THE IMPACT OF SELECTED STRUCTURAL PARAMETERS IN UNSPRUNG INDUSTRIAL VEHICLES ON THEIR LONGITUDINAL OSCILLATIONS

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Abstract

The study presents an analysis of the impact of selected structural parameters in unsprung industrial vehicles on their longitudinal oscillations. It was conducted on the basis of a simplified, linear, two-dimensional model in the form of a parametric spectral transmittance. Transfer function was developed on a number of simplifying assumptions, of which the more significant ones include: the assumption of small oscillations, the assumption that wheels are in constant contact with the ground, lack of the impact of the horizontal components of the input on the vehicle dynamics, lack of the impact of the wheel slip and the drive system on the vehicle dynamics, single-point contact of the wheels with the ground etc.

The qualitative correctness of the obtained results was partly verified by means of computations using a more complex simulation model of a vehicle by means of the MSC.Adams system. The conducted analyses demonstrate that structural parameters such as the moment of inertia and the value of the tire damping have a crucial impact on the intensity of longitudinal oscillations in unsprung industrial vehicles. The impact of the oscillation coupling coefficient and the mass is significantly smaller.

Keywords: *oscillations, modelling, industrial vehicles, longitudinal dynamics of industrial vehicles, spectral transmittance*

1. Introduction

The vast majority of industrial vehicles used, for instance, in the construction industry and mining are unsprung. Consequently, when these vehicles are operated at high velocities certain unfavorable dynamic phenomena occur. High velocities in this respect mean velocities exceeding 10 km/h (2,8 m/s). These phenomena may include, for instance, the phenomenon of snaking (it occurs in articulated vehicles), the phenomenon of the wheel breaking away from the surface or the phenomenon referred to as porpoising (large amplitude longitudinal oscillations of the vehicle). The majority of dynamic phenomena in real objects are interrelated in different ways. Their simultaneous analysis usually leads to difficulties in an unambiguous assessment of the obtained results. Therefore it is common to conduct a preliminary, simplified analysis of a single, separate phenomenon. This paper will include the presentation of the results of such an analysis with reference to the phenomenon of porpoising in unsprung vehicles.

2. The determination of a vehicle model in the form of spectral transmittance

The dynamics of the system may be relatively easily analyzed using standardized procedures if it can be described, with a satisfactory approximation, by means of the transfer function.
In the present study the parametric transfer function was determined assuming as the starting point the two-dimensional physical model of a vehicle as presented in Fig. 1.

This transfer function was developed on a number of simplifying assumptions, of which the more significant ones include:

- the assumption of small oscillations,
- the assumption that wheels are in constant contact with the ground,
- lack of the impact of the horizontal components of the input on the vehicle dynamics,
- lack of the impact of the wheel slip and the drive system on the vehicle dynamics,
- single-point contact of the wheels with the ground etc.

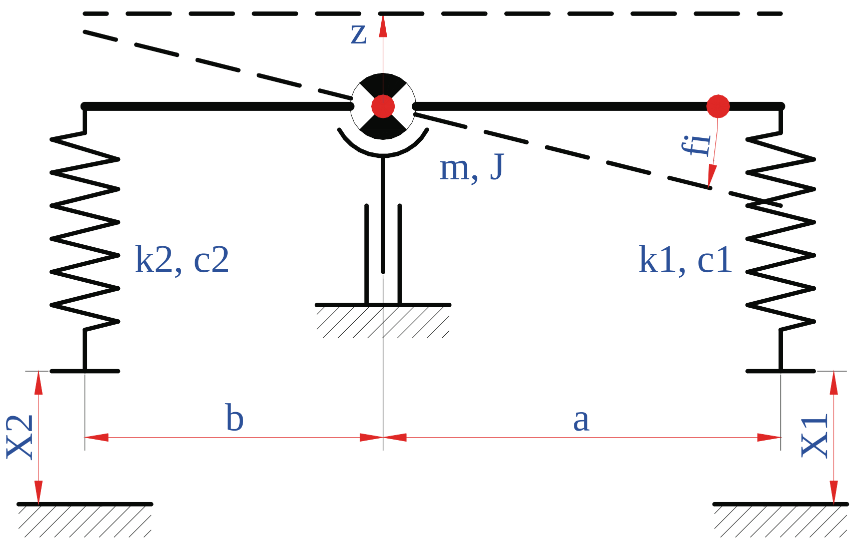

Fig. 1. The applied physical model of the vehicle

The equations of motion for the mathematical model were derived using the Lagrange's equations of the second kind. Having applied Laplace transforms and the necessary transformations to the derived equations two transfer functions of the form [2] were obtained:

