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Reduction of low frequency vibration of truck driver and seating system through system parameter identification, sensitivity analysis and active control

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ABSTRACT

This paper aims to develop an 5-degree-of-freedom driver and seating system model for optimal vibration control. A new method for identification of the driver seating system parameters from experimental vibration measurement has been developed. The parameter sensitivity analysis has been conducted considering the random excitation frequency and system parameter uncertainty. The most and least sensitive system parameters for the transmissibility ratio have been identified. The optimised PID controllers have been developed to reduce the driver's body vibration.

1. Introduction

Commercial vehicles such as trucks, buses produce vibration when they are driven. Passengers may feel uncomfortable and even suffer injuries [1,2] when the vertical vibration is large and its frequency falls within the frequency range of 4–8 Hz [2]. A thorough study on dynamic comfort [3] of vehicle occupants can help improve the riding experience of the passengers. All of the literatures on vehicle dynamic systems considering the body model as a lumped parameter model presented the model in the frequency domain as transfer functions or in the time domain as the differential equation of motions, which is analogous to a mechanical system of springs, masses and dampers. Although there has been an increase in the recent volume of research works on non-lumped parameter models, to date there has been a lack of research applying the non-lumped parameter models to seating suspension systems, possibly due to the increased complexity and efficiency of integrating them with control systems. A majority of lumped parameter vehicle body motion modeling studies is focused exclusively on the vertical motion [11]. In the research published by so far, a majority of lumped parameter models are purely prismatic and consist of a single dimension, that is, they do not take into account the effects of rotation of any elements within the system. This means that the springs and dampers are not able to rotate.

Pandurangan et al. [8] adopted a single degree of freedom dynamic system to simulate floor-seat configuration. A two degrees of freedom system that considered the stiffness and damping characteristics of the soft foam seat cushion, was also used for refining the design. The stiffness and damping parameters for the passive isolation system were determined from the measurement results and isolation requirement.

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Du et al. [7] studied an integrated 3DOF seating suspension model, which included a quarter-car suspension. A seating suspension with a driver body model is used to design a sub-optimal $H\infty$ controller for an active isolator. The desired control force generated by this active isolator is then generated by the MRE isolator through its continuously variable stiffness property when the actuating condition is met. However, the parameters that describe an optimized quarter-car suspension and seating suspension model with a driver body were obtained from the literatures.

Existing research into analysis of the vibration response has modelled the human body using lumped parameter models with varying configurations and DOFs ranging from 1 up to 12 [3,15]. Zhang et al. [9] proposed a 17DOF bidirectional linear model in order to improve the driving and ride comfort of ground vehicles. However, as a matter of fact, increasing degrees of freedom of the models is of little help to improve the model accuracy. Instead, it is more important to accurately describe the dynamic responses of each part and individual organs of seated occupants [10,11]. Liang and Chiang [12] compared and evaluated 12 types of lumped-parameter models of seated occupants with different degrees of freedom, and recommended the 4DOF series-to-parallel model proposed by Wan and Schimmels [6]. Up to date, the 4DOF models of human body are most popular. It has been shown that in general, the 4-DOF model provides more accurate estimation of the human body response to vibrations, than do the higher DOF models. This may be due to the over fitting of these higher DOF models [3]. 4DOF models have appropriate number of parameters and provide reasonable fitting performance. Meanwhile, it is convenient to realize model extension on the basis of 4DOF models. To date, series models such as Wan and Schimmels' model [6], Boileau and Rakheja's model [4], and Zhang et al.'s model [5] and series-to-parallel such as Liu et al's model [13] and Singh and Wereley's model [14] have been developed. Although these 4DOF series models and series-to-parallel models can be used, the systematic parameter identification method from experimental measurement results has not been developed for the model extension including the seat mass and its suspension stiffness and damping coefficients.

