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Vibration analysis of bus under random road conditions

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ABSTRACT:Vehicle vibration analysis for optimal vehicle design is one of the important issues for vehicle designers and manufacturers. To analyze bus vibrations, a three-dimensional vibration model of bus with 10 DOF (degree of freedom) based on Dragan Sekulić modelis establish underrandom road conditions. The differential equations describing the motion of the mechanical system are written by usinga combined method of the multi-body system theory and D'Alembert's principle. The different operating conditionsare analysed their influence on the vertical acceleration responses of the space of a driver, passenger in the middle part of the bus and passenger in the rear overhang. The obtained results show thatthe road surface conditions have a great influence on vehicle vibration.

KEYWORDS: Bus, Dynamic model, Vehicle vibration; Random road condition

I. INTRODUCTION

Bus vibrations not only affect the durability of the vehicle's parts, but also the vehicle ride comfort of the movement and the health of the driver and passengers. Vibration comfort is an important factor affecting the quality of service of bus. A full-vehicle vibration model with 10 DOF for 2-axle bus considering air suspension systems and seat suspension systems was established based on Dragan Sekulić modelto analyse the influences of vehicle suspension parameters such as stiffness and damping coefficients on seat's driver/passenger comfort [1]. Vehicle dynamics model in type of 1/4 was used for vibration analysis under the effect of random road profile with different grades to evaluate the influence of the air spring stiffness and road surface conditions on the comfort of the vehicle, as the basis for changing the air spring stiffness in accordance with adjusting the pressure of it according to the type of road profile quality and the mathematical model to describe the used random road profile was able to change the type of investigated road grades by selecting the corresponding power spectral density parameter of the road grade according to the ISO 8608 standard [2].The effects of vibrations on the comfort of intercity bus IK-301 users were analyzed and evaluatedaccording to the criteria set out in the 1997 ISO 2631-1 standard for comfort in public means of transport [3]. A validated nonlinearoscillatory 16 degrees of freedom (DOF) bus model defined in the software Matlab/Simulink and procedure from ISO 2631/1997 standard was proposed to analyze the effect of the waviness parameter won the oscillatory comfort of bus passengers for one road in good condition [4].In order to make people involved in supervising bus's vibration comfort and improve passengers' riding experience, a novel mode of passenger crowdsourcing is introduced. The comfort degree of bus vibration was calculated from bus's vibration signals collected by passengers' smartphones and sent through WiFi to the Boa web server which shows the vibration comfort on the LCD deployed in bus and maybe trigger alarm lamp when the vibration was beyond the threshold [5]. A 10-Degree-Of-Freedom (10-DOF) model of an intercity bus vehicle under harmonic and random excitations caused by road roughnesswas proposed to suppress undesirable vibrations, mass-spring damper passive absorbers. Finally, the optimization of the characteristics of embedded passive absorbers under each seat and implementation of the designed absorbers was proceed using a 13-DOF dynamic model of the intercity bus including the three passivemass-spring-damper shock absorbers embedded under each of the three seats [6].The major goal of this study is to establish a mathematical model of the bus vibrationsunder the road surface roughness excitationsto analyzetheir influence on the vertical acceleration responses of the space of a driver, passenger in the middle part of the bus and passenger in the rear overhang.

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II. MATHEMATICAL MODELING OF BUS

A full-vehicle dynamic model: A bus IK-301 with the dependent suspension systems including an air bag and two air bags for front and rear suspension systems was selected for establishing vehicle vibration model. To study the dynamics and vibrations of the bus, a 10-DOF model is used in the simulations according to the mathematical model proposed by Dragan Sekulić [3] to analyze the seat comfort of bus IK-301, as shown in Fig. 1.

