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Vibration analysis of double-drum vibratory vibrations Part 1: Modeling and simulation

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ABSTRACT: *The vibrations of construction machines not only affect the ride comfort but also the machine's durability and longevity. In order to analyze vibrations of a double-drum vibratory roller, a physical model and a mathematical model are established to describe the system of differential equations that describe the vibrations of objects in the mechanical system, and then the system of differential equations describing the vibration of a double-drum vibratory roller are simulated by matlab/simulink software. The results of the study simulated the vibration of the driver's seat and the vehicle's cabin shaking angle. In addition, the study results are the theoretical basis for researching the isolation systems to improve the vehicle's ride comfort*

KEYWORDS: *Double-drum vibratory roller, Vibration, Model, Simulation.*

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I. INTRODUCTION

The construction machinery market has required increasingly not only on working capacity but also ride comfort quality, therefore it has required increasing toward researchers and manufacturers. The performance of semi-active hydraulic cab mount system (SHCMs) of a double-drum vibratory roller in the direction of enhancing vehicle ride comfort proposed and evaluated under different operating conditions [1]. A half-vehicle dynamic model of a double-drum vibratory roller with hydraulic cab mount system (HCMs) established to evaluate influence of different operating conditions on a double-drum vibratory roller ride comfort [2]. A half-vehicle ride dynamic model established based on the drum-ground interactions to analyze the effect of the operating conditions of a double-drum vibratory roller on vehicle ride comfort [3]. A half-vehicle ride dynamic model of a double-drum vibratory roller established to compare the ride performance of a liquid-filled cab mount system (LCMs) with an annular orifice and an original rubber cab mount system (RCMs) under the operating conditions [4]. A nonlinear dynamics model of the vibratory roller under interaction of the deformable terrain and drum established to evaluate the ride quality of vibratory roller's cab used the different isolation mounts including the traditional rubbers, hydraulics, and hydro-pneumatics [5], as shown in Fig.1.

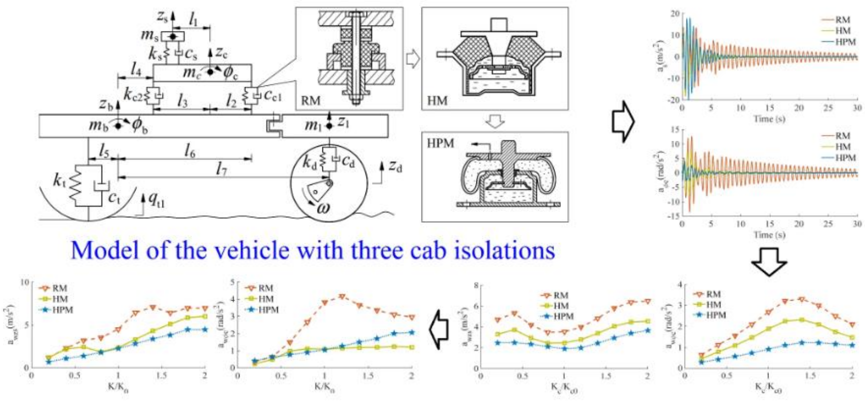
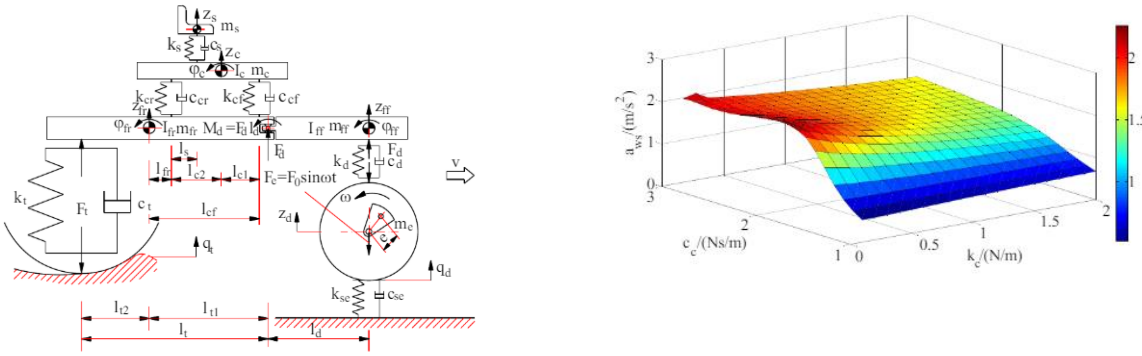


Fig.1. Effective simulation results [5]

