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Modeling and Vertical Dynamic Analysis of Wheel Loaders

Canh Chi Huan
Faculty of Vehicle and Energy Engineering, Thai Nguyen University of Technology,Thai Nguyen, Vietnam

ABSTRACT: *This paper presents a dynamic analysis of a wheel loader equipped with a rubber vibration isolation system. A 7-degree-of-freedom dynamic model is developed to simulate the loader's behavior under road conditions, with simulations performed using Matlab/Simulink software. The weighted root mean square (RMS) of the acceleration responses for the vertical driver's seat, vertical cab, and the pitching and rolling cab angles, in accordance with the international standard ISO 2631-1 (1997), is selected as the objective function for performance evaluation. The results indicate that road surface quality significantly influences the loader's dynamic performance, with rougher surfaces leading to higher vibrational impacts. These findings underscore the importance of road conditions in the dynamic behavior of wheel loaders.*

KEYWORDS: *wheel loaders, cab isolation system, simulation and analysis, ride comfort, dynamic model.*

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I. INTRODUCTION

Wheel loaders are a type of heavy construction equipment extensively used in industries such as mining, construction, and agriculture due to their ability to handle large volumes of material in various challenging environments. These working conditions often involve uneven terrain, harsh impacts, and frequent vibrations that can compromise machine stability and operator comfort. Unlike most other vehicles, wheel loaders are not equipped with suspension systems. As a result, vibrations generated from the ground are transmitted directly to the cab and the operator's seat, exposing the driver to high levels of mechanical shock.Continuous exposure to severe vibrations can lead to health issues such as musculoskeletal disorders, fatigue, and reduced cognitive focus, ultimately impacting the productivity and safety of operators. To evaluate the vibration level of wheel loaders, Xiaojing Zhao et al. [1] applied ISO 2631-1 (1985) and ISO 2631-1 (1997) standards to comprehensively analyze the impact of wheel loader vibration on human occupants. In [2], a study investigated the impact of tires equipped with ladder chains, which may heighten the risk of negative health consequences when operated for over 8 hours, as per the guidelines outlined in ISO 2631-1. In another study, Xiaojing Zhao et al. [3] measured vibrations in various scenarios corresponding to real-world conditions, using ISO 2631-1:1997 and ISO 2631-5:2004 standards to evaluate whole-body vibration effects on human health.Many approaches have been explored to address this issue, focusing on enhancing cab isolation systems, optimizing seat designs, and improving machine structural dynamics to mitigate the impact of vibrations.Consequently, studies have examined the effects of various parameters and optimization for cab isolation systems [4, 5]. Additionally, several structural development studies have proposed hydraulic mounts (HM) and hydro-pneumatic mounts (HPM) for cab isolation systems to improve ride comfort [6, 7, 8]. These mounts have shown promising results in reducing vibrations and enhancing stability. Furthermore, control strategies such as Fuzzy and PID have been applied to manage semi-active cab isolation systems [9, 10, 11]. Research findings indicate a significant improvement in ride comfort with these control methods. By establishing a detailed dynamic model of the wheel loader, this study focuses on analyzing the loader's dynamics under various road inputs. This approach serves as a foundation for evaluating the loader's vibration response and its impact on ride comfort.

II MATERIALS AND METHODS

2.1. Wheel Loader Model

The basic structure of a wheel loader includes the seat, cab, body, wheels, and actuator mechanism. Unlike other vehicles, wheel loaders are not equipped with a suspension system, so vibrations are primarily limited by the cab vibration isolation system. Based on the actual structure of the machine, a full-vehicle dynamic model with 7 degrees of freedom is established, as shown in Figure 1.