$$T_{x_1} = \frac{\varphi(s)}{x_1(s)} , \tag{1}$$

$$T_{x_2} = \frac{\varphi(s)}{x_2(s)} . \tag{2}$$

where:

x_1, x_2 – kinematic inputs from the surface affecting the front and the rear bridge of the vehicle respectively,

φ – the angle of the body displacement in the longitudinal plane of the vehicle,

s – operational variable.

Given that the inputs affecting the front axle of the vehicle are connected with the inputs affecting the rear axle of the wheel additional relations were formulated as follows:

$$x_1(t) = x_2(t + \Delta t) , \tag{3}$$

$$\Delta t = \frac{a+b}{V} . \tag{4}$$

where:

V – vehicle velocity.

Taking the relations (3) and (4) into account in the mathematical model and applying the substitution $s=j\omega$ transfer functions (1) and (2) were finally substituted with one spectral transmittance in the form [1]:

$$T\ (j\omega) = \frac{\varphi(j\omega)}{x_1(j\omega)}\ . \tag{5}$$

3. The investigation of the longitudinal dynamics of an industrial vehicle on the basis of its spectral transmittance

Prior to the investigation numerical values typical of an exemplary real industrial vehicle were entered in the position of the parameters of spectral transmittance. These values are presented in Tab. 1. The model developed in this manner is referred to as the base vehicle model (base model). The investigation commenced with the development of the amplitude characteristic of the longitudinal oscillations of a base vehicle versus velocity. Fig. 2 presents the developed characteristic. This characteristic demonstrates that the maximum values of the spectral transmittance module (maximum gains) occur at the input function frequency of 1,7 Hz (referred to as the resonant frequency). However, gains at the resonant frequency do not assume equally high values for all velocities. In the case when the wheelbase is the integral multiple of the exciting wavelength the resonant oscillations practically do not occur.

Tab. 1. Parameters of the base vehicle

k_1	1 100 000 N/m	m	11 053 kg
k_2	1 100 000 N/m	J	38 412 kg m^2
c_1	11 400 Ns/m	a	1,475 m
c_2	11 400 Ns/m	b	1,505 m

Real vehicles usually do not move on rough roads, which can be described by means of a single harmonic function. The generally adopted description is that of the random road roughness by means of the power spectral density of a rough surface. In order to analyze the nature of longitudinal displacements of a vehicle with this type of the input function, the power spectral density of these displacements was determined applying the following equation [2]:

$$G_w(\omega) = \left|T(j\omega)\right|^2 \cdot G_d(\omega) = \left|T(j\omega)\right|^2 \cdot G_d(\omega_0) \cdot \left(\frac{\omega_0}{\omega}\right)^w . \tag{6}$$

where:

$G_d\ (\omega)$ – the spectral density of a rough road at the angular frequency ω,

w – undulation coefficient (assumed w=2),

ω_0 – reference angular frequency (ω_0 =1m^{-1}),

$G_d\ (\omega_0)$ – road roughness index (power spectral density for the reference angular frequency, assumed $G_d\ (\omega_0)$ =155cm^3).

The obtained values of the power spectral density of the vehicle displacements versus the velocity and the input function frequency are presented in Fig. 3. The crucial qualitative difference between the values in Fig. 3 and the values in Fig. 4 is that in the case of the latter there is a significant increase in the power spectral density at the resonant frequency as a function of the velocity.

This tendency is caused by the nature of random inputs. The amplitudes of harmonic components increase as the wavelength increases. Given this fact and the observation that as the velocity increases the resonant oscillations cause inequalities with increasing wavelengths the curve shapes in Fig. 4 seem to be obvious.

