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Modeling and simulating of a passenger car vibrations

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ABSTRACT: *Vehicle vibrations have a great impact on the comfort and health of passengers. The goal of this paper is to propose a full dynamic model of a passenger car with 7 degrees of freedom under random road roughness. The mathematical model is set up and solved by Matlab/simulink software. The research results have built a vibration model of a car passenger and simulated vehicle body vibrations. Especially, these research results are the theoretical basis for researchers to improve the suspension system to improve vehicle ride comfort.*

KEYWORDS: *Car passenger, Vibration, Model, Simulation.*

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I. INTRODUCTION

The influence of the secondary suspension model on the vertical vibration behavior of vehicle body was discussed based on the results of numerical simulations on the frequency response functions of the acceleration, the power spectral density of the acceleration and the root mean square of the acceleration of the car body [1]. A simplified modeling approach to predict the vibration response, about an operating point, of mannequin occupied car seats was demonstrated to be feasible based on previously developed tools for car seat designers [2]. The development of vertical vibration simulation for a seated passenger in a moving vehicle was presented to be resulting from the bounce effect of the vehicle under various conditions [3]. The tools for vibration evaluation of power train mounting system of a passenger car were recommended to meet demands for passenger's comfort based on the Multi-Body System principles and includes submodules as were engine, clutch, gearbox, final drive, drive shafts and rheological models of all power train mountings [4]. The passenger body vibration control using an Adaptive Neuro Fuzzy Inference System (ANFIS) based super twisting sliding mode controller (ASTSMC) in active quarter car system was recommended based on quarter car model with three degrees of freedom composed of passenger body, sprung mass and unsprung mass [5].

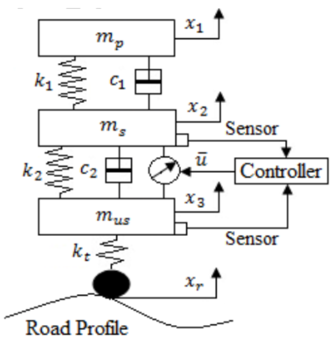

Fig.1. Active quarter dynamic model [5]

A quarter car model with passenger body under random road excitation was considered to capture the dynamic behaviour of a real car system to improve the passenger ride comfort and safety in an active quarter car model and three different controllers such as ANFIS controller, Hybrid ANFIS PID controller (HANFISPID) and Hybrid ANFIS PID Controller with Coupled Rules (HANFISPIDCR) were designed [6]. A four degrees of freedom biodynamic human body model and a three degrees of freedom quarter car model were used for ride comfort analysis. A super-twisting sliding mode control (STSMC) and adaptive neuro-fuzzy inference system (ANFIS) based super-twisting sliding mode control (ASTSMC) strategy was used in the main suspension of the active quarter car model [7]. The effectiveness of an Adaptive Neuro-Fuzzy Inference System (ANFIS)-based semi-active controlled suspension system using a magnetorheological fluid damper in reducing nonlinear lateral vibrations of a passenger rail vehicle was recommended to investigate the performance of the suspension system in attenuating vibrations induced by track irregularities [8]. The aim of this paper is to establish a full dynamic model of a passenger car with 7 degrees of freedom under random road roughness, and then we it is simulated in matlab/simulink software environment.

II. VEHICLE DYNAMIC MODEL

A full dynamic model of a passenger car is shown in Fig.2.

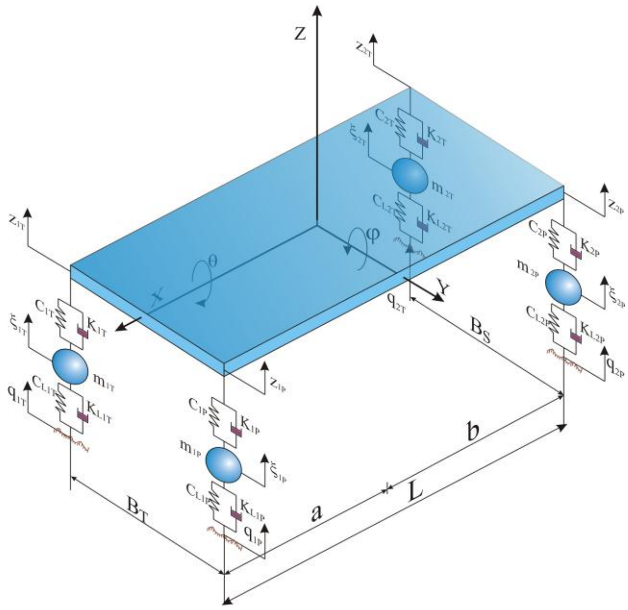

Fig.2 Full dynamic model of a passenger car

Explanation of the symbols for Fig. 2, C_{ij} are the stiffness of suspension systems; K_{ij} are the damping coefficients of the suspension systems; C_{Lij} are the stiffness of of the tires; K_{Lij} are the damping coefficients of the tires; ξ_{ij} are the displacement of front and rear axle masses in the vertical direction; z, φ, θ are the vertical, pitching and rolling displacements of the vehicle body; q_{ij} are the excitation of road surface roughness; a, b, L, B_t and B_s are the basic dimensions of the vehicle; m_{ij} and M are axle and vehicle body (i=1,2 and j=P,T).