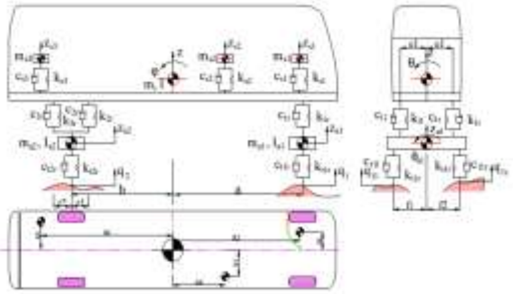

Figure 1.A full-vehicle dynamic model [1]

Explain the symbols inFigure 1: $k_{ij}$are the stiffness of the suspension systems, respectively; $c_{ij}$is the damping coefficient of the suspension system, respectively; $k_{tnj}$are the stiffness of the tires respectively; $c_{tnj}$are the damping coefficients of the tires, respectively; m is the mass of vehicle body; k_{si} and $c_{si}$are the stiffness and damping coefficients of the driver's seat suspension systems, respectively, $m_{an}$are the masses of the front and rear axles, respectively; m_{si} are the masses ofthe space of a driver, passenger in the middle part of the bus and passenger in the rear overhang; a, b, s_n, r_n,f_n, e_n are the distances, respectivel; z_{an}, z and $z_{s1}$is the vertical displacement at the center of gravity of the axle, vehicle body and driver's seat, respectively; φ, θ and$\theta_{an}$are the angular displacements at the center of gravity of the elastic-suspended mass of the fully loaded bus and axles, respectively; I and Ianare the moments of inertia of the elastic-suspended mass of the fully loaded bus and axles, respectively; q_{nj} are the random road surface profiles; v is the speed of vehicle; v isthe velocity of the vehicle (i=1,2,3; n=1,2; j=l, r).

The equations of vehicle motion: The equations of vehicle motion can be formulated in different ways such as Lagrange's equation, Newton-Euler equation, Jourdain's principle. However, in order to facilitate the description of vehicle dynamic systems using computer simulation, a combined method of the multi-body system theory and D'Alembert's principle is chosen in this study. The multi-body system theory is used to separate the system into subsystems which are linked by the force and moment equations. D'Alembert's principle is used to set up force and moment equations to describe vehicle dynamic subsystems. The equations of motion of Figure 1 can be written as Eq.(1).

Road surface excitations:Many research results reveal that the term "ride quality" is used to describe vehicle vibrations in the frequency range of about 0-25Hz [1-5]. Vibrations caused by the vehicle itself, such as driveline or engine vibrations, are generally of higher frequency and are hence associated with noise rather than ride comfort.

