THE EFFECT OF STIFFNESS AND DAMPING OF THE SUSPENSION SYSTEM ELEMENTS ON THE OPTIMISATION OF THE VIBRATIONAL BEHAVIOUR OF A BUS

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Abstract: The effects of spring stiffness and shock absorber damping on the vertical acceleration of the driver's body, suspension deformation and dynamic wheel load were investigated, with the purpose to define recommendations for selecting oscillation parameters while designing the suspension system of a (intercity) bus. Oscillatory responses were analysed by means of a bus oscillatory model with linear characteristics and three degrees of freedom, with excitation by the Power Spectral Density (PSD) of the roughness of asphalt-concrete pavement in good condition. The analysis was conducted through a simulation, in frequency domain, using statistical dynamics equations. A programme created in the software pack *MATLAB* was used to analyse the transfer functions, spectral density and RMS of oscillatory parameters. The results of the analysis show that the parameters which ensured good oscillatory comfort of the driver were conflicting with the parameters which ensured the greatest stability of the bus and the corresponding wheel travel. In terms of the driver's oscillatory comfort, the bus suspension system should have a spring of small stiffness and a shock absorber with a low damping coefficient. In terms of active safety, it should have a spring of small stiffness and a shock absorber with a high damping coefficient, while minimum wheel motion requests for springs of great stiffness and shock absorbers with a high damping coefficient.

Keywords: bus, oscillatory behaviour, spring, shock absorber, simulation.

1. Introduction

The suspension system is one of the main vehicle systems, significant for the realisation of comfort, stability and safety parameters. The main purpose of this system is to increase the comfort of vehicle occupants (passengers and drivers), to maintain the contact between the tyre and the road surface and to eliminate (minimize) dynamic forces which act on the load bearing vehicle structure and road surface along which the vehicle is moving.

Elastic and damping elements are built into the suspension system for the above-mentioned purposes can be successfully achieved. Pneumatic spring elements (also known as bellows or pneumatic balloons) are present as elastic elements in the suspension system of modern buses, while hydraulic shock absorbers are damping elements.

Some of the main advantages of pneumatic system compared to other suspension systems (featuring elastic springs, combined

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suspension system) are the preservation of the same floor height of buses even with different static and dynamic loads, the same oscillation frequency of sprung mass irrespective of the bus load and greater oscillatory comfort. The disadvantages of the system are its relative complexity and higher price.

The choice of the suspension system parameters comes down to the resolution of different, conflicting demands. When developing the suspension system, an optimal compromise has to be made between comfort, rattle space and the variation of forces acting in contact between tyre and road surface.

The aim of this paper is to determine the spring stiffness and shock absorber damping values of the bus suspension system, needed to have acceptable oscillatory behaviour. Three important oscillatory parameters (driver's vertical acceleration, bus suspension deformation and dynamic wheel load) in frequency domain were analysed. This type of analysis allows the choice of the values of oscillatory parameters of the bus suspension system depending on different excitation frequency values. Similarly, the analysis facilitates the choice of the oscillatory parameter values for the excitation frequency range which exerts considerable influence on the oscillatory behaviour of the bus, which is, in turn, of great importance when designing the bus suspension system.

2. Oscillatory Model of the Bus

A linear oscillatory model of the bus used in simulation, with three degrees of freedom, is shown in Fig. 1. Table 1 provides the meaning of the values from the same figure, together with parameters used in the simulation.
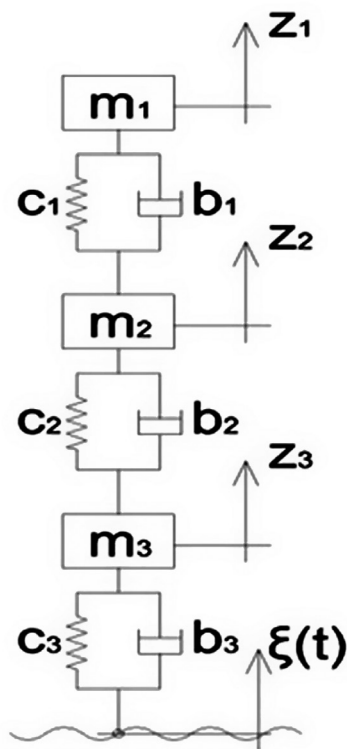
Fig. 1.
Linear Oscillatory Model of the Bus

Table 2 shows calculated eigenvalues (characteristic values), undamped natural frequencies and damped natural frequencies of the oscillatory system for the parameters in Table 1. The computed values of natural frequencies of oscillation are close to the values of typical oscillation frequencies of the bus in service (Dedović, 2004; Demić and Diligenski, 2003).

