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An evaluation of a double-drum vibratory roller ride comfort under different operating conditions

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ABSTRACT: The purpose of this study is to evaluate influence of different operating conditions on a double-drum vibratory roller ride comfort. A half-vehicle dynamic model of a double-drum vibratory roller with hydraulic cab mount system (HCMs) under the different operating conditions is set up to analyze vehicle ride comfort. The weighted root mean square (RMS) acceleration responses of the cab according to ISO 2631-1(1997) are chosen as objective functions. The influence of different operating conditions on a double-drum vibratory roller ride comfort is evaluated. The results show that the operating conditions have a great influence on vehicle ride comfort.

KEYWORDS: Vibratory roller, drum, operating condition, ride comfort.

I. INTRODUCTION

Vibrating rollers often operate on harsh environments. In terms of structure, the vehicle is usually not equipped with a suspension system connecting the bridge and the chassis. The vibratory roller operates and moves on various kinds of soil ground and when it does vibration excitation sources, such as soil ground, drum and engine are transmitted to the driver through the isolation systems of the cab and seat, which has direct influence on the driver's health and their working efficiency. The riding comfort of a vibratory roller under the different soil grounds was evaluated using a nonlinear dynamics model of a single drum vibratory roller based on the analysis of the contact physics of the wheel with different soil grounds [1]. The half-vehicle dynamic model of a double-drum vibratory roller was established for analyzing and evaluating the effect of the operating conditions of a double-drum vibratory roller on vehicle ride comfort according to the ISO 2631:1997(E) standard [2]. The 3D nonlinear dynamic model of a single drum vibratory roller was developed based on Adam D. and Kopf F's elastic-plastic soil model and Bekker hypothesis of the soft soil ground to simulate and compare with experimental results [3]. The stiffness and damping coefficients of the cab's isolation system were analyzed its effect on vehicle ride comfort using a 3D nonlinear dynamic model of a single drum vibratory roller based on the analysis of nonlinear geometric characteristics of wheel-deformation of soil ground contact [4]. The dynamic characteristics of the vibratory roller test-bed vibration isolation system were analyzed and evaluated using theoretical analysis, experimental research and simulation analysis methods [5]. A half-vehicle ride dynamic model was established based on the drum-ground interaction to find the optimal design parameters of drum's isolation systems for a double-drum vibrating roller so that the ride comfort [6]. The 3D nonlinear dynamic model of a single drum vibratory roller was developed based on the analysis of the interaction between vibratory roller and soil to find the optimal design parameters of cab's isolation system using a multi-objective optimization method [7]. The half-vehicle ride dynamic model of a single drum vibratory roller was established under various operating conditions to find out the optimal parameters of cab's isolation system to improve the vehicle ride comfort using a genetic algorithm (GA) and a multi-objective optimization algorithm [8]. In this study, a half-vehicle dynamic model of a double-drum vibratory roller with hydraulic cab mount system (HCMs) is established to evaluate influence of different operating conditions on a double-drum vibratory roller ride comfort. The weighted r.m.s acceleration responses of the vertical driver's seat (a_{ws}) and pitch angle of the cab (a_{wphi}) according to the ISO 2631:1997(E) standard [10] are chosen as objective functions which uses Matlab/Simulink software to simulate and analyze the objective functions.

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II. HALF-VEHICLE DYNAMIC MODEL OF A DOUBLE –DRUM VIBRATORY ROLLER

A double- double drum vibratory roller with with hydraulic cab mount system (HCMs) is selected for analyzing the influence of the various operating conditions on vehicle ride comfort. A half-vehicle dynamic model is established, as shown in Fig.1. In Fig. 1, m_{di}, m_b, m_c and m_s are the masses of the dynamic drums, frame, cab and driver's seat, respectively; I_b and I_c are the moment of inertia of the vehicle (including frame, internal combustion engine and and other parts) and cab, respectively; k_{si}, k_{di}, k_s and c_{si}, c_{di}, c_s are the stiffness and damping coefficients of elastic road surfaces, front and rear mount systems of drums and driver's seat suspension system, respectively; z_{di}, z_b, z_c and z_s are the vertical displacements at centre of gravity of the front and rear drums, vehicle body, cab and driver's seat, respectively; φ_b and φ_c are the pitch angle displacements of vehicle body and cab, respectively; q_i are the excitation of road surface roughness at drums, respectively; l_j are the distances; $F_{ei}=F_{0i}\sin(\omega_i t)$ are the force excitations of the vibrating drums; F_{0i} are the amplitude of force excitations; ω_i are the angular frequencies of the vibrators; F_{ci} are the vertical forces of hydraulic cab mount systems (HCMs) and v is the vehicle speed (i=1÷2, j=1÷6).