$$\begin{cases}
m_{s1}\ddot{z}_{s1} = -\left[k_{s1}(z_{s1}-z_{01})+c_{s1}(\dot{z}_{s1}-\dot{z}_{01})\right]\\
m_{s2}\ddot{z}_{s2} = -\left[k_{s2}(z_{s2}-z_{02})+c_{s2}(\dot{z}_{s2}-\dot{z}_{02})\right]\\
m_{s3}\ddot{z}_{s3} = -\left[k_{s3}(z_{s3}-z_{03})+c_{s3}(\dot{z}_{s3}-\dot{z}_{03})\right]\\
m_{a1}\ddot{z}_{a1} = \left[\mathrm{k}_{1l}(z_{1l}-z_{a1l})+c_{1l}(\dot{z}_{1l}-\dot{z}_{a1l})+\mathrm{k}_{1r}(z_{1r}-z_{a1r})+c_{1r}(\dot{z}_{1r}-\dot{z}_{a1r})\right]-\\
-\left[\mathrm{k}_{t1l}(z_{t1l}-q_{1l})+c_{t1l}(\dot{z}_{t1l}-\dot{q}_{1l})+\mathrm{k}_{t1r}(z_{t1r}-q_{1r})+c_{t1r}(\dot{z}_{a1r}-\dot{q}_{1r})\right]\\
I_{a1}\ddot{\theta}_{a1} = \left[\mathrm{k}_{t1l}(z_{t1l}-q_{1l})+c_{t1l}(\dot{z}_{t1l}-\dot{q}_{1l})\right].f_1+\left[\mathrm{k}_{1r}(z_{1r}-z_{a1r})+c_{1r}(\dot{z}_{1r}-\dot{z}_{a1r})\right].e_1-\\
-\left[\mathrm{k}_{t1r}(z_{t1r}-q_{1r})+c_{t1r}(\dot{z}_{t1r}-\dot{q}_{1r})\right].f_1-\left[\mathrm{k}_{1l}(z_{1l}-z_{a1l})+c_{1l}(\dot{z}_{1l}-\dot{z}_{a1l})\right].e_1\\
m_{a2}\ddot{z}_{a2} = \mathrm{k}_{23r}(z_{2r}-z_{a2r})+c_{23l}(\dot{z}_{2r}-\dot{z}_{a2r})+\mathrm{k}_{23l}(z_{2l}-z_{a2l})+c_{23l}(\dot{z}_{2l}-\dot{z}_{a2l})-\\
-\mathrm{k}_{t2l}(z_{t2l}-q_{2l})-c_{t2l}(\dot{z}_{t2l}-\dot{q}_{2l})-\mathrm{k}_{t2r}(z_{t2r}-q_{2r})-c_{t2r}(\dot{z}_{t2r}-\dot{q}_{2r})\\
I_{a2}\ddot{\theta}_{a2} = \left[\mathrm{k}_{23l}(z_{23l}-z_{a2l})+c_{23l}(\dot{z}_{23l}-\dot{z}_{a2l})-\mathrm{k}_{23r}(z_{23r}-z_{a2r})-c_{23r}(\dot{z}_{23r}-\dot{z}_{a2r})\right].e_2\\
-\left[\mathrm{k}_{t2l}(z_{t2l}-q_{2l})+c_{t2l}(\dot{z}_{t2l}-\dot{q}_{2l})-\mathrm{k}_{t2r}(z_{t2r}-q_{2r})-c_{t2r}(\dot{z}_{t2r}-\dot{q}_{2r})\right].f_2\\
m\ddot{z} = \left[k_{s1}(z_{s1}-z_{01})+c_{s1}(\dot{z}_{s1}-\dot{z}_{01})\right]-\mathrm{k}_{1l}(z_{1l}-z_{a1l})-c_{1l}(\dot{z}_{1l}-\dot{z}_{a1l})-\\
-\mathrm{k}_{1r}(z_{1r}-z_{a1r})-c_{1r}(\dot{z}_{1r}-\dot{z}_{a1r})-\mathrm{k}_{23l}(z_{23l}-z_{a2l})-c_{23l}(\dot{z}_{23l}-\dot{z}_{a2l})-\\
-\mathrm{k}_{23r}(z_{23r}-z_{a2r})+c_{23r}(\dot{z}_{23r}-\dot{z}_{a2r})\\
I_1\ddot{\varphi} = \left[\mathrm{k}_{1l}(z_{1l}-z_{a1l})+c_{1l}(\dot{z}_{1l}-\dot{z}_{a1l})\mathrm{k}_{1r}(z_{1r}-z_{a1r})+c_{1r}(\dot{z}_{1r}-\dot{z}_{a1r})\right].a-\\
-\left[\mathrm{k}_{23l}(z_{23l}-z_{a2l})+c_{23l}(\dot{z}_{23l}-\dot{z}_{a2l})-\mathrm{k}_{23r}(z_{23r}-z_{a2r})+c_{23r}(\dot{z}_{23r}-\dot{z}_{a2r})\right].b-\\
-\left[k_{s1}(z_{s1}-z_{01})+c_{s1}(\dot{z}_{s1}-\dot{z}_{01})\right]\\
I_2\ddot{\theta} = -\left[k_{s1}(z_{s1}-z_{01})+c_{s1}(\dot{z}_{s1}-\dot{z}_{01})\right].\mathrm{s}_1+\\
+\left[\mathrm{k}_{1l}(z_{1l}-z_{a1l})+c_{1l}(\dot{z}_{1l}-\dot{z}_{a1l})\right].e_1+\left[\mathrm{k}_{23l}(z_{23l}-z_{a2l})+c_{23l}(\dot{z}_{23l}-\dot{z}_{a2l})\right].e_2-\\
-\left[\mathrm{k}_{23r}(z_{23r}-z_{a2r})+c_{23r}(\dot{z}_{23r}-\dot{z}_{a2r})\right].e_2+\left[\mathrm{k}_{1r}(z_{1r}-z_{a1r})+c_{1r}(\dot{z}_{1r}-\dot{z}_{a1r})\right].e_1
\end{cases} \quad (1)$$