The differential equations of motion for the oscillatory system in matrix form are formulated as follows, Eq. (1):

$$[M]\{\ddot{q}\} + [B]\{\dot{q}\} + [C]\{q\} = [D]\{\dot{\xi}\} + [E]\{\xi\} \quad (1)$$

where:

Table 1
Bus Parameters

Bus parameter	Value
m_1 - mass of the driver and the seat	100 (kg)
c_1 - driver's seat suspension spring stiffness	25000 (N/m)
b_1 - driver's seat shock absorber damping	1000 (Ns/m)
m_2 - sprung bus mass on front axle	4500 (kg)
c_2 - front axle bellows stiffness	300000 (N/m)
b_2 - front axle shock absorbers damping	20000 (Ns/m)
m_3 - unsprung front axle mass	500 (kg)
c_3 - bus tyre stiffness	1600000 (N/m)
b_3 - bus tyre damping	150 (Ns/m)
$\xi(t)$ - bus excitation	

Table 2
Characteristic Values, Undamped Natural Frequencies and Damped Natural Frequencies of the Oscillatory System

Concentrated mass	Eigenvalues	Undamped natural frequencies (Hz)	Damped natural frequencies (Hz)
Driver and seat	-5.1634 ±15.1915i	2.5536	2.4178
Bus body	-1.5598 ± 7.3436i	1.1949	1.1688
Axle	-20.7600 ±56.9619i	9.6491	9.0658

$[M],[B],[C]$ - symmetric inertia matrix, damping matrix and stiffness matrix

$[E],[D]$ - stiffness and damping matrices of the input (excitation)

$\{q\},\{\dot{q}\},\{\ddot{q}\}$ - the column vector of generalised coordinates, generalised velocities and generalised accelerations

$\{\xi\},\{\dot{\xi}\}$ - the column vector of input vertical motion and velocity (excitation).

The column vector of the oscillatory system generalised coordinates is equal to (Eq. (2)):

$$\{q\} = [z_1\ z_2\ z_3]^T \tag{2}$$

The transfer functions of the oscillatory system motion, in complex form, are formulated as follows, Eq. (3):

$$H_q(if) = [[C] - (2\pi f)^2[M] + i(2\pi f)[B]]^{-1}[E + i(2\pi f)D] \tag{3}$$

where:

f – is the temporal frequency of excitation (Hz)

i – is the imaginary unit.

The transfer functions of the driver's vertical acceleration, bus suspension deflection and dynamic wheel load in complex form are represented by following expressions (Eqs. (4), (5) and (6)):

$$H_{\dot{z}_1}(if) = [(i2\pi f)^2\ 0\ 0] \cdot H_q(if) \tag{4}$$

$$H_{(z_2-z_3)}(if) = [0\ 1\ -1] \cdot H_q(if) \tag{5}$$

$$H_Z(if) = (i(2\pi f)b3 + c3) \cdot ([0\ 0\ 1] \cdot H_q(if) - 1) \tag{6}$$

3. Excitation from Road Surface

The PSD of the roughness of asphalt-concrete pavement in good condition, as a function of temporal excitation frequency, based on (Wong, 2001), is expressed as (Eq. (7)):

$$\Phi_\xi(f) = \frac{C_{sp} \cdot V^{N-1}}{f^N} \tag{7}$$

where:

f – is the temporal frequency of excitation (Hz)

V – is the speed of the bus (m/s)

C_{sp} – is the roughness coefficient (m)

N – is the wave number (-).

Table 3 provides the values of the roughness coefficient C_{sp} and wave number N for the asphalt-concrete pavement in good condition, according to (Wong, 2001).

