cases} \left[a_1(\mu) - \mu^2\right] H_{\ddot{y}_1 \ddot{x}_0}(\mu) - \text{«} a_2(\mu) H_{\ddot{y}_2 \ddot{x}_0}(\mu) = \mu^2 \\ a_1(\mu) H_{\ddot{y}_1 \ddot{x}_0}(\mu) - \left[(1+\text{«}) a_2(\mu) - \mu^2\right] H_{\ddot{y}_2 \ddot{x}_0}(\mu) = 0 \end{cases} \qquad (8)$$

where

$$a_k(\mu) = \mu_k^2 + 2i\text{¸}_k \mu_k \mu,\ \ k = 1,2 \qquad (9)$$

The solution of system (8) is

$$H_{\ddot{y}_1 \ddot{x}_0}(\mu) = \frac{\left[(1+\text{«}) a_2(\mu) - \mu^2\right] \mu^2}{a_1(\mu) a_2(\mu) - \left[(1+\text{«}) a_2(\mu) + a_1(\mu)\right] \mu^2 + \mu^4}$$

$$H_{\ddot{y}_2 \ddot{x}_0}(\mu) = \frac{\mu^2 a_1(\mu)}{a_1(\mu) a_2(\mu) - \left[(1+\text{«}) a_2(\mu) + a_1(\mu)\right] \mu^2 + \mu^4} \qquad (10)$$

The efficiency of primary and secondary suspensions dampers is measured by the values of damping ratios ¸_1 and ¸_2. Usually, the minimum and maximum values of these coefficients are $\text{¸}_{min} = 0.2$ and

$¸_{max} = 0.45$. In order to assess the condition of secondary suspension one must determine the most relevant frequency domain for filtering the measured signals $\ddot{x}_0(t), \ddot{x}_1(t), \ddot{x}_2(t)$. This information is provided by the behavior of frequency characteristic of the phase difference $cos¯_{\ddot{x}_2\ddot{x}_1}(\mu, ¸_1, ¸_2)$ between the car body and bogy accelerations. Introducing the damping efficiency index $I_d(\mu, ¸_1, ¸_2) = \left|cos¯_{\ddot{x}_2\ddot{x}_1}(\mu, ¸_1, ¸_2)\right|$, its variation versus frequency is given by

$$I_d(\mu, ¸_1, ¸_2) = \left| \frac{Re\left[H_{\ddot{x}_2\ddot{x}_1}(\mu, ¸_1, ¸_2)\right]}{H_{\ddot{x}_2\ddot{x}_1}(\mu, ¸_1, ¸_2)} \right| \tag{11}$$

Alternatively, by taking $\ddot{x}_1(t) = \ddot{X}_1 cos\,\mu t$ as bogie input, the body output can be written as $\ddot{x}_2(t) = \ddot{X}_2(\mu, ¸_1, ¸_2) cos\left[\mu t + ¯(\mu)\right]$. Then the damping efficiency index $I_d(\mu, ¸_2, ¸_1)$ is given by

$$I_d(\mu, ¸_1, ¸_2) = \left| \frac{\mu}{2® \pm_{\ddot{x}_2} \pm_{\ddot{x}_1}} \int_0^{\frac{2®}{\mu}} \ddot{x}_2(t)\ddot{x}_1(t)dt \right| \tag{12}$$

where $\pm_{\ddot{x}_2}, \pm_{\ddot{x}_1}$ are the mean square values of body and bogy accelerations.

4. EXPERIMENTAL RESULTS

The vertical accelerations $\ddot{x}_0(t), \ddot{x}_1(t), \ddot{x}_2(t)$ were measured simultaneously in operating conditions on a railway car with bogies Y32. The vehicle speed $V = 128\,km/h$, was kept approximately constant during the measurements. The recorded sample length was $T = 60s$, which corresponds to a railway length of 2133m. The continuous random records were sampled at points $at = 0.005s$ apart. The resulting sampling size $N = 12000$ is sufficiently large to provide a good accuracy for various estimates of statistical characteristics of stationary random processes [11],[12]. In Fig.3 are shown the time histories of recorded acceleration signals. In Fig.4 are plotted the Power Spectral Densities (PSD) of recorded accelerations, estimated by band-pass filtering with constant percentage bandwidth of one fourth octave (17.3%).

