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Vibration analysis of double-drum vibratory vibrations Part 3: Effects of design parameters of driver's seat suspension system

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ABSTRACT: *In order to analyze effects of design parameters of driver seat suspension system on ride comfort of a double-drum vibratory roller, based on the physical model and mathematical model presented in part 1 of this study. The design parameters of driver's seat suspension system (SSS) such as stiffness and damping coefficients are analyzed for their influence on the vehicle's ride comfort according to the ISO 2631:1997(E). The study results have shown that the design parameters of driver's SSS have a direct impact on vehicle ride comfort. In addition, the study is a theoretical basis for researchers to improve the vehicle's ride comfort.*

KEYWORDS: *Double-drum vibratory roller, Vibration, Design parameters, Ride comfort*

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I. INTRODUCTION

Active, passive and semi-active suspension systems were summarized in vehicle seats to alleviate the harmful and damaging effects due to the transmitted vibration to the human body [1]. Three isolation models of the negative-stiffness-element (NSE), the damping-element (DE), and the NSE embedded to the small mass (NSE-SM) recommended that was connected with the seat suspension to ameliorate the ride comfort of the driver [2]. Three different models of the steel spring (SS), the roller-spring (RS), and the air spring (AS) in the negative-stiffness-structure (NSS) used for the driver's seat suspension system were proposed to ameliorate the vehicle's ride comfort [3]. The results of an industrial contract work, optimizing the parameters of a pneumatically suspended seat suspension system were recommended the parametric identification process of the seat suspension, and analyzes the measured results from point of view of the human discomfort [4]. The negative stiffness structure including steel springs (SS), roller springs (RS), and tuned mass damper (TMD) were proposed and studied to improve the driver's seat ride comfort. Their isolation efficiency and driver's ride comfort were evaluated via two indexes of the root mean square displacement and acceleration of the seat under two excitations of the random road surface and bumpy road surface based on the dynamic models of the SS, RS, and TMD [5]. An optimization of a four-degrees-of-freedom quarter car seat and suspension system using genetic algorithms was recommendedto determine a set of parameters to achieve the best performance of the driver [6]. The development of a hybrid method of an artificial neural network (ANN) and response surface methodology (RSM) was recommended to predict the peak seat-to-head transmissibility ratio of a seating suspension system and to evaluate its ride comfort for different seat design parameters. Four seat design parameters were selected as input parameters and arranged using the central composite design method [7]. A method for parameter identification of the magnetorheological damper (MRD) model with an improved firefly algorithm (IFA) based on a semi-active seat control system with three-degree-of-freedom (3-DOF) was proposed to investigate the ride comfort [8]. A systematic and effective optimization process based on a 3-D vehicle suspension dynamic model with eight DOF was proposed to attain the best compromise between ride comfort [9]. An innovative semi-active suspension system for off-road vehicles using a central pneumatic spring, double-acting pneumatic linear actuators for adjustable stiffness, and magnetorheological dampers for controlled damping, incorporating a quasi-zero stiffness suspension was proposed to enhance driver comfort [10].

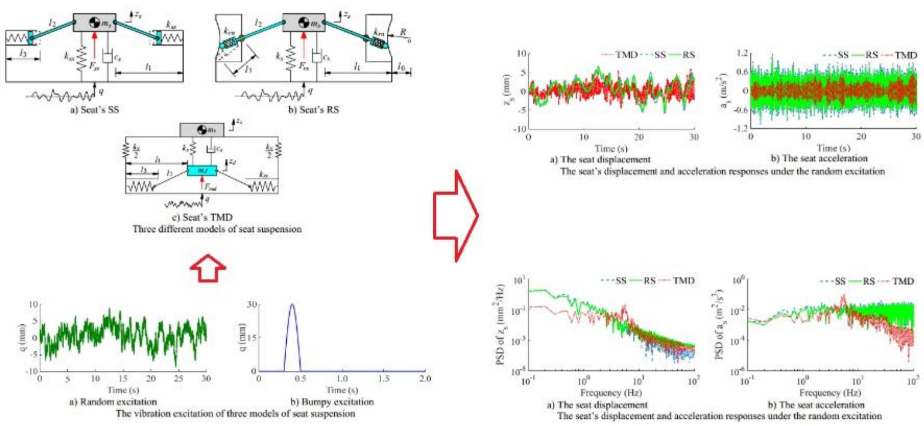

Fig.1. Arrangement of measuring points and acceleration sensors [7]

The aim of this paper is to analyze effects of design parameters of driver seat suspension system on ride comfort of a double-drum vibratory roller according to the ISO 2631:1997(E) [11] based on the modeling and simulation results in part 1 of the research team.