Table 3
Roughness Coefficient and Wave Number for the Asphalt-Concrete Pavement

Pavement type and condition	N	C_{sp}
Asphalt-concrete (good)	2.1	$4.8*10^{-7}$

In the frequency range 0.5 Hz to 50 Hz (Kawamura and Kaku, 1985) the pavement roughness has the most significant impact on the vehicle's oscillatory behaviour. The PSD of the asphalt-concrete pavement roughness for the parameters in Table 3 and within the above-referenced frequency range is shown in Fig. 2.
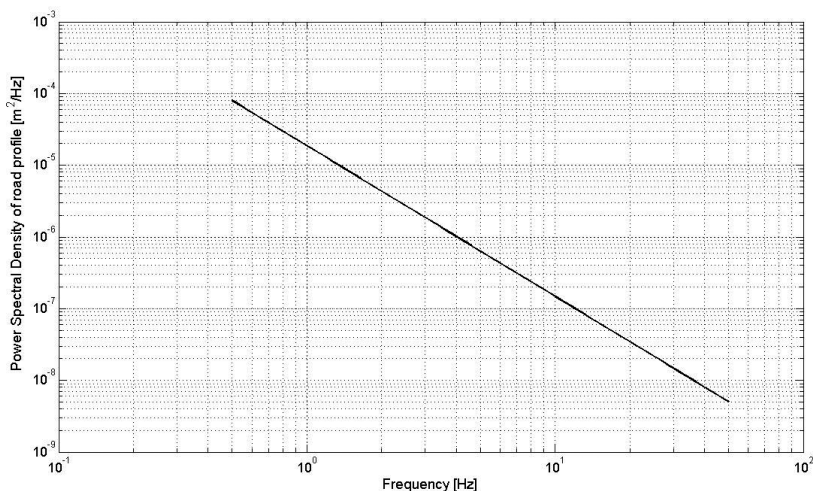
Fig. 2.
PSD of the Asphalt-Concrete Pavement Roughness as a Function of Temporal Frequency

4. Results of the Simulation

The responses of the linear oscillatory bus model are obtained using the Eq. (8):

$$\Phi_q(f) = \left| H_q(if) \right|^2 \cdot \Phi_\xi(f) \qquad (8)$$

where:

$\Phi_q(f)$ - is the PSD of the considered response of the oscillatory model (oscillatory parameter)

$\left| H_q(if) \right|^2$ - the root square of the transfer function modulus of the considered response of the oscillatory model

$\Phi_\xi(f)$ - the PSD of pavement roughness.

The paper analyses the effects of spring stiffness and shock absorber damping of the bus suspension system on three oscillatory parameters – the driver's vertical acceleration, suspension deformation and dynamic wheel load, at a constant bus speed of 100 km/h.

Figs. 3a-3c show the transfer functions and Figs. 4a-4c show the PSD of the considered oscillatory parameters for constant shock absorber damping (b2=20000 Ns/m) and different values of spring stiffness (c_2) of the bus suspension system.

The amplitudes of the driver's vertical acceleration rose with an increase of suspension spring stiffness c_2 within the domain of resonant frequency of sprung mass and resonant frequency of the driver's seat (Fig. 3a). A change in the amplitudes of acceleration was more significant within the domain of resonant frequency of sprung
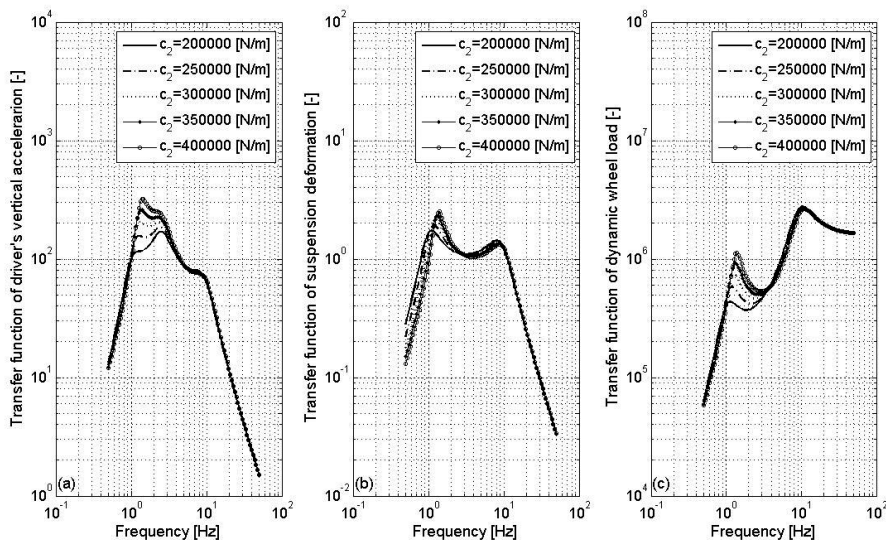
Fig. 3.