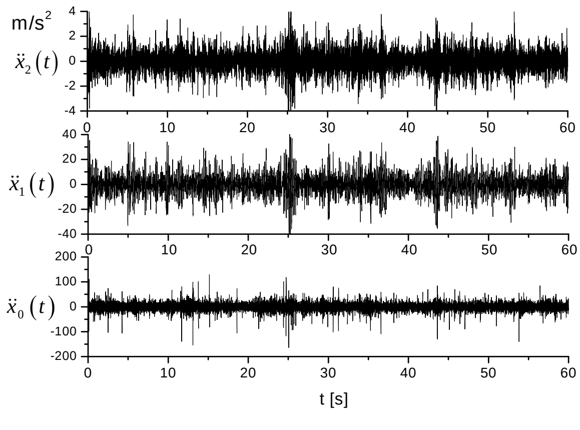


Figure 3. Time histories of recorded accelerations

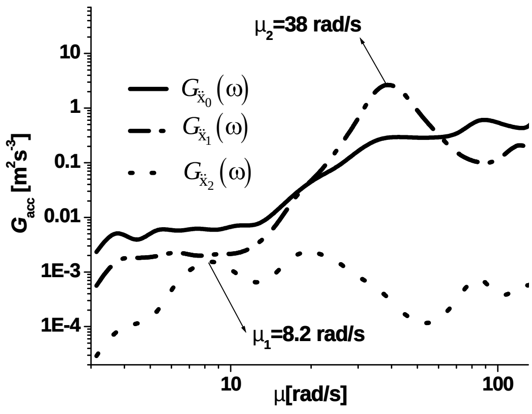


Figure 4. PSD of measured accelerations

The central frequencies of used filters are given by:

$$f_{c,k} = 2^{\frac{k-1}{4}} f_{c,1},\; k = 1,2,...,K,\; f_{c,1} = 0.5Hz, K=25,\; f_{c,K} = 32Hz \tag{13}$$

As shown in Fig.4, from the plots of $G_{\ddot{x}_1}(\mu)$ and $G_{\ddot{x}_2}(\mu)$ one can estimate the resonance frequencies of primary and secondary suspensions:

$$\mu_1 = 2®\, f_1 = 8.2rad/s,\; f_1 = 1.3Hz;\; \mu_2 = 2®\, f_2 = 38rad/s, f_2 = 6\,Hz \tag{14}$$

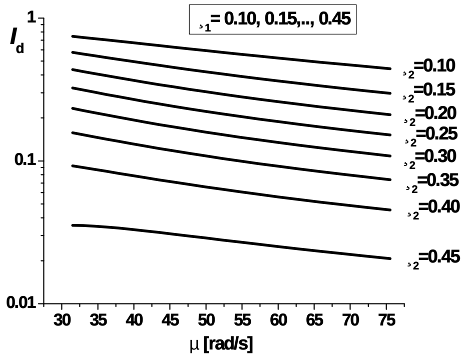
These values and the following inertial parameters:

$$m_1 = 485Kg,\ m_2 = 9000Kg,\ « = \frac{m_2}{m_1} = 18.557 \qquad (15)$$

will be used throughout of this paper.

5. ANALYSIS BEHAVIOR OF DAMPING EFFICIENCY INDEX IN FREQUENCY DOMAIN

In what follows, the application of damping efficiency index method is illustrated for assessment of the damping capacity of shock absorbers from secondary suspension. In this case, the values of damping efficiency index were calculated by using the relation (11). Fig. 5 shows the variation of $I_d\left(μ, ¸_2, ¸_1\right)$ for: $30rad/s \le μ \le 75rad/s,\ \ 0.1 \le ¸_2, ¸_1 \le 0.45$



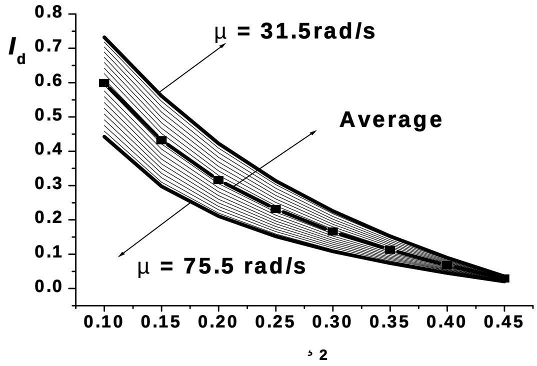


Figure 5. Variation of I_d in frequency domain

Figure 6. Variation of damping efficiency index vs. $¸_2$

It should be mentioned that for any chosen value of damping ratio $¸_2$ one obtains the same plot for all values $0.1 \le ¸_1 \le 0.45$ .This result shows that the assessment of damping efficiency for shock absorbers of secondary suspension is not influenced by the condition of primary suspension dampers. The numerical data plotted in Fig.5 can be rearranged such that to obtain the variation of damping efficiency index as function of damping ratio $¸_2$ for different values of frequency $μ$. The resulting curves and their average