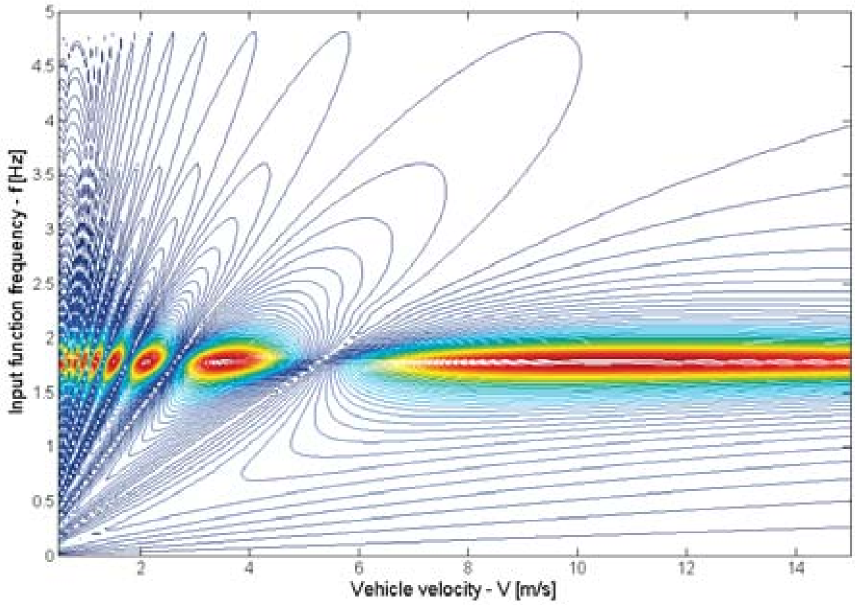

Fig. 2. The amplitude characteristic of the longitudinal displacements of the base vehicle vs. velocity

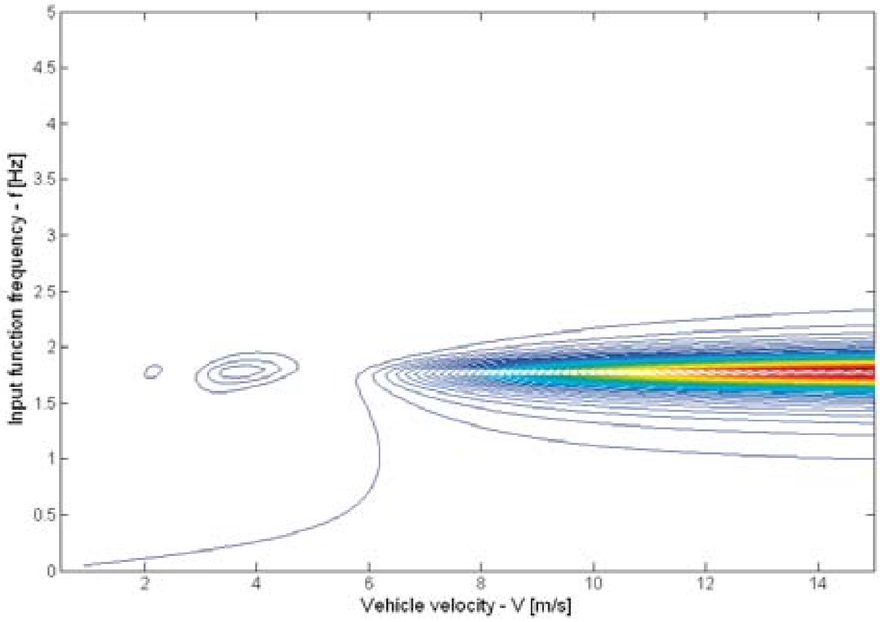

Fig. 3. Power spectral densities of the longitudinal displacements of the base vehicle at an exemplary random input function

The subsequent stage of the investigation involved checking the impact of the values of the oscillation coupling coefficient "s" on the values of the power spectral density of the vehicle longitudinal displacements at resonant frequencies. In this study the oscillation coupling coefficient is understood as the coefficient computed from the relation:

$$s = \frac{k_1 \cdot a - k_2 \cdot b}{J} \qquad [\frac{1}{m \cdot s^2}] \ . \tag{7}$$