Low frequency vibrations in vehicles are usually caused by road surface roughness which plays an important role in analyzing driver ride comfort. It is assumed to be a zero-mean stationary Gaussian random process [7] which can be generated through an inverse Fourier transformation

$$q(t)=\sum_{i=1}^{N}\sqrt{\frac{2vn_0^2G_q(n_0)}{f_{mid,i}^2}\Delta f}.\cos\left(2\pi f_{mid,i}t+\varphi_i\right) \quad (2)$$

where, $G_q(n_0)$ is the road roughness coefficient which is defined for typical road classes from A ($G_q(n_0)$ =4x10^{-6} m^3) to H ($G_q(n_0)$ = 16384x10^{-6} m^3) according to ISO 8608(1995) [7], n_0 is a reference spatial frequency; v is the speed of vehicle; φ_i is an uniform probabilistic distribution within the 0-2π range; $f_{mid,i}$ is the temporal frequencies of the road roughness which is defined Eq.(2).

III. RESULTS AND ANALYSIS

MatLab software and Simulink toolbox are selected to solve Eq. (1) with the vehicle simulation parameters [5, 8] under the random road surface excitation.The vertical acceleration responses of the space of a driver, passenger in the middle part of the

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bus and passenger in the rear overhangwhen vehicle moves on the ISO good road (grade B) condition at the vehicle speed of 80 km/h with full load is shown in Figure 2 to Figure 4. From the results from Figure 2 to Figure 3, we show that the peak values of a_{s1}, a_{s2} and a_{s3}gradually increase which leads to a decrease in bus ride comfort. Thus, different seat positions affect the vibrations of the differentseats.
The vertical acceleration responses of the space of a driver, passenger in the middle part of the bus and passenger in the rear overhangwhen vehicle moves on the ISO average road (grade C) condition at the vehicle speed of 60 km/h with full load is shown in Figure 5 to Figure 6.Similarity,from the results from Figure 5 to Figure 6, we show that the peak values of a_{s1}, a_{s2} and a_{s3} gradually increase which leads to a decrease in bus ride comfort. However, the influence of the road surface conditions are very large on the peak amplitude valueswhich leads to a decrease in bus ride comfort.

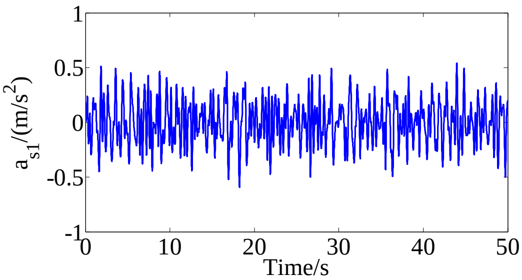

Figure 2. The vertical acceleration responses of the space of a driverwhen vehicle moves on the ISO good road (grade B) condition at the vehicle speed of 80 km/h with full load