Transfer Functions of (a) Driver's Vertical Acceleration, (b) Suspension Deformation and (c) Dynamic Wheel Load, for Different Values of Spring Stiffness c_2 at the Constant Value of Shock Absorber Damping $b_2 = 20000$ Ns/m
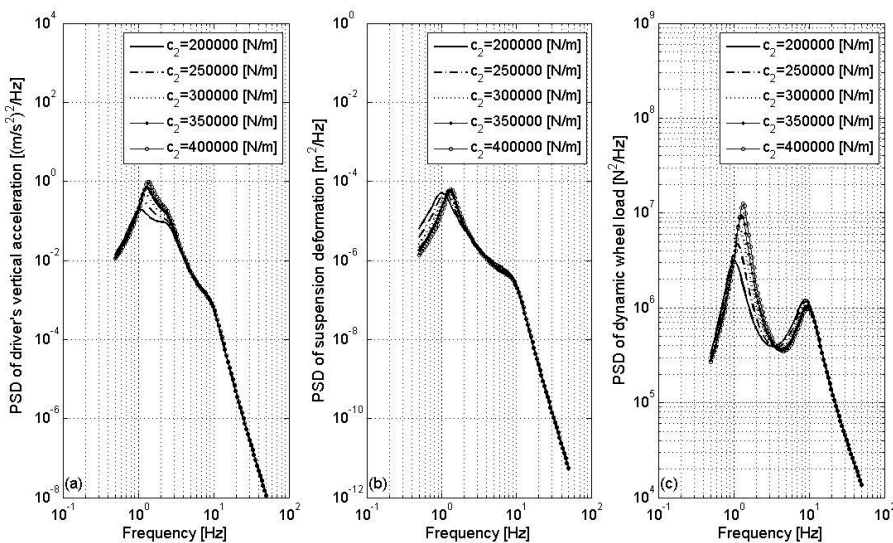
Fig. 4.

Power Spectral Density of (a) Driver's Vertical Acceleration, (b) Suspension Deformation and (c) Dynamic Wheel Load for Different Values of Spring Stiffness c_2 at the Constant Value of Shock Absorber Damping $b_2 = 20000$ Ns/m

mass (around 1.5 Hz). A change of spring stiffness did not produce any effect on the change of the driver's vertical acceleration within the domain of resonant frequency of unsprung mass of the bus (around 10 Hz).

A change in the PSD of the driver's vertical acceleration (Fig. 4a) led to the same conclusion. The driver's comfort decrease at excitation frequencies close to the resonant frequencies of the bus body as spring stiffness of the bus suspension system c_2 increases.

It was noted that resonant frequencies of sprung mass rose and the amplitudes of suspension deformation grew slightly as suspension spring stiffness c_2 increases (Figs. 3b and 4b). At lower excitation frequencies, below 1 Hz, suspension deformation decreases as spring stiffness c_2 increases (Figs. 3b and 4b).

The amplitudes of dynamic wheel load were increased for the excitation frequency range of 1 Hz to approximately 4 Hz, and for the frequency range 4 Hz to 10 Hz they were slightly decreased as suspension spring stiffness c_2 increased (Fig. 3c).

It was noted that a change in the values of the vertical load spectral densities as a function of spring stiffness c_2, was most evident at lower excitation frequencies, around 1.5 Hz (Fig. 4c). Softer springs provide better contact between the wheel and road surface, improving the dynamic performances and stability of the bus.

Figs. 5a-5c show transfer functions, while Figs. 6a-6c show the PSD of three oscillatory parameters for constant suspension spring stiffness c_2=300000 N/m and different values of shock absorber damping of the bus suspension system b_2.

The amplitudes of the driver's vertical acceleration were increased within the domain of resonant frequency of the sprung mass as shock absorber damping b_2 was increased (Fig. 5a). For excitation frequencies approximating the resonant frequency of the wheel, a change in shock absorber damping b_2 did not produce a significant effect on the driver's vertical acceleration. For the two frequency ranges of excitation (1.5 Hz to 8 Hz and above 10.5 Hz) the amplitudes of vertical acceleration were decreased as shock absorber damping b_2 was increased.

Higher values of shock absorber damping provided better oscillatory comfort for the driver at excitation frequencies approximating the resonant frequency of the bus body (Fig. 6a). Lower values of the PSD of the driver's vertical acceleration, i.e. enhanced oscillatory comfort at excitation frequencies higher than 1.5 Hz, were provided by the shock absorber of low damping coefficient values (Fig. 6a).
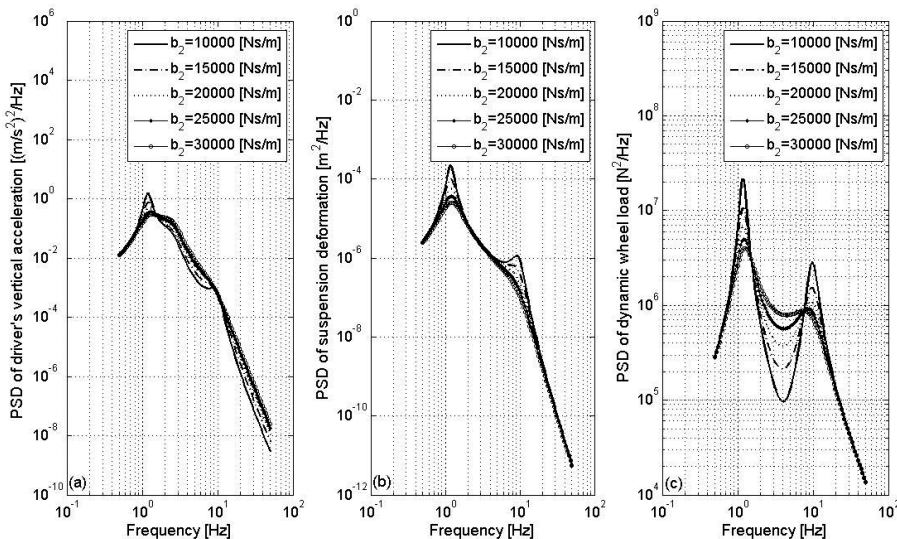
As the damping coefficient of the shock absorber b_2 increased the amplitudes of suspension deformation were decreased for excitation frequencies close to the resonant frequencies of elastically sprung and unsprung bus mass (Fig. 5b).