In order to conduct the aforementioned checks two new versions of the vehicle numbered 1 and 2 were defined, which differed from the base version by the values of the "s" coefficient. The change of this coefficient for versions 1 and 2 was obtained through the change of the base vehicle wheel rigidity and the proportional change of their corresponding viscous damping coefficients. The modified parameters of the new versions are presented in Tab. 2. The results of the computations for the vehicle model in versions 1 and 2 are presented in Fig. 4. They demonstrate a relatively insignificant impact of the oscillation coupling coefficient on the values of the power spectral density of the longitudinal displacements of the vehicle at resonant frequencies.

Apart from checking the impact of the value of the oscillation coupling coefficient on the phenomenon of porpoising the impact of the vehicle mass, the moment of inertia of the vehicle and the tire damping on this phenomenon was additionally checked. The results of the relevant computations are presented in Fig. 5. They may be summed up with a conclusion that the vehicle mass has practically no impact on the phenomenon of porpoising, whereas the impact of the moment of inertia and the tire damping is very significant.

Tab. 2. The differences between the base model and the models in versions 1 and 2

	Base model	Version 1	Version 2
k_1	1 100 000 N/m	1 620 547 N/m	558 064 N/m
k_2	1 100 000 N/m	600 000 N/m	1 620 547 N/m
c_1	11 400 Ns/m	16 794 Ns/m	5783 Ns/m
c_2	11 400 Ns/m	6 218 Ns/m	16 794 Ns/m
s	-0,86 1/(ms^2)	38,72 1/(ms^2)	-42,06 1/(ms^2)

Tab. 3. The differences between the base model and the models in versions 3, 4 and 5

	Base model	Version 3	Version 4	Version 5
c_1 [Ns/m]	11400	11400	11400	14 820
c_2 [Ns/m]	11400	11400	11400	14 820
m [kg]	11 053	25 781	11 053	11 053
J [kgm^2]	38 412	38 412	64 671	38 412

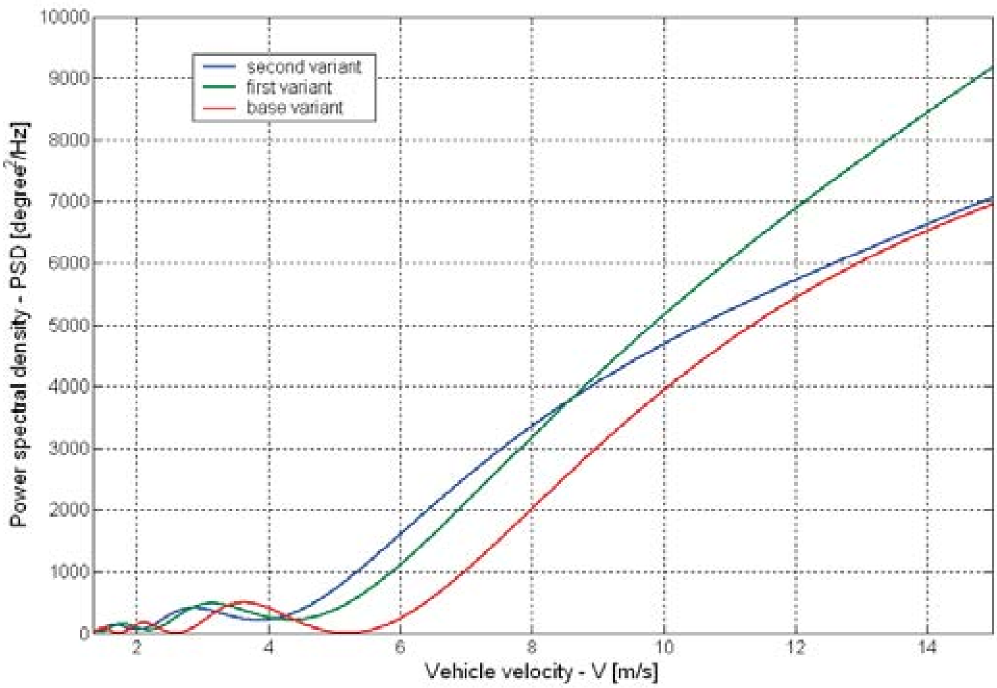

Fig. 4. The computed power spectral densities of vehicle longitudinal displacements in different versions