A dynamic model of the off-road vibratory roller built and researched under various interactions of the drum/wheel and off-road terrain to analyze and evaluate a new design of the seat suspension equipped with the quasi-zero stiffness (QZS) and hydraulic mount (HM) of the cab isolation system to improve the ride comfort of off-road vibratory rollers [6]. A quarter-vehicle dynamic model of vibratory roller established to see the effect of the values of the damping coefficient of drum's isolation system on vehicle ride comfort [7]. A half-vehicle ride dynamic model established under the ground surface deformation to analyze the effects of drum's excitation frequency on vehicle ride [8]. A half vehicle dynamic model of a single-drum vibratory roller established to analyze the effects of parameters of cab's isolation system on ride comfort [9], as show in Fig.2.



(a) Half vehicle dynamic model (b) a_z values with the change of k_c and c_c values

Fig.2. Study results [9]

The aim of this paper is to establish physical and mathematical models for vibrations of a double-drum vibratory roller using the traditional rubber isolation of cab, and then we conduct simulation using matlab/simulink software.

II. PHYSICAL AND MATHEMATICAL

2.1. Physical Model of a Double-drum Vibratory Roller

A half-vehicle dynamic model of a double-drum vibratory roller is shown in Fig.3 [11].

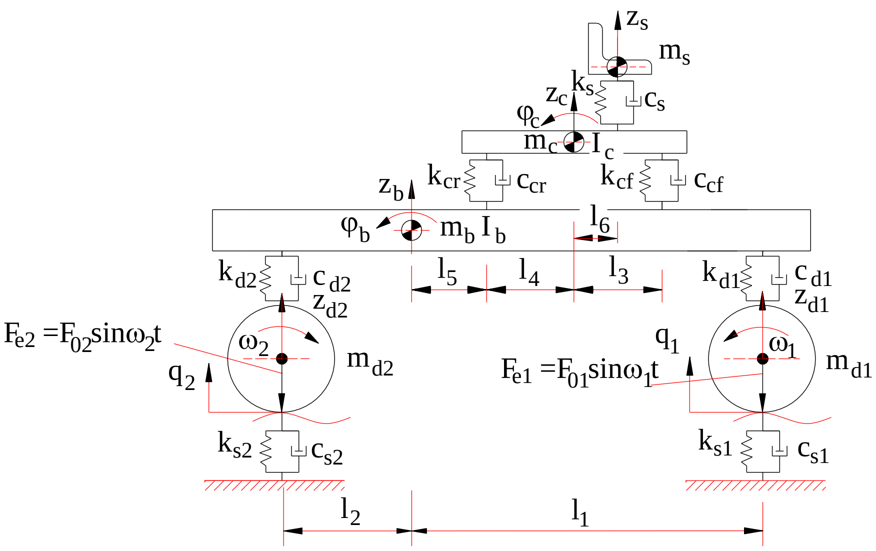


Fig.3 Half-vehicle dynamic model of a double-drum vibratory roller

Explanation of the symbols for Fig. 2, 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).

2.2. Mathematical Model of a Double-drum Vibratory Roller

A combined method of the multi-body system theory and D'Alembert's principle is chosen. 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 make up force and moment equations to describe vehicle dynamic subsystems. The equations of motion of objects in Fig.3 could be written by a combined method as follows.