With regard to the entire frequency range of excitation considered, the lowest values of the PSD of suspension deformation were ensured by the shock absorber with the highest damping coefficient of b_2=30000 Ns/m (Fig. 6b).

The amplitudes of dynamic wheel loads were decreased as the shock absorber damping coefficient was increased for excitation frequencies close to resonant frequencies of elastically sprung and unsprung bus mass. For the frequency range of excitation of
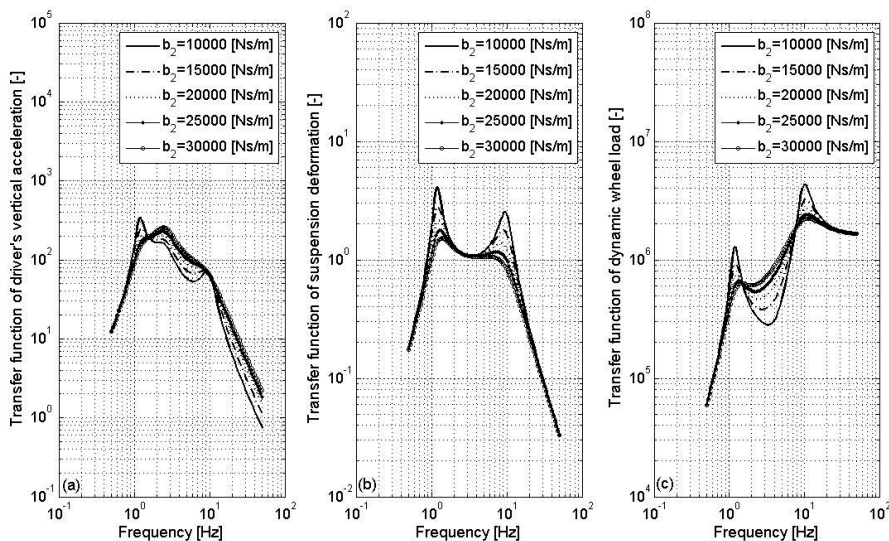
Fig. 5.

Transfer Functions of (a) Driver's Vertical Acceleration, (b) Suspension Deformation and (c) Dynamic Wheel Load for Different Values of Shock Absorber Damping b_2 at the Constant Value of Spring Stiffness c_2=300000 N/m

Fig. 6.

Power Spectral Density of (a) Driver's Vertical Acceleration, (b) Suspension Deformation and (c) Dynamic Wheel Load for Different Values of Shock Absorber Damping Coefficient b_2 at the Constant Value of Spring Stiffness c_2=300000 N/m

approximately 1.5 Hz to 9 Hz, dynamic wheel loads with the increase of shock absorber damping were increased (Fig. 5c).

For excitation frequencies close to resonant frequencies of the bus body and wheels, lower PSD of dynamic wheel load values, i.e. greater bus stability was provided by the shock absorber with a higher damping coefficient. The opposite applied for the frequency range 1.5 Hz to 9 Hz (Fig. 6c).

The summary of the analysis for the two excitation frequency ranges is given in Table 4.

4.1. The Analysis of Oscillatory Parameters for the Excitation Frequency Range 0.5Hz – 50Hz

The longitudinal roughness of pavement is of stochastic nature and has different wavelengths (i.e. frequencies) and amplitudes. The frequency range of excitation 0.5 Hz to 50 Hz has the greatest influence on oscillatory motions of vehicles (Kawamura and Kaku, 1985). The RMS values of the weighted driver's vertical acceleration and bus suspension deformation, as well as the coefficient of active safety for the given frequency range, were analysed.