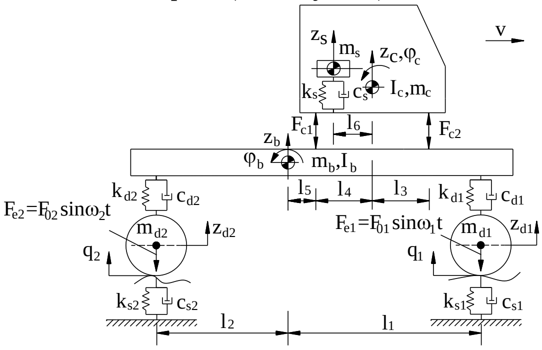

Fig. 1. Half-vehicle dynamic model of a double-double drum vibratory roller [11]

Equations of motion [11]: The equations of motion for a double-drum vibratory roller using Newton's second law of motion are written in two operating conditions below. From Fig. 1, the motion equations of vehicle mass are written as follows:
The equation of motion for driver's seat is written as follows

$$m_s\ddot{z}_s = -F_s \tag{1}$$

The equations of motion for cab are written as follows

$$m_c\ddot{z}_c = F_s - \sum_{i=1}^{i=2} F_{ci} \tag{2}$$

$$I_c\varphi_c = F_s l_6 + \sum_{i=1}^{i=2} (-1)^i F_{ci} l_{i+2} \tag{3}$$

The equations of motion for frame are written as follows

$$m_f\ddot{z}_f = \sum_{i=1}^{i=2} F_{ci} - \sum_{i=1}^{i=2} F_{di} \tag{4}$$

$$I_f\varphi_f = \sum_{i=1}^{i=2} (-1)^i F_{di} l_i + F_{c2} l_5 + F_{c1} \sum_{j=3}^{j=5} l_j \tag{5}$$

where, F_s is the vertical force of driver's seat which is determined by Eq. (6), F_{ci} are the vertical forces of HCMs which are determined by Eq.(7) and Eq.(8), F_{di} are the vertical forces of front and rear mount systems of drums which could be determined through two cases.

$$F_s = [k_s(z_s - z_c + l_6\varphi_c) + c_s(\dot{z}_s - \dot{z}_c + l_6\dot{\varphi}_c] \tag{6}$$

$$F_{c1} = [k_{c1}(z_c - l_4\varphi_c - z_b - l_5\varphi_b) + c_{c1}(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b) \tag{7}$$
$$+ c_{h1}(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b)\left|\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b\right|]$$

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$$F_{c2}=[k_{c2}(z_c+l_3\varphi_c-z_b-\sum_{n=3}^{n=5}l_n\varphi_b)+c_{c2}(\dot{z}_c+l_3\dot{\varphi}_c-\dot{z}_b-\sum_{n=3}^{n=5}l_n\dot{\varphi}_b)$$
$$+c_{h2}(\dot{z}_c+l_3\dot{\varphi}_c-\dot{z}_b-\sum_{n=3}^{n=5}l_n\dot{\varphi}_b)\left|(\dot{z}_c+l_3\dot{\varphi}_c-\dot{z}_b-\sum_{n=3}^{n=5}l_n\dot{\varphi}_b)\right|] \quad (8)$$

Case 1: Vehicle moves into the workshop: The drum of vibratory roller in contact with the rigid road surface is the contact point which is considered in this study. The vertical forces of front and rear mount systems of drums are defined as

$$F_{d1}=k_{d1}\left(z_f+l_1\varphi_f-q_d\right)+c_d\left(\dot{z}_f+l_1\dot{\varphi}_f-\dot{q}_d\right) \quad (9)$$
$$F_{d2}=k_{d2}\left(z_f-l_2\varphi_f-q_d\right)+c_{d2}\left(\dot{z}_f-l_2\dot{\varphi}_f-\dot{q}_d\right) \quad (10)$$

where, q_d and q_d are the excitation of road surface roughness at drum and tire. The road surface roughness according to the International Standards Organization (ISO 8608) [9] is road excitation which is simulated in space domain and acts as an input to the vehicle-road model.