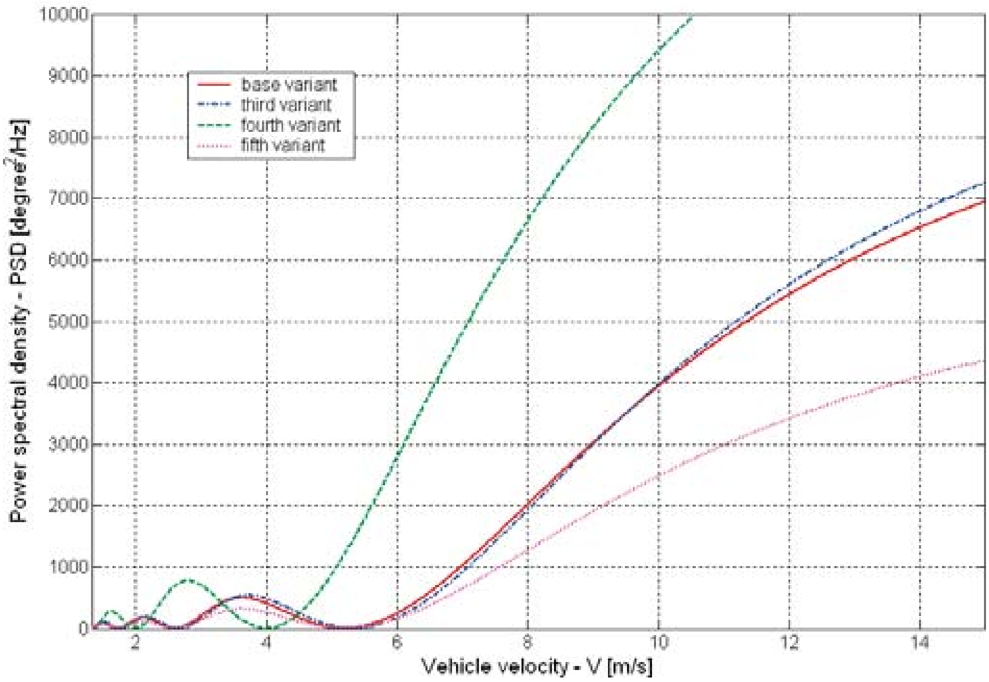

Fig. 5. The computed power spectral densities of vehicle longitudinal displacements in different versions

4. Checking simulation

The simplicity of the model applied to conduct the analyses presented in item 3 raised doubts as to the reliability of the obtained results. Given the above sample checking computations based on a more extended mathematical description of the vehicle were conducted. This description included the interaction of the tires with the surface according to the model referred to as the Fiala model. The checking and simulation computations were conducted using the MSC.Adams environment. These included driving a virtual vehicle on roads with harmonic inputs and on a road with random inputs. The data obtained from driving on roads with harmonic roughness were used to develop, for selected velocities, the diagrams of the relation between the amplitude of the vehicle longitudinal displacements and the harmonic input function frequency. These are presented in Fig. 6.