II. EVALUATION OF THE RIDE QUALITY

Currently there are many methods to evaluate the vehicle ride comfort such as frequency-domain method, time-domain method, etc. This study is based on ISO 2631-1 (1997), vibration evaluation based on the basic evaluation method including measurements of the weighted root-mean-square (rms) acceleration is defined by:

$$a_w = \left[\frac{1}{T}\int_0^T a_w{}^2(t)dt\right]^{1/2} \tag{1}$$

where, $a_w(t)$ is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurements.

In this way, for indications of likely reactions to various magnitudes of overall vibration in the public transport a synthetic index-called the weighted r.m.s acceleration, aw can be calculated from formula Eq.(1) and the r.m.s value of the vertical acceleration in vehicle would be compared with the values in Tab. 1.

Table 1 Comfort levels related to a_w threshold values

$a_w/(m.s^2)$	**Comfort level**
< 0.315	Not uncomfortable
0.315÷0.63	A little uncomfortable
0.5 ÷ 1.0	Fairly uncomfortable
0.8 ÷ 1.6	Uncomfortable
1.25 ÷ 2.5	Very uncomfortable
> 2	Extremely uncomfortable

III. RESULTS AND DISCUSSION

In order to simulate vibration of a double-drum vibratory roller, the differential equations of motion of Fig.3 are simulated under two operating cases by the MATLAB/Simulink with design parameters of double-drum vibratory roller in reference [12].

Effects of driver's SSS stiffness on vehicle ride comfort: The stiffness values of driver's SSS, k_s=[0.5 1 1.5] k_{s0} in which k_{s0} is value of the original stiffness value of driver's SSS are selected to analyze its influence on vehicle ride comfort. The time domain acceleration responses of driver's seat (a_s) and pitching cab angle (a_{cphi}) when the front drum compacts on the elastic soil ground with the excitation force of drum as F_{01}=0.128 x10^6 N, f_1=48 Hz and rear drum moves the ISO class E road surface at vehicle speed of 2.0 km/h (Case 2) are shown Fig. 2 and Fig. 3.

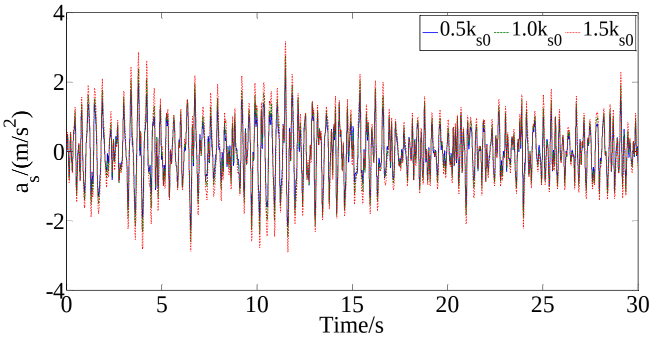

Fig. 2 Time domain acceleration response of driver's seat with varying stiffness (Case 2)

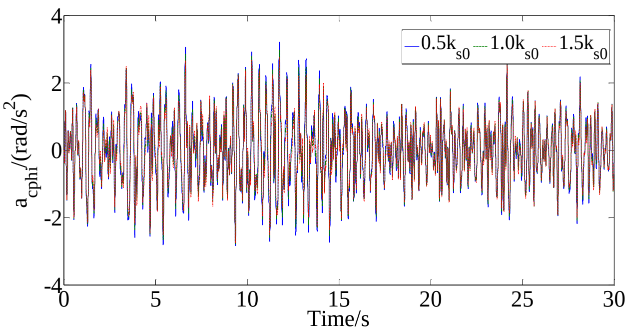

Fig. 3 Time domain acceleration response of pitching cab angle with varying stiffness (Case 2)

From the obtained results of Fig.2 and Fig.3, we could be determined the values of a_{ws} and a_{wcphi} as a_{ws}=0.6908m/s^2, a_{wcphi}=0.9812 rad/s^2 with k_s=0.5k_{s0}; a_{ws}=0.8005 m/s^2, a_{wcphi}=0.9327rad/s^2 with k_s=1.0k_{s0}; and a_{ws}=0.9450 m/s^2, a_{wcphi}=0.9025 rad/s^2 with k_s=1.5k_{s0}. From the obtained results of Fig.2, the k_s value decrease from 1.0xk_{s0} to 0.5 k_{s0}, the a_{ws} value decrease leading to improved vertical comfort of driver's seat and the k_s value increase from 1.0xk_{s0} to 1.5 k_{s0}, the a_{ws} value increase leading to reduced vertical comfort of the driver's seat. From the obtained results of Fig.3, the k_s value decrease from 1.0xk_{s0} to 0.5 k_{s0}, the a_{wcphi} value decrease that leads to reduced driver's seat shaking and the k_s value increase from 1.0xk_{s0} to 1.5 k_{s0}, the a_{ws} value increase that leads to improved driver's seat shaking.