$$
\begin{aligned}
&m_s\ddot{z}_s = k_s\left(z_s - z_c - l_6\varphi_c\right) + c_s\left(\dot{z}_s - \dot{z}_c - l_6\dot{\varphi}_c\right)\\
&m_c\ddot{z}_c = \left[k_s\left(z_s - z_c - l_6\varphi_c\right) + c_s\left(\dot{z}_s - \dot{z}_c - l_6\dot{\varphi}_c\right)\right] - \left[k_{cf}\left(z_c + l_3\varphi_c - z_b - l_5\varphi_b\right) + c_{cf}\left(\dot{z}_c + l_3\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b\right)\right]\\
&-\left[k_{cr}\left(z_c - l_4\varphi_c - z_b - \left(l_3+l_4+l_5\right)\varphi_b\right) + c_{cr}\left(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - \left(l_3+l_4+l_5\right)\dot{\varphi}_b\right)\right]\\
&I_c\varphi_c = \left[k_s\left(z_s - z_c - l_6\varphi_c\right) + c_s\left(\dot{z}_s - \dot{z}_c - l_6\dot{\varphi}_c\right)\right]l_6 - \left[k_{cf}\left(z_c + l_3\varphi_c - z_b - l_5\varphi_b\right) + c_{cf}\left(\dot{z}_c + l_3\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b\right)\right]l_3\\
&+\left[k_{cr}\left(z_c - l_4\varphi_c - z_b - \left(l_3+l_4+l_5\right)\varphi_b\right) + c_{cr}\left(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - \left(l_3+l_4+l_5\right)\dot{\varphi}_b\right)\right]l_4\\
&m_b\ddot{z}_b = \left[k_{cf}\left(z_c + l_3\varphi_c - z_b - l_5\varphi_b\right) + c_{cf}\left(\dot{z}_c + l_3\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b\right)\right]\\
&+\left[k_{cr}\left(z_c - l_4\varphi_c - z_b - \left(l_3+l_4+l_5\right)\varphi_b\right) + c_{cr}\left(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - \left(l_3+l_4+l_5\right)\dot{\varphi}_b\right)\right]\\
&-\left[k_{d1}\left(z_b + l_1\varphi_b - z_{d1}\right) + c_s\left(\dot{z}_b + l_1\dot{\varphi}_b - \dot{z}_{d1}\right)\right] - \left[k_{d2}\left(z_b - l_2\varphi_b - z_{d2}\right) + c_s\left(\dot{z}_b - l_2\dot{\varphi}_b - \dot{z}_{d2}\right)\right]\\
&I_b\ddot{\varphi}_b = \left[k_{cf}\left(z_c + l_3\varphi_c - z_b - l_5\varphi_b\right) + c_{cf}\left(\dot{z}_c + l_3\dot{\varphi}_c - \dot{z}_b - l_5\dot{\varphi}_b\right)\right]\left(l_3+l_4+l_5\right)\\
&+\left[k_{cr}\left(z_c - l_4\varphi_c - z_b - \left(l_3+l_4+l_5\right)\varphi_b\right) + c_{cr}\left(\dot{z}_c - l_4\dot{\varphi}_c - \dot{z}_b - \left(l_3+l_4+l_5\right)\dot{\varphi}_b\right)\right]l_5\\
&-\left[k_{d1}\left(z_b + l_1\varphi_b - z_{d1}\right) + c_s\left(\dot{z}_b + l_1\dot{\varphi}_b - \dot{z}_{d1}\right)\right]l_1 + \left[k_{d2}\left(z_b - l_2\varphi_b - z_{d2}\right) + c_s\left(\dot{z}_b - l_2\dot{\varphi}_b - \dot{z}_{d2}\right)\right]l_2\\
&m_{d1}\ddot{z}_{d1} = F_{01}\sin\omega_1 t + \left[k_{d1}\left(z_b + l_1\varphi_b - z_{d1}\right) + c_s\left(\dot{z}_b + l_1\dot{\varphi}_b - \dot{z}_{d1}\right)\right] - \left[k_{s1}z_{d1} + c_{d1}\dot{z}_{d1}\right]\\
&m_{d2}\ddot{z}_{d2} = F_{02}\sin\omega_2 t + \left[k_{d2}\left(z_b - l_2\varphi_b - z_{d2}\right) + c_s\left(\dot{z}_b - l_2\dot{\varphi}_b - \dot{z}_{d2}\right)\right] - \left[k_{s2}z_{d2} + c_{d2}\dot{z}_{d2}\right]
\end{aligned}
\tag{1}
$$

Case 1: *Vehicle moves into the construction site* [1]

Vehicle moves on the ground surface and dynamic drums is uncompressed. The drums contact with the rigid ground surface. The vertical forces of front and rear mount systems of drums are defined as

$$F_{d1} = k_{d1}\left(z_f + l_1\varphi_f - q_d\right) + c_d\left(\dot{z}_f + l_1\dot{\varphi}_f - \dot{q}_d\right) \tag{2}$$

$$F_{d2} = k_{d2}\left(z_f - l_2\varphi_f - q_d\right) + c_{d2}\left(\dot{z}_f - l_2\dot{\varphi}_f - \dot{q}_d\right) \tag{3}$$

where, q_i are the excitations of the rigid ground surface which is described based on the International Standards Organization (ISO 8608) [10].

Case 2: *Vehicle operates in the construction site* [1]

When both front and rear dynamic drums compact on the original place, the motion equations of the front and rear dynamic drums are written as follows

$$m_{d1}\ddot{z}_{d1} = F_{e1} + F_{d1} - k_{s1}z_{d1} - c_{s1}\dot{z}_{d1} \tag{4}$$

$$m_{d2}\ddot{z}_{d2} = F_{e2} + F_{d2} - k_{s2}z_{d2} - c_{s2}\dot{z}_{d2} \tag{5}$$

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 [11]. The time domain acceleration responses of driver's seat (a_s) and pitching cab angle (a_{cphi}) when the vehicle moves on very poor ground surface condition (ISO class E) at v=3 km/h (Case1) are shown Fig. 4 and Fig. 5.