The RMS value of acceleration reflects the vibrations intensity and is used to evaluate

their influence on comfort, individual's ability to work and its health. The RMS value of suspension deformation indicates the size of the space for wheels and suspension system elements, required for their proper function. The coefficient of active safety is the value suitable for the analysis of the contact between the tyre and the road surface, which is related to bus stability.

The RMS value of the weighted vertical acceleration of the driver is determined by means of the Eq. (9), according to (ISO 2631, 1978):

$$\ddot{z}_{w,eff} = \sqrt{\sum \left(w_i \ddot{z}_{i,eff} \right)^2} \qquad (9)$$

Where:

$\ddot{z}_{w,eff}$ - is the RMS of the weighted vertical driver acceleration (m/s2)

w_i - is the weight factor for the *i-th* central frequency of the one-third octave band excitation range (-)

$\ddot{z}_{i,eff}$ - is the RMS of acceleration for the *i-th* central frequency of the one-third octave band excitation range (m/s^2).

Table 4

Analysis of the Effect of Bus Suspension Parameters Change on the Three Considered Oscillatory Parameters

If the parameter value increase	Driver's vertical acceleration within resonant frequency area of		Suspension deformation within resonant frequency area of		Dynamic wheel load within resonant frequency area of	
	sprung mass	**unsprung mass**	**sprung mass**	**unsprung mass**	**sprung mass**	**unsprung mass**
c_2	increase	no effect	increase	decrease	increase	decrease
b_2	decrease	increase	decrease	decrease	decrease	decrease

Table 5

Comfort Criteria in the Means of Public Transport for Different Accelerations

Vibration intensity	Comfort estimation
<0.315 m/s²	not uncomfortable
0.315 -0.63 m/s²	a little uncomfortable
0.5 -1.0m/s²	fairly uncomfortable
0.8 -1.6 m/s²	uncomfortable
1.25 -2.5 m/s²	very uncomfortable
>2 m/s²	extremely uncomfortable

Source: ISO 2631 Standard

Table 5 shows the public transport users' estimate of comfort with regard to various intensities of overall vibrations, according to (ISO 2631, 1978).

The Eq. (10) represents the RMS value of bus suspension dynamic deformation:

$$\Delta_{(z_2-z_3)_{eff}} = \sqrt{\int_{f_1}^{f_2} \Phi_{(z_2-z_3)}(f)df} \qquad (10)$$

Where:

$\Delta_{(z_2-z_3)_{eff}}$ - is the RMS of suspension system dynamic deformation (m)

$\Phi_{(z_2-z_3)}(f)$ - is the PSD of suspension system deformation (m²/Hz) and

f_1 and f_2 - are the lower and upper limits of the excitation frequency range (Hz)

The coefficient of active safety - K represents the relationship between standard deviation of axle vertical force (the RMS of dynamic wheel load[2]) and static axle wheels load. Lower values of this parameter indicate better contact between the wheel and road surface, being the measure of better vehicle stability. The K parameter greater than 1 indicates the loss of the contact between the wheel and the road surface. The coefficient of active safety is calculated according to the Eq. (11):

$$K = \frac{\sigma_Z}{Z_{st}} = \frac{Z_{din,eff}}{Z_{st}} \qquad (11)$$

Where:

K – the coefficient of active safety (-)

$Z_{din,eff}$ - the RMS of dynamic wheel load (N)

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[2] The mean value of axle vertical force is normally equal to the static axle vertical force. It follows that the standard value of axle vertical force is equal to the RMS value of the variable (dynamic) part of axle vertical force.

Z_{st} - static wheel load (N).

The RMS value of dynamic wheel load is equal to (Eq. (12)):

$$Z_{din,eff} = \sqrt{\int_{f_1}^{f_2} \Phi_Z(f)df} \qquad (12)$$

Where:

$\Phi_Z(f)$ - is the PSD of dynamic wheel load (N²/Hz).

Figs. 7a-7c show changes in the RMS value of the weighted vertical driver acceleration, the RMS values of suspension deformation and the coefficient of active safety of the bus, as a function of suspension system spring stiffness and shock absorber damping at a constant bus speed of 100 km/h and for asphalt-concrete road surface in good condition.

Minimum values, maximum values and oscillatory parameters for the bus suspension system from Table 2 are shown in Figs. 7a-7c.