The major objectives of this research are to identify the lumped mass-spring-dashpot model parameters of the 5DOF combined driver and seating system of a truck for vibration control design and simulation. First of all, the vibration response is measured from a running vehicle at idle in park with air condition off, then, a vibration simulation model of the driver and seating system in the vertical direction is developed. The system parameters are identified where the mass and stiffness coefficients are adjusted until the calculated resonant frequencies coincide with the measured resonant frequencies, and damping coefficients are adjusted until the calculated frequency response amplitude peak values coincide with the measured frequency response amplitude peak values. The sensitivity analysis of the seating suspension system parameters has been conducted considering the random excitation frequency and the seating suspension system parameter uncertainty. The simulation model of the driver and seating system with the identified parameters is reduced from five degrees of freedoms to a single degree of freedom using the balanced modal reduction method. Primary and secondary PID controllers are designed for reducing vibration of the driver and seating system. In this paper, it is assumed that system parameters do not change with operation conditions and drivers. For example, the changes of the system parameters caused by variation of drivers and driver's postures are not considered in this paper. It is also assumed that the human body and seating system is a linear system. For a nonlinear human body seating system of variable parameters, a PID control is not applicable, a Fuzzy-PID control has to be adopted, which will be studied in our future work.

2. Mathematic and simulation model

In order to reduce vibration of a seating system, an analytical biodynamic model of a driver and seating system is developed. The analytical biodynamic model includes a seating suspension and seated driver which is described using the lumped mass-spring-dashpot parameter model as shown in Fig. 1. The seat mass denoted by M_1 is fixed to the floor through the seating suspension, which consists of a coil spring and a damper and is modelled by spring K_1, damping C_1 and the floor excitation displacement z_0. The soft foam seat cushion made from polyurethane foam is configured on the seat and, since its mass is small, its mechanical model is simply represented by spring K_{2c} and damping C_{2c}. In this study, the driver sits upright on the soft foam seat cushion on the seat and consists of four parts: pelvis, upper torso, viscera, and head. Four parts of the driver body are modelled by a linear lumped mass-spring-dashpot parameter system comprising mass M_i, spring K_i and damping C_i for i = 2, 3, 4, and 5. x_i (i = 2, 3, 4, 5) coordinates are the displacement of the driver body for pelvis, upper torso, viscera and head, respectively. Note that when a driver sits on a seat, only 71 per cent of the total weight of the driver is supported by the seat, and the remaining body weight is supported by the feet [2]. Therefore, in this driver body model, the pelvis includes femur, but excludes the lower legs and feet. In addition, z_0 and $\dot{z}_0$ are the displacement and velocity of the floor vibration due to the running engine or uneven road excitation during transport.

The equations of the motion of the five degrees of freedom driver and seating suspension system are governed by

$$\begin{cases} M_1\cdot\ddot{x}_1 + C_1\cdot(\dot{x}_1-\dot{z}_0) + K_1\cdot(x_1-z_0) + \frac{C_2\cdot C_{2c}}{C_2+C_{2c}}\cdot(\dot{x}_1-\dot{x}_2) + \frac{K_2\cdot K_{2c}}{K_2+K_{2c}}\cdot(x_1-x_2) = 0 \\ M_2\cdot\ddot{x}_2 + C_3\cdot(\dot{x}_2-\dot{x}_3) + K_3\cdot(x_2-x_3) + \frac{C_2\cdot C_{2c}}{C_2+C_{2c}}\cdot(\dot{x}_2-\dot{x}_1) + \frac{K_2\cdot K_{2c}}{K_2+K_{2c}}\cdot(x_2-x_1) = 0 \\ M_3\cdot\ddot{x}_3 + C_5\cdot(\dot{x}_3-\dot{x}_5) + K_5(x_3-x_5) + C_3\cdot(\dot{x}_3-\dot{x}_2) + K_3(x_3-x_2) + C_4\cdot(\dot{x}_3-\dot{x}_4) + K_4(x_3-x_4) = 0 \\ M_4\cdot\ddot{x}_4 + C_4\cdot(\dot{x}_4-\dot{x}_3) + K_4(x_4-x_3) = 0 \\ M_5\cdot\ddot{x}_5 + C_5\cdot(\dot{x}_5-\dot{x}_3) + K_5(x_5-x_3) = 0 \end{cases} \tag{1}$$