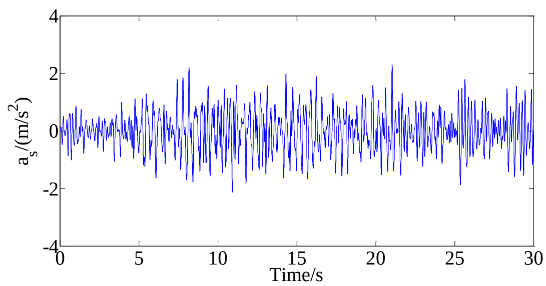


Fig.4 Time domain acceleration response of driver's seat (Case 1)

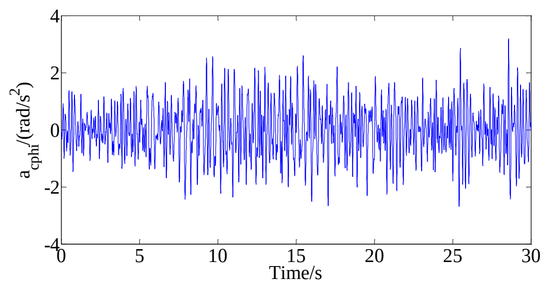


Fig. 5 Time domain acceleration response of pitching cab angle (Case 1)

From the obtained results of Fig.5, we show that the peak maximum amplitude value of a_s= 2.2995 m/s^2 and the obtained results of Fig.6, we show that the peak maximum amplitude value of a_{cphi}= 3.1915 rad/s^2. From these values, the vehicle's ride comfort is relatively uncomfortable for the driver.

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. 6 and Fig. 7.

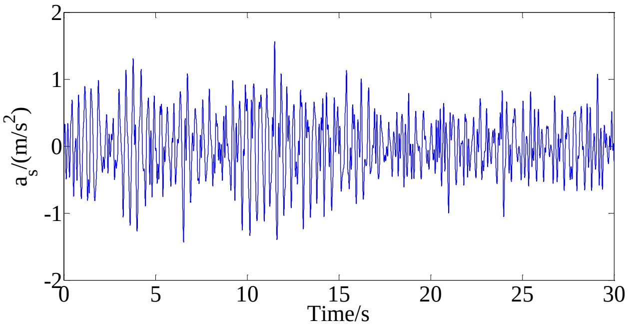


Fig.6 Time domain acceleration response of driver's seat (Case 2)

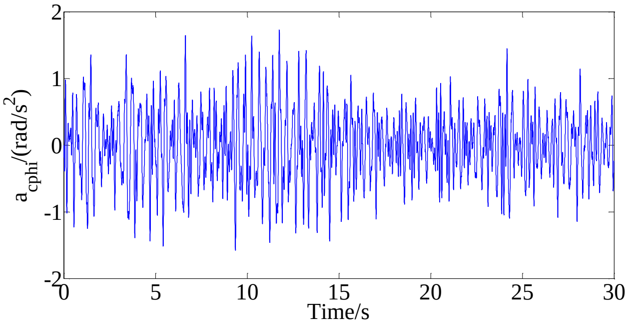


Fig. 7 Time domain acceleration response of pitching cab angle (Case 2)

From the obtained results of Fig. 6, we show that the peak maximum amplitude value of a_s= 1.5589 m/s^2 and the obtained results of Fig.7, we show that the peak maximum amplitude value of a_{cphi}= 1.7265 rad/s^2. From these values, the vehicle's ride comfort is relatively uncomfortable for the driver.

IV. CONCLUSION

This study is to analyze vibrations of a double-drum vibratory roller, physical and mathematical models of a double-drum vibratory roller are established to simulate under two operating cases. The conclusions could be drawn: (1) The peak maximum amplitude values of a_s= 2.2995 m/s^2 and a_{cphi}= 3.1915 rad/s^2 in Case 2; (2) The peak maximum amplitude values of a_s= 1.5589 m/s^2 and a_{cphi}= 1.7265 rad/s^2 in Case 2. Both conditions lead to the driver feeling very uncomfortable. In addition, the results of this paper are the basis for analyzing the effects of operating conditions on vehicle ride comfort which will be published by our research team.

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