It should be noted that the most favourable oscillatory parameters were achieved with different values of the bus suspension oscillatory characteristics. Low values of the spring stiffness coefficient and low values of the shock absorber damping coefficient ensured the low RMS values of the weighted acceleration (lower than 0.315 m/s²-Fig. 7a), but also the highest effective values of suspension deformation (Fig. 7b). Low values of the coefficient of active safety were achieved for low values of spring stiffness and high values of shock absorber damping (Fig. 7c). On the other hand, for low values of spring stiffness and high values of shock absorber damping, the RMS values of the weighted vertical driver acceleration were higher than 0.315 m/s² (Fig. 7a). The requirement for good oscillatory comfort was conflicting with the requirements for the bus stability and the
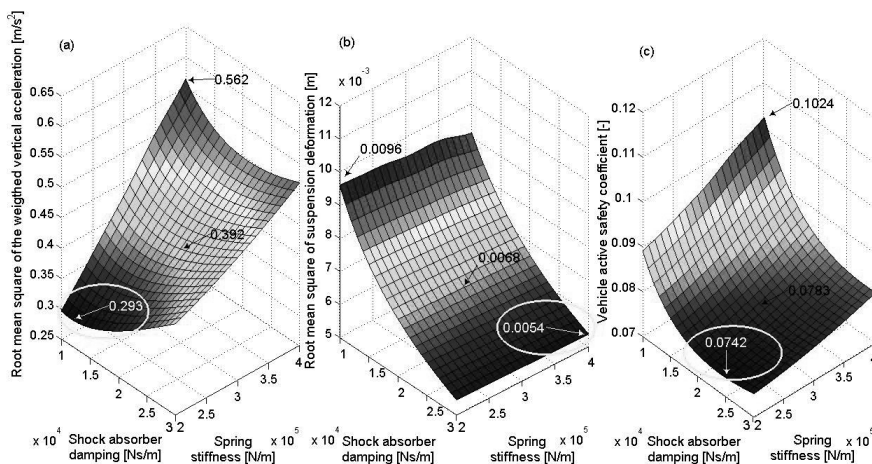
Fig. 7.

Change of (a) RMS Value of the Driver's Vertical Acceleration, (b) RMS of Bus Suspension Deformation and (c) Coefficient of Active Safety, as a Function of the Spring Stiffness Coefficient c_2 and the Shock Absorber Damping Coefficient b_2 for Constant Bus Speed at 100 km/h

available rattle space necessary for proper function of suspension system.

It could be observed that the RMS values of suspension deformation and the values of coefficient of active safety, determined by the simulation, were very low (the maximum RMS value of suspension deformation was approximately equal to 0.01 (m), and the maximum coefficient K value amounted to 0.1). This was expected, especially if one considers the type of bus excitation. Due to this the choice of spring stiffness and shock absorber damping can be made considering the most favourable RMS values of the weighted driver's acceleration that they provide. In Fig. 7a it can be seen that these are the softer springs and shock absorbers with low damping coefficient.

Fig. 8 shows the change in the RMS of the weighted acceleration as a function of bus speed for the spring stiffness and the shock absorber damping values that

correspond to the minimum and maximum RMS values of the driver's acceleration (c_2=200000 N/m, b_2=12000 Ns/m and c_2=400000 N/m, b_2=10000 Ns/m). Fig. 8 also shows the change in the RMS of weighted acceleration as a function of bus speed for the bus suspension oscillatory parameters from Table 1.

The RMS values of acceleration for the softest spring c_2=200000 N/m and shock absorber with damping b_2=12000 Ns/m were lower than 0.315 m/s² for the considered bus speed range of 60 km/h to 100 km/h. For the listed bus suspension oscillatory parameters, vibrations, according to the criteria from ISO 2631 Standard, have no influence on the driver's oscillatory comfort (Table 5). The RMS value of the driver's acceleration was, according to the simulation results, the highest in case of the suspension system with the stiffest spring (c_2=400000 N/m) and shock absorber with the lowest damping
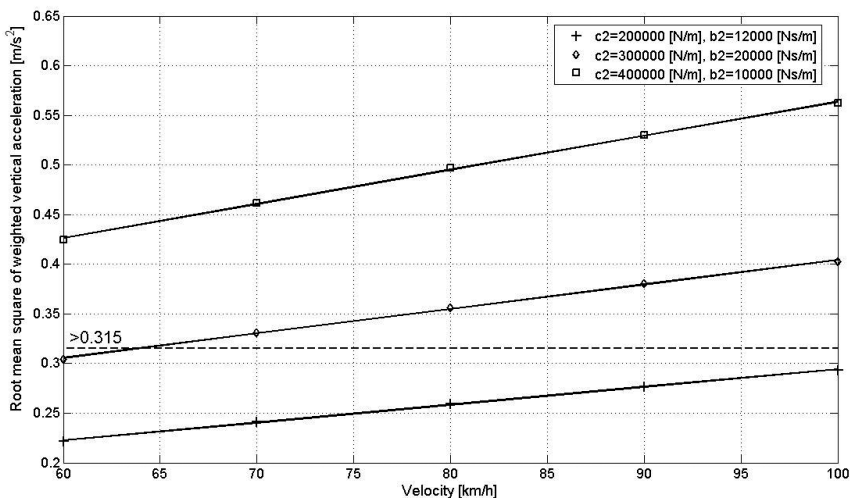
Fig. 8.
Change in the RMS Value of the Weighted Bus Driver Vertical Acceleration as a Function of Bus Speed