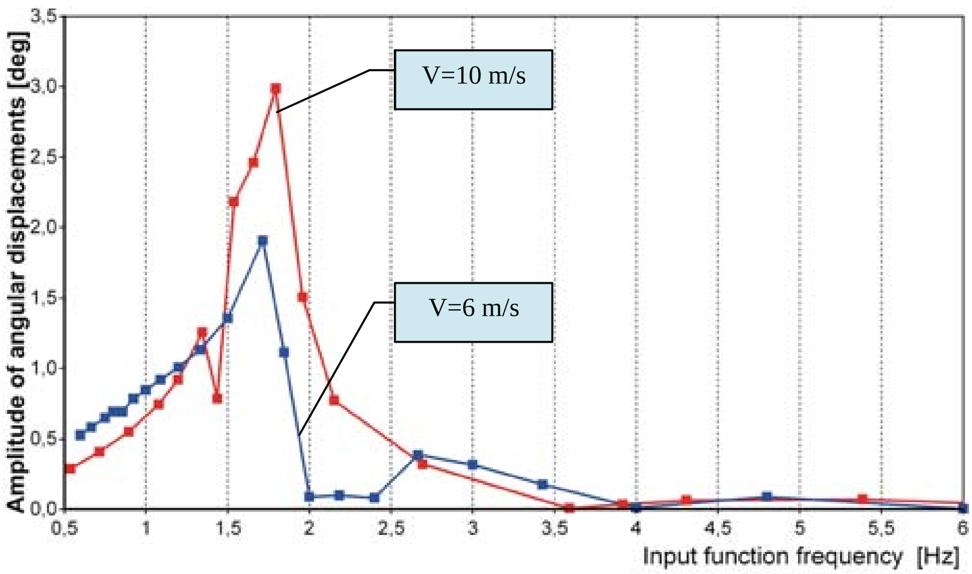

Fig. 6. The values of the angular displacement amplitudes of the virtual vehicle body riding on roads with harmonic roughness of the 15mm amplitude and various wavelengths

The analysis of these diagrams and their comparison with the diagram in Fig. 2 allowed the author to draw a conclusion about the large qualitative similarity of the results obtained from the investigations conducted on the basis of both models of the vehicle. However, at this point it is noteworthy that the diagrams in Fig. 6 were drawn on the basis of rides with virtual vehicles where vehicle wheels were (except when the frequency of the input function and the resonant frequency overlapped) in constant contact with the ground.

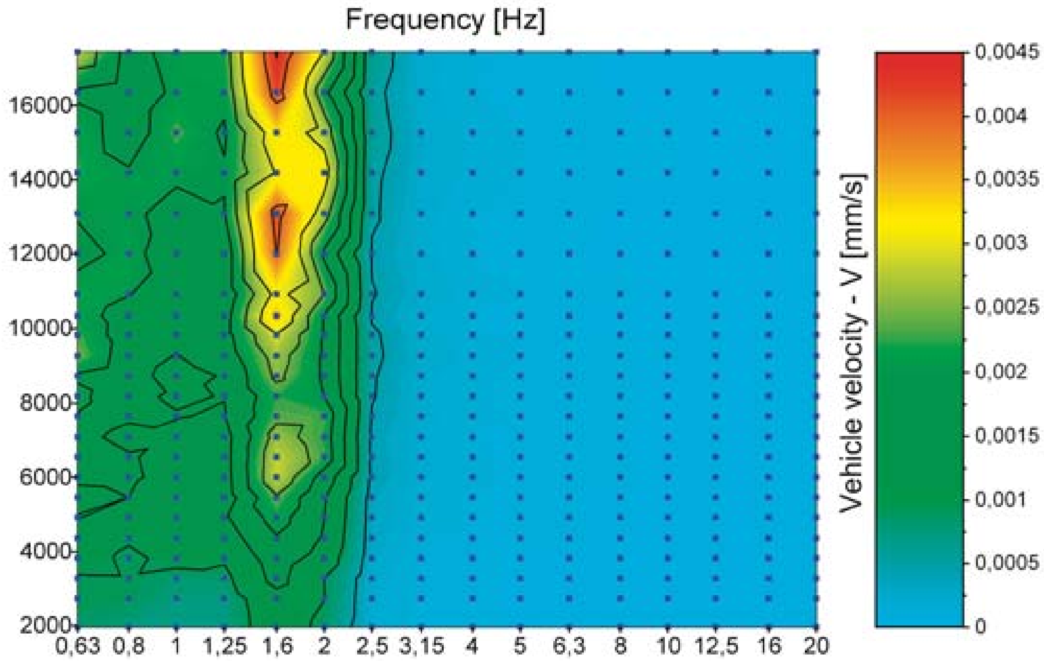

Fig. 7. The rms values of the angular displacements of the virtual vehicle body riding on a road with random roughness (G_d (ω_0) =155cm^3, w=2)