Case 2: Vehicle operates in the workshop: The equations of motion for the dynamic drums are written as follows

$$m_{d1}\ddot{z}_{d1}=F_{e1}+F_{d1}-k_{s1}z_{d1}-c_{s1}\dot{z}_{d1} \quad (11)$$
$$m_{d2}\ddot{z}_{d2}=F_{e2}+F_{d2}-k_{s2}z_{d2}-c_{s2}\dot{z}_{d2} \quad (12)$$

III. VEHICLE RIDE COMFORT EVALUATION METHOD [12]

The time-domain method can be applied to evaluate the vehicle ride comfort according to ISO 2631-1 (1997) [10], in this study, the vibration evaluation based on the basic evaluation methods including measurements of the weighted root-mean-square (RMS) acceleration defined as

$$a_w=\left[\frac{1}{T}\int_0^T a_z^{\,2}(t)dt\right]^{1/2} \quad (13)$$

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

For indications of the likely reactions to various magnitudes of overall vibration in the public transport and vehicle, a synthetic index-called the root-mean-square (RMS) acceleration, a_{wz} can be calculated from formula Eq. (4); besides, the RMS value of the acceleration in vehicle would be compared with the values in Tab.1.

Tab.1: Comfort levels related to aw threshold values [10]

$a_w/(m/s^2)$	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

IV. SIMULATION AND ANALYSIS RESULTS

In order to evaluate the influence of the various operating conditions on vehicle ride comfort, the differential equations of motion of Fig.1 are simulated under two operating cases by the MATLAB/Simulink with design parameters and hydraulic cab mount system (HCMs) in the reference [11]. The simulation results of the time domain acceleration responses of the vertical driver's seat (a_s) and cab's pitch angle (a_{phi}) when the vehicle moves on the ISO class D road surface (poor condition) at the vehicle speed of 10 km/h (Case1) are shown Fig.2 in Case 1.

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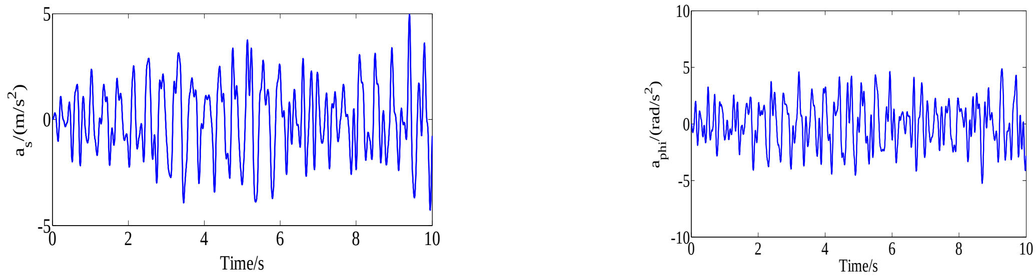

(a) Vertical driver's seat (b) Cab's pitch angle

Fig.2. Time domain acceleration responses when the vehicle moves on the ISO class D road surface (poor condition) at the vehicle speed of 10 km/h (Case1)

From the results of Fig.2, the values of root-mean-square (RMS) of acceleration responses of the vertical driver's seat (a_{ws}) and cab pitching angle (a_{wphi}) are respectively determined through Eq. (13) as a_{ws}= 1.6600 m/s^2, a_{wcphi}= 1.9442 rad/s^2. This result, compared with Tab.1, shows that human may feel uncomfortable. The vehicle's operating conditions will continue to be reviewed and evaluated in in the following section.

Influence of road surface conditions at Case 1: The vehicle moving on the different road surface conditions, five road surface conditions from ISO class A road surface to ISO class E road surface according to ISO 8068(1995) are selected for evaluation when vehicle moves at speed of 10 km/h. The values of the root mean square (RMS) acceleration responses of the vertical driver's seat (a_{ws}) and cab pitching angle (a_{wphi}) with variable road surface conditions based on the International Standard ISO 2631-1: 1997 which are determined through Eq. (13), as shown in Fig 3. From the results of Fig 3, it shows that the a_{ws} and a_{wphi} values increase height fast when the vehicle moves on poor and very poor road surface conditions, especially ISO class D road surface and ISO class E road surface. The a_{ws} and a_{wphi} avalues respectively increase when the vehicle moves on from ISO class A road surface to ISO class E road surface which make the vehicle comfort worse, when the road surface conditions become worse.