$$I_d\left(¸_{2j}\right) = \frac{1}{20}\sum_{i=1}^{20} I_d\left(μ_i, ¸_{2j}\right),\ 0.1 \le ¸_{2j} \le 0.45,\ j = \overline{1,8} \qquad (16)$$

are shown in Fig.6. The average plot can be used to assess the equivalent damping ratio $¸_2$ for secondary suspension independent of equivalent damping ratio $¸_1$ of primary suspension.

6. RESULTS OF NUMERICAL SIMULATIONS BASED ON EXPERIMENTAL DATA

The measured acceleration signal $\ddot{x}_0\left(t\right)$ was taken as the input of analytical model portrayed by equations (4),(5) with parameters (14) and (15) in order to simulate the time histories of acceleration output $\ddot{x}_1\left(t\right), \ddot{x}_2\left(t\right)$ for different values of damping ratios $¸_1$ and $¸_2$. The simulated output accelerations were filtered by a band-pass filter with the frequency bandwidth determined from the results obtained in previous section: $30rad/s < μ < 75rad/s$ ($5Hz < f < 12Hz$). In Fig. 7 are presented samples from filtered body and bogie acceleration signals.

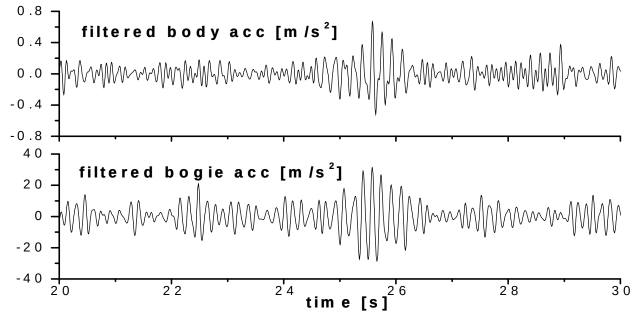


Figure 7. Samples from band –pass filtered body and bogie accelerations

The values of damping efficiency index were estimated using the filtered acceleration outputs $\ddot{x}_{1f}\left(t\right), \ddot{x}_{2f}\left(t\right)$ by applying the following relations [11],[12], similar to relation (12):

$$I_d = \frac{1}{T\sigma_{\ddot{x}_{1f}}\sigma_{\ddot{x}_{2f}}}\int_0^T \ddot{x}_{1f}\left(t\right)\ddot{x}_{2f}\left(t\right)dt,\ \sigma_{\ddot{x}_{if}} = \sqrt{\frac{1}{T}\int_0^T \ddot{x}_{if}^2\left(t\right)dt},\ i = 1,2 \tag{17}$$

Fig.8 presents the variation of simulated values (15) of cross correlation as function of ζ_2 for different values of ζ_1. In order to highlight the fact that the damping efficiency index is practically independent of the wheel vertical acceleration $\ddot{x}_0\left(t\right)$, a sample time history of a band limited white noise, within the frequency range 0.5-30Hz, was simulated as the input of system (5).

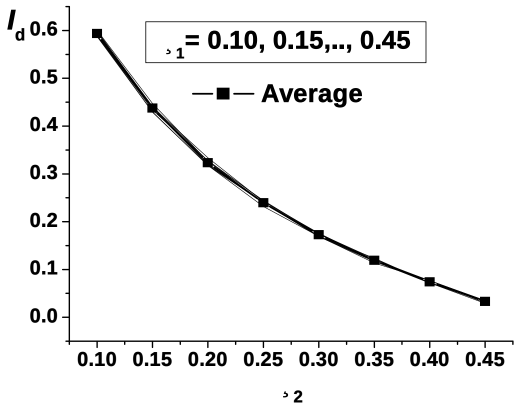


Figure 8. Variation of simulated damping efficiency index for measured input acceleration $\ddot{x}_0\left(t\right)$