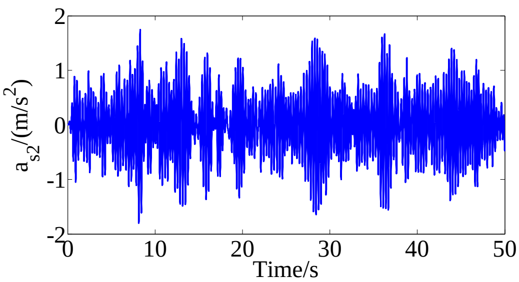

Figure 3. The vertical acceleration responses of passenger in the middle partwhen vehicle moves on the ISO good road (grade B) condition at the vehicle speed of 80 km/h with full load

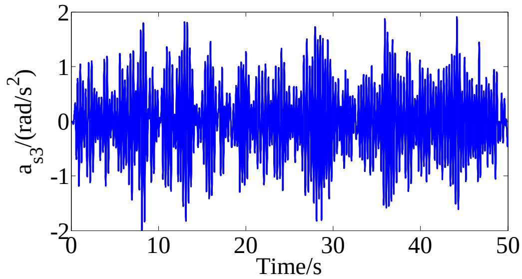

Figure 4. The vertical acceleration responses of passenger in the rear overhangwhen vehicle moves on the ISO good road (grade B) condition at the vehicle speed of 80 km/h with full load

From the survey results, we know that the peak amplitude values of a_{s1}, a_{s2} and a_{s3} gradually increase when the vehiclespeed reduces we reduce the speed of vehicle.

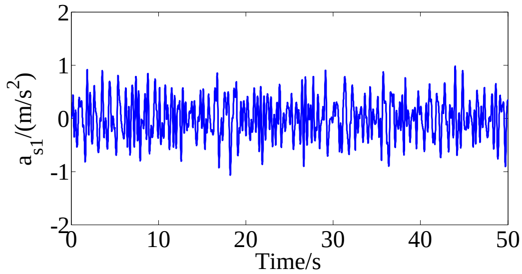

Figure 5. The vertical acceleration responses of the space of a driver when vehicle moves on the ISO averageroad (grade C) condition at the vehicle speed of 60 km/h with full load

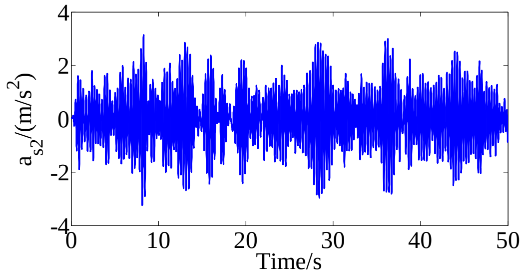

Figure 6. The vertical acceleration responses of passenger in the middle part when vehicle moves on the ISO average road (grade C) condition at the vehicle speed of 60 km/h with full load

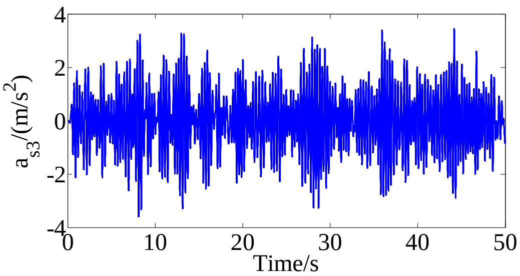

Figure 7. The vertical acceleration responses of passenger in the rear overhang when vehicle moves on the ISO average road (grade C) condition at the vehicle speed of 60 km/h with full load

IV.CONCLUSION

In this study, a three-dimensional vibration model of bus with 10 DOF (degree of freedom) based on Dragan Sekulić model is establish underrandom road conditions toanalyze bus vibrations. The differential equations describing the motion of the mechanical system are written by using a combined method of the multi-body system theory and D'Alembert's principle. The major conclusions that can be drawn from the analysis can be summarized as follows: (1)the peak values of a_{s1}, a_{s2} and a_{s3} gradually increase which leads to a decrease in bus ride comfort; and (2) the peak amplitude values of a_{s1}, a_{s2} and a_{s3} gradually increase when vehicle moves on a bad road, we reduce the speed of vehicle; (3) the different seat positions affect the vibrations of the differentseats.

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