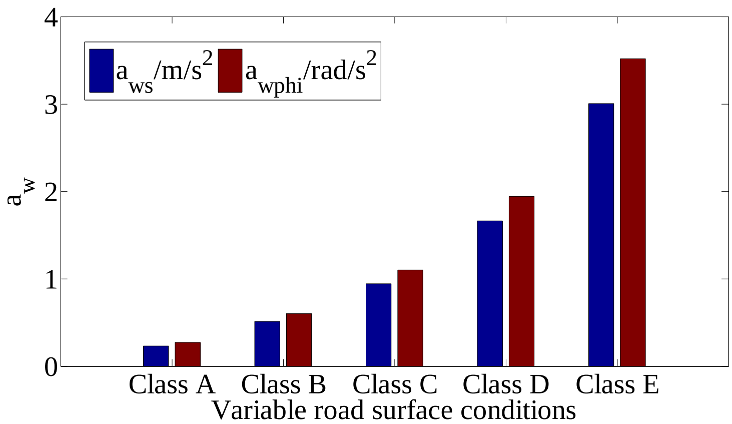

Fig.3. a_{ws} and a_{wphi} values with variable road surface conditions

Influence of the stiffness and damping coefficients of elastic ground surfaces on vehicle ride comfort at Case 2: The variable k_{s1}, k_{s2} and c_{s1}, c_{s2} values of the elastic ground surface such as 0.5x[k_{s1}, k_{s2},c_{s1},c_{s2}] (Condition 1-C1), 1.0x[k_{s1}, k_{s2},c_{s1},c_{s2}] (Condition 2-C2), and 1.5x[k_{s1}, k_{s2},c_{s1},c_{s2}] (Condition 3-C3), where [k_{s1}, k_{s2},c_{s1},c_{s2}] are the original values in the reference [11] when the vehicle compacts on the original place at the front and rear drums with the excitation forces as F_{01}=0.128 x10^6 N, f_{01}=48 Hz and F_{02}=0.96 x10^5 N, f_{01}=54 Hz. The values of the root mean square (RMS) acceleration responses of the vertical driver's seat (a_{ws}) and cab pitching angle (a_{wphi}) with variable road surface conditions based on the International Standard ISO 2631-1: 1997 which are determined through Eq. (13), as shown in Fig 4. From the results of Fig 4, it shows that the a_{ws} and

a_{wphi} increase height fast when k_{s1}, k_{s2} and c_{s1}, c_{s2} values of the elastic ground surface increase. The a_{ws} and a_{wphi} avalues respectively increase when k_{s1}, k_{s2} and c_{s1}, c_{s2} values of the elastic ground surface increase which make the vehicle comfort worse.

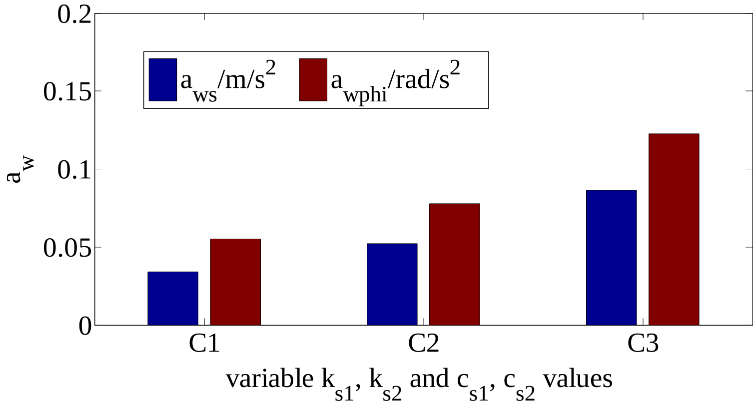

Fig.4. a_{ws} and a_{wphi} values with variable k_{s1}, k_{s2} and c_{s1}, c_{s2} values of the elastic ground surface

V. CONCLUSION

In this paper, a half-vehicle dynamic model of a double-drum vibratory roller with hydraulic cab mount system (HCMs) is established to evaluate the influence of the operating conditions on the ride comfort of a double-drum vibratory roller according to the international standard ISO 2631-1 (1997). The major conclusions drawn from the analysis can be summarized as follows: (1) The a_{ws} and a_{wphi} values increase height fast when the vehicle moves on poor and very poor road surface conditions (Case 1) and (2) The a_{ws} and a_{wphi} increase height fast when k_{s1}, k_{s2} and c_{s1}, c_{s2} values of the elastic ground surface increase (Case 2). In addition, the research results are useful references for designers and manufacturers in the field of designing isolation systems of the construction machines to improve the vehicle ride comfort.

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