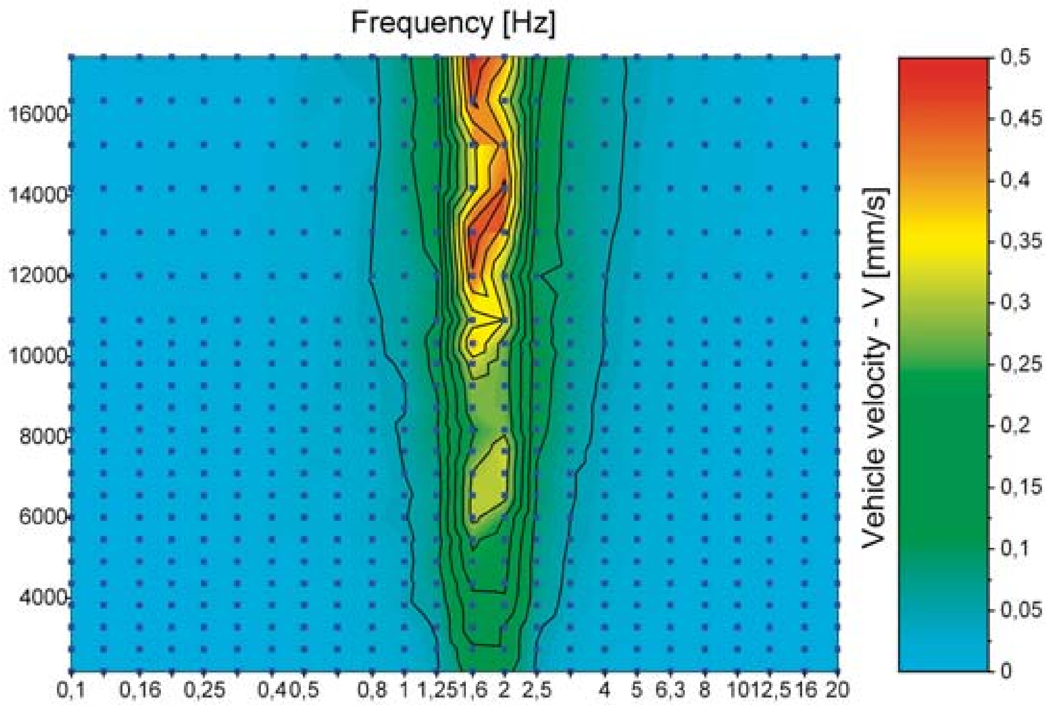

Fig. 8. The rms values of the angular acceleration of the virtual vehicle body driving on a road with random roughness (G_d (ω_0) =155cm^3, w=2)

The virtual investigation on a road with random roughness was conducted in order to obtain the time history of the longitudinal displacements of the vehicle and the accelerations of these displacements. They were the basis to determine the rms values efficient in standard third-octave bands. This method of the data analysis, instead of the classic spectral analyses, was selected in order to obtain transparent diagrams. The already processed results of the simulation computations in the manner described above are presented in Fig. 7 and 8. When compared with the results in Fig. 3 one may notice certain qualitative differences. First of all, all of the local minima in Fig. 3 do not overlap with the local minima in Fig. 7.

It is supposed that this may be due to the manner in which the virtual random road roughness in the MSC.Adams system was generated and to the fact that the time the vehicle traveled on them was too short (150s). However, the tendency for the increase of the value along with the vehicle velocity is maintained in both diagrams.

5. Summary

The conducted analyses demonstrate that structural parameters such as the moment of inertia and the value of the tire damping have a crucial impact on the intensity of longitudinal oscillations in unsprung industrial vehicles. The impact of the oscillation coupling coefficient and the mass is significantly smaller.

References

[1] Kamiński, E., Pokorski, J., *Dynamika zawieszeń i układów napędowych pojazdów samochodowych*, WKiŁ, Warszawa, 1983.

[2] Kosiara, A., *Opracowanie układu redukcji drgań wzdłużnych nieresorowanego pojazdu przemysłowego*, Instytut Konstrukcji i Eksploatacji Maszyn Politechniki Wrocławskiej, (unpublished report), Wrocław, 2008.