(b_2=10000 Ns/m). In this case, the effective acceleration was higher than 0.315 m/s², resulting the vibrations affect the driver's oscillatory comfort at all bus speeds. In the case of the suspension system with spring stiffness coefficient of c_2=300000 N/m and the shock absorber damping coefficient b_2=20000 Ns/m, vibrations affect the driver's oscillatory comfort at speeds over 65 km/h only.

5. Conclusion

The results of the analysis presented in this paper enabled us to consider the effects of different spring stiffness and different shock absorber damping on the vibration behaviour of the oscillatory bus model in the frequency domain. Three responses of the linear oscillatory bus model were analyzed (driver's vertical acceleration, suspension deformation and vertical dynamic wheel loads). It was found that the changes in the suspension oscillatory parameters have the following effect:

- The driver's riding comfort was decreased as bus suspension spring stiffness was increased for excitation frequencies close to resonant frequencies of the bus body (approximately 1.5 Hz).

- Suspension deformation was reduced as bus suspension spring stiffness was increased at excitation frequencies below 1 Hz. Within the zone of resonant frequency of sprung mass, the deformation amplitudes were increased as spring stiffness increased.

- A change in the PSD values of vertical wheel loads was particularly significant at lower excitation frequencies, about 1.5 Hz. Softer springs provided better

contact between the wheel and the road surface, i.e. greater bus stability.

Higher shock absorber damping values provided better oscillatory comfort for the driver at excitation frequencies close to the resonant frequency of the bus body. At excitation frequencies above 1.5 Hz, the shock absorber with lower damping coefficient values ensured greater oscillatory comfort.

With regard to the entire frequency range of excitation considered, the lowest PSD values of suspension deformation were provided by the shock absorber with the highest damping coefficient of b_2=30000 Ns/m.

The amplitudes of dynamic wheel load were decreased as the values of the shock absorber damping coefficient were increased, at excitation frequencies approximating resonant frequencies of sprung and unsprung bus masses (approximately at 1.5 Hz and 10 Hz).

The RMS values of suspension deformation and the values of the coefficient of active safety determined by the simulation, for the excitation from good asphalt-concrete pavement, were very low. Consequently, the choice of spring stiffness and shock absorber damping should be made according to the most favourable values of the weighted driver's acceleration.

Acknowledgements

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UTICAJ KRUTOSTI I PRIGUŠENJA ELEMENATA SISTEMA ZA OSLANJANJE NA OPTIMIZACIJU OSCILATORNOG PONAŠANJA AUTOBUSA

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Sažetak: U cilju definisanja preporuka za izbor oscilatornih parametara prilikom konstruisanja sistema za oslanjanje (međugradskog) autobusa u radu je izvršena analiza uticaja krutosti opruge i prigušenja amortizera na vertikalno ubrzanje tela vozača, deformaciju sistema za oslanjanje i vertikalne dinamičke reakcije tla. Oscilatorni odzivi analizirani su pomoću linijskog oscilatornog modela autobusa sa linearnim karakteristikama i tri stepena slobode, sa pobudom pomoću spektralne gustine neravnosti asfalt-betonskog kolovoza u dobrom stanju. Analiza je sprovedena simulacijom, u frekventnom domenu, pomoću jednačina statističke dinamike. Za analizu prenosnih funkcija, spektralnih gustina i efektivnih vrednosti oscilatornih veličina korišćen je program napisan u programskom paketu Matlab. Rezultati sprovedene analize pokazuju da parametri koji obezbeđuju dobar oscilatorni komfor vozača nisu u saglasnosti sa parametrima koji omogućuju najveću stabilnost autobusa i odgovarajući radni hod točka. Sa aspekta oscilatornog komfora vozača sistem oslanjanja autobusa treba da ima oprugu male krutosti i amortizer sa niskim koeficijentom prigušenja, sa aspekta aktivne bezbednosti oprugu male krutosti i amortizer sa visokim koeficijentom prigušenja, dok minimalni radni hodovi točka traže oprugu velike krutosti i amortizer sa visokim koeficijentom prigušenja.

Ključne reči: autobus, oscilatorno ponašanje, opruga, amortizer, simulacija.