Effects of damping coefficients of driver's SSS on vehicle ride comfort: The damping coefficient values of driver's SSS, c_s=[0.5 1 1.5] c_{s0} in which c_{s0} is value of the original damping coefficient value of driver's SSS are selected to analyze its influence on vehicle ride comfort. The time domain acceleration responses of driver's seat (a_s) and pitching cab angle (a_{cphi}) when the front drum compacts on the elastic soil ground with the excitation force of drum as F_{01}=0.128 x10^6 N, f_1=48 Hz and rear drum moves the ISO class E road surface at vehicle speed of 2.0 km/h (Case 2) are shown Fig. 4 and Fig. 5.

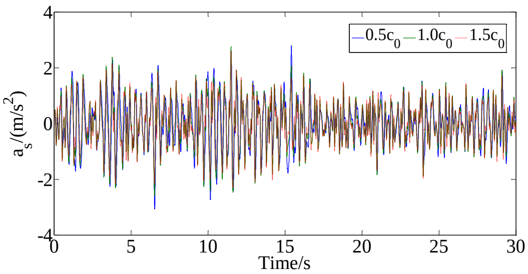

Fig. 4 Time domain acceleration response of driver's seat with varying stiffness (Case 2)

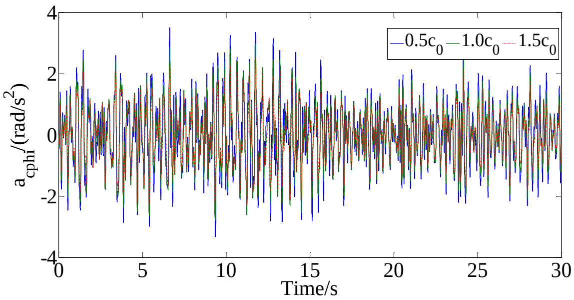

Fig. 5 Time domain acceleration response of pitching cab angle with varying stiffness (Case 2)

From the obtained results of Fig.4 and Fig.5, we could be determined the values of a_{ws} and a_{wcphi} as a_{ws}=0.8191m/s^2, a_{wcphi}=1.0626 rad/s^2 with c_s=0.5c_{s0}; a_{ws}=0.8005 m/s^2, a_{wcphi}=0.9327rad/s^2 with c_s=1.0c_{s0}; and a_{ws}=0.7452m/s^2, a_{wcphi}=0.7958 rad/s^2 with c_s=1.5c_{s0} From the obtained results of Fig.4, the c_s value decrease from 1.0xc_{s0} to 0.5 c_{s0}, the a_{ws} value decrease leading to reduced vertical comfort of driver's seat and the k_s value increase from 1.0xc_{s0} to 1.5 c_{s0}, the a_{ws} value increase leading to improved vertical comfort of the driver's seat. From the obtained results of Fig.6, the c_s value decrease from 1.0xc_{s0} to 0.5 c_{s0}, the a_{wcphi} value decrease that leads to reduced driver's seat shaking and the c_s value increase from 1.0xc_{s0} to 1.5 c_{s0}, the a_{ws} value increase that leads to improved driver's seat shaking.

IV. CONCLUSION

This study is to analyze effects of design parameters of driver seat suspension system on ride comfort of a double-drum vibratory roller according to the ISO 2631:1997(E) based on the modeling and simulation results in part 1 of the research team. The conclusions could be drawn: (1) The k_s value decrease from 1.0xk_{s0} to 0.5 k_{s0}, the a_{ws} value decrease leading to improved vertical comfort of driver's seat and the k_s value increase from 1.0xk_{s0} to 1.5 k_{s0}, the a_{ws} value increase leading to reduced vertical comfort of the driver's seat; (2) the c_s

value decrease from $1.0xc_{s0}$ to $0.5\ c_{s0}$, the a_{ws} value decrease leading to reduced vertical comfort of driver's seat and the k_s value increase from $1.0xc_{s0}$ to $1.5\ c_{s0}$, the a_{ws} value increase leading to improved vertical comfort of the driver's seat; (3) The k_s value decrease from $1.0xk_{s0}$ to $0.5\ k_{s0}$, the a_{wcphi} value decrease that leads to reduced driver's seat shaking and the k_s value increase from $1.0xk_{s0}$ to $1.5\ k_{s0}$, the a_{ws} value increase that leads to improved driver's seat shaking.

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