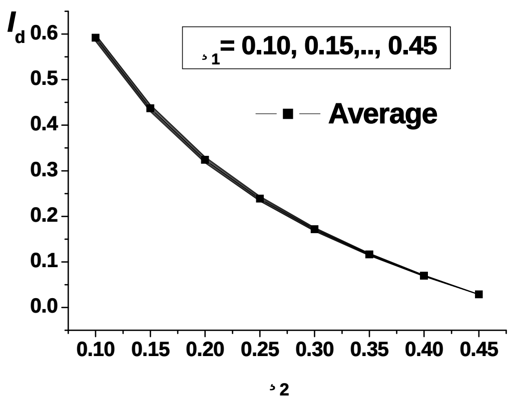


Figure 9. Variation of simulated damping efficiency index for band limited white noise input $\ddot{x}_0\left(t\right)$

The variation of index $I_d\left(\zeta_2\right)$ was calculated, as before, for different values of damping ratio ζ_1 by band-pass filtering of the simulated output accelerations. The results are plotted in Fig.9. The average curves from Figs.6, 8 and 9 are shown comparatively in Fig.10.

The results obtained in time domain by numerical integration of equations (5) using the measured and simulated input $\ddot{x}_0\left(t\right)$ are in very well agreement with those calculated in frequency domain by using harmonic inputs. In Fig.11 is shown the analytical fit (18) of damping ration ζ_2 as function of damping efficiency index.

$$\zeta_2 = \zeta_a + \zeta_b \exp\left(-\frac{I_d}{\sim}\right),\ \zeta_a = 0.03167,\ \zeta_b = 0.46062,\ \sim = 0.31583 \tag{18}$$

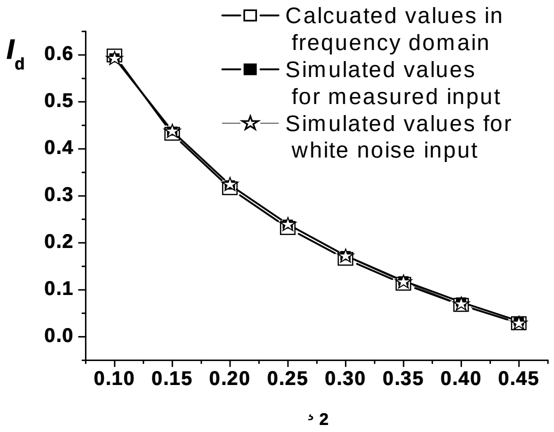


Figure 10. Comparison of simulated and calculated values of damping efficiency index $I_d(\zeta_2)$

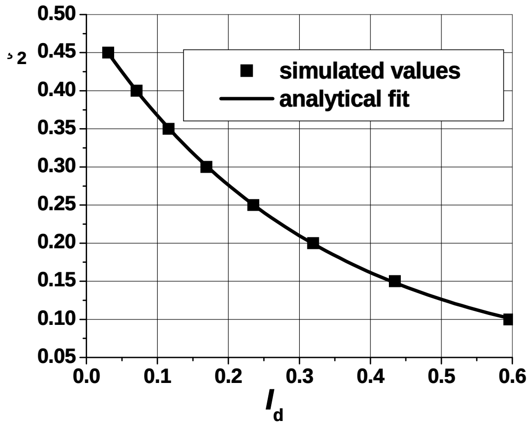


Figure 11. Variation of damping ratio ζ_2 vs. I_d

The value of damping efficiency index, obtained by applying relation (17) for measured and filtered acceleration signals $\ddot{x}_1(t)$ and $\ddot{x}_2(t)$ is $I_d = 0.247$. Introducing this value in relation (18) yields the value of linear equivalent damping ratio $\zeta_2 = 0.242$. This result shows that the damping capacity of the secondary suspension was between the allowed limits.

7. CONCLUSIONS

In this paper is proposed a simple and effective method to assess the condition of secondary suspension dampers for railway vehicles. The method is based on the measurement of vertical accelerations in normal operating conditions on bogie frame and body floor. The values of the damping efficiency index used in the paper depend only on the equivalent damping ratio of secondary suspension, being practically independent of the equivalent damping ratio of primary suspension. By appropriate band-pass filtering of the bogy and body vertical acceleration, an analytical relation between the equivalent damping ratios of secondary suspension and the values of damping efficiency was assessed. Applying this relation to the recorded signals, it was obtained an admissible value for equivalent damping ratio of secondary suspension.

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