The equations of vehicle motion can be formulated in different ways such as Lagrange's equation, Newton-Euler equation, Jourdain's principle. In this study, Newton-Euler equation is chosen to describe the equations of vertical motion of electric vehicle. From Full dynamic model of a passenger car in Fig. 2, the general equation is written using the matrix to be written as

$$[M]\ddot{z}+[K]\dot{z}+[C]z=[F(t)] \quad (1)$$

where, [M], [K], [C] are mass, damping coefficient and stiffness matrices and [F(t)] is is the excitation force vector

Road surface excitation: It is described by various mathematical functions. In this study, the random road surface function is selected as the input function of the vehicle dynamic analysis problem. The random road surface roughness of random white noise is used as excitation source waveform for vehicle [9], the random road profile is produced by filtering the white noise using the following mathematical model of the road roughness.

$$\dot{q}(t)+2\pi f_0 q(t)=2\pi n_0\sqrt{G_q(n_0)v(t)}w(t) \qquad (2)$$

where, $G_q(n_0)$ is the road roughness coefficient which is defined for typical road classes from A to H according to ISO 8068(1995) [10], n_0 is a reference spatial frequency which is equal to 0.1 m, v(t) is the speed of vehicle; f_0 is a minimal boundary frequency with a value of 0.0628 Hz, n_0 is a reference spatial frequency which is equal to 0.1 m, w(t) is a white noise signal.

III. RESULTS AND DISCUSSION

In order to simulate vibrations of a passenger car, the general equation of Fig.2 are simulated under random road roughness by the MATLAB/Simulink with design parameters of a passenger car in reference [12]. The time domain acceleration responses of vertical vehicle body (a_{bz}), pitching vehicle body angle (a_{bphi}) and rolling vehicle body angle (a_{bteta}) when the vehicle moves on good surface condition (ISO class B) at v=72 km/h are shown Fig. 3, Fig.4, and Fig.5.

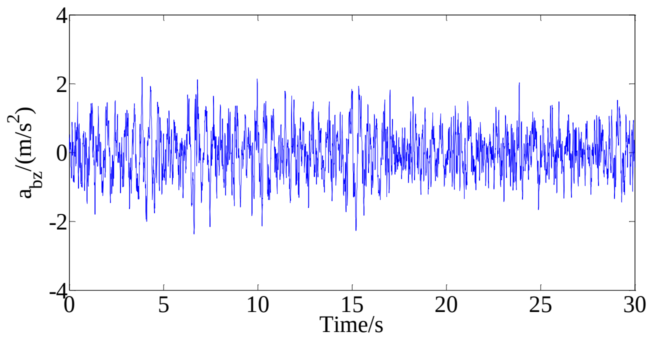

Fig.3 Time domain acceleration response of vertical vehicle body (a_{bz})

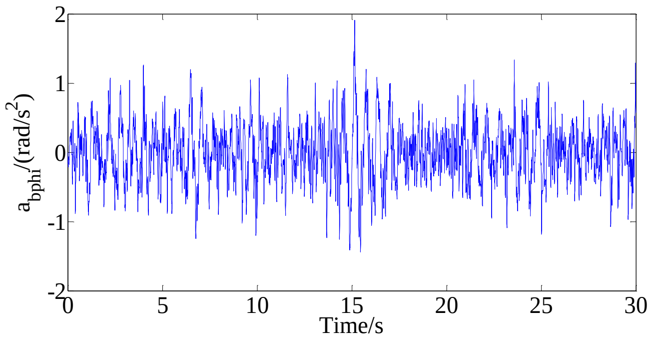

Fig.4 Time domain acceleration response of pitching vehicle body angle (a_{bphi})

From the obtained results of Fig.3, we show that the peak maximum amplitude value of a_{bzmax}= 2.1915 m/s^2. The obtained results of Fig.4, we show that the peak maximum amplitude value of $a_{bphimax}$= 1.9092 rad/s^2. The obtained results of Fig.5, we show that the peak maximum amplitude value of $a_{btetamax}$= 0.0926 rad/s^2. From these values, we only have the largest peak amplitude as a basis for evaluating vehicle ride comfort according to ISO standards ISO 2631-1 [11].

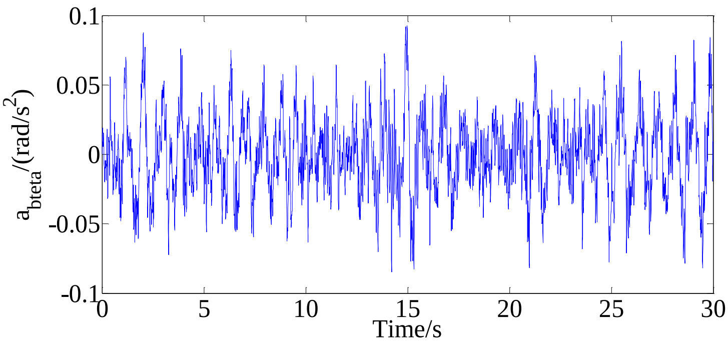

Fig.5 Time domain acceleration response of rolling vehicle body angle (a_{bteta})

IV. CONCLUSION

This study is to analyze vibrations of a passenger car, a full dynamic model of a passenger car with 7 degrees of freedom are established to simulate under random road roughness. The conclusions could be drawn: The peak maximum amplitude values of a_{bzmax}= 2.1915 m/s^2, $a_{bphimax}$= 1.9092 rad/s^2 and $a_{btetamax}$= 0.0926 rad/s^2. In addition, the results of this paper are the basis for analyzing the effects of design parameters of the suspension system of a passenger car on vehicle ride comfort which will be published by our research team.

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