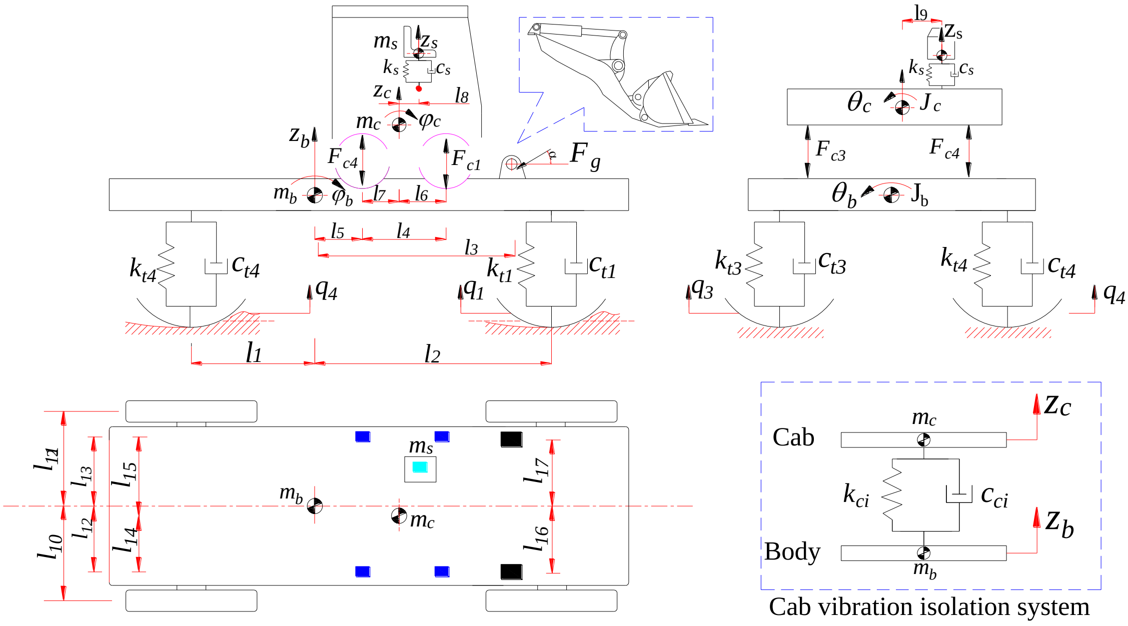


Fig. 1. *The dynamic model of the wheel loader.*

In Fig. 1, m_s, m_c, and $m_b$are the mass of the driver's seat, cab, and body;I_b and I_c are the moment of inertia of the body and the cab;z_s, z_c, and z_b are the vertical displacements of the driver's seat, cab and body, respectively; $\varphi_c$and $\varphi_b$are the pitching angles of the cab and body; θ_c and θ_b are the rolling angles of the cab and body;k_s and $c_s$are stiffness and damping coefficients of the driver's seat suspension; k_{ci} and c_{ci} are stiffness and damping coefficients of cab isolation system; l_i is the distances of the seat, cab, and vehicle (i=1-17); and q_j is the vibration excitation of road (j=1-4)

Based on the dynamic model of the wheel loaders, the equations of motion for the driver's seat, cab, and body are given as follows:

$$\begin{cases} m_s\ddot{z}_s = -F_s = -[k_s(z_s - z_{sb}) + c_s(\dot{z}_s - \dot{z}_{sb})] \\ m_c\ddot{z}_c = F_s - \sum_{i=1}^{4} F_{ci} \\ I_c\ddot{\varphi}_c = \sum_{i=1,2} F_{ci}.l_6 - \sum_{i=3,3} F_{ci}.l_7 - F_s.l_8 \\ J_c\ddot{\theta}_c = \sum_{i=2,3} F_{ci}.l_{15} - \sum_{i=1,4} F_{ci}.l_{14} + F_s.l_9 \\ m_b\ddot{z}_b = \sum_{i=1}^{4} F_{ci} - \sum_{i=1}^{4} F_{ti} - \sum_{i=1,2} F_{gi} \\ I_b\ddot{\varphi}_b = \sum_{i=1,2} F_{ti}.l_2 - \sum_{i=3,4} F_{ti}.l_1 - \sum_{i=1,2} F_{ci}.l_4 - \sum_{i=3,3} F_{ci}.l_5 - \sum_{i=1,2} F_{gi}.l_3 \\ J_b\ddot{\theta}_b = \sum_{i=2,3} F_{ti}.l_{11} - \sum_{i=1,4} F_{ti}.l_{10} + \sum_{i=1,4} F_{ci}.l_{12} - \sum_{i=2,3} F_{ci}.l_{13} + F_{g1}.l_{16} - F_{g1}.l_{17} \end{cases} \quad (1)$$

Where:

$$\begin{aligned} F_{ci} &= k_{ci}(z_{ci} - z_{cbi}) + c_{ci}(\dot{z}_{ci} - \dot{z}_{cbi}) \\ F_{ti} &= k_{ti}(z_{bi} - q_i) + c_{ti}(\dot{z}_{bi} - \dot{q}_i) \end{aligned} \tag{2}$$

In cases where the pitch angle and roll angle are minimal, the vertical displacement of the four endpoints of the cab and body:

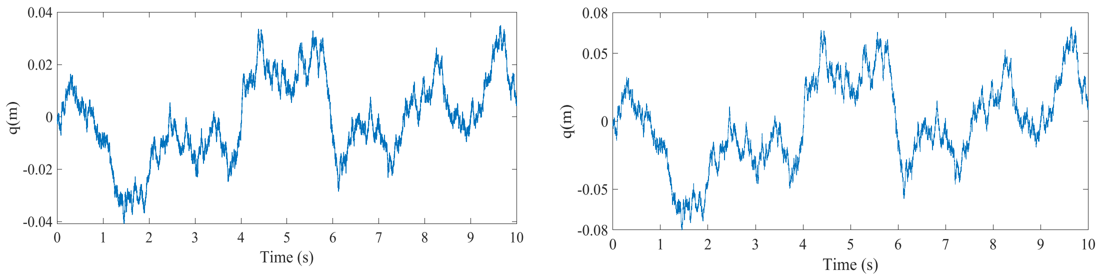
$$\begin{cases} z_{c1} = z_c - l_6.\varphi_c + l_{14}.\theta_c \\ z_{c2} = z_c - l_6.\varphi_c - l_{14}.\theta_c \\ z_{c3} = z_c + l_7.\varphi_c - l_{15}.\theta_c \\ z_{c1} = z_c + l_7.\varphi_c + l_{14}.\theta_c \end{cases} \quad \begin{cases} z_{cb1} = z_b - l_4.\varphi_b + l_{12}.\theta_b \\ z_{cb2} = z_b - l_4.\varphi_b - l_{13}.\theta_b \\ z_{cb3} = z_b - (l_4 + l_5).\varphi_b - l_{13}.\theta_b \\ z_{cb4} = z_b + (l_4 + l_5).\varphi_b + l_{12}.\theta_b \end{cases} \quad \begin{cases} z_{b1} = z_b - l_2.\varphi_b + l_{10}.\theta_b \\ z_{b2} = z_b - l_2.\varphi_b - l_{13}.\theta_b \\ z_{b3} = z_b + l_1.\varphi_b - l_{13}.\theta_b \\ z_{b4} = z_b + l_1.\varphi_b + l_{10}.\theta_b \end{cases} \tag{3}$$

2.2. Vibration excitations

Wheel loaders are versatile vehicles among construction machinery, frequently encountering a wide variety of road surface conditions. However, for this study, the random road surface roughness represented by random white noise is chosen as the excitation source waveform for the wheel loader. The random road profile is generated by filtering the white noise through the following mathematical model of road roughness.

$$\dot{q}(t) = -0.111\left[vq(t) + 40\sqrt{G_q(n_0)V}\,w(t)\right] \tag{4}$$

Where *q(t)* represents the vertical displacement of the random road input, *w(t)* signifies the Gauss white noise, and $G_q(n0)$ denotes the road roughness, *V* is the speed of the vehicle. The results of the typical random road excitation theo are shown in Fig. 2.



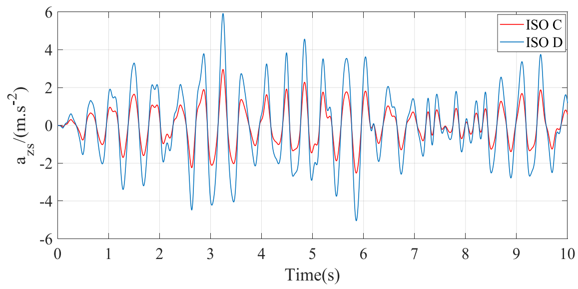
(a) Level C (b) Level D

Fig. 2. *The vibration excitation of the random road surface*

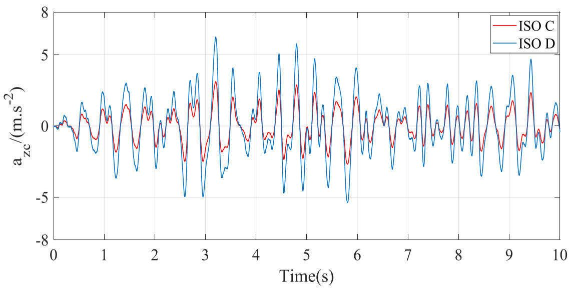
III RESULTS AND DISCUSSION

The motion equations of Eq. (1) are solved by using Matlab/Simulink software with the reference parameters [12, 13]. The acceleration responses of the vertical driver seat, vertical cab, pitching angle, and rolling angle of cab (a_{zs}, a_{zc}, $a_{\varphi c}$, and $a_{\theta c}$) are shown in Fig. 3 when vehicle moves on ISO class C and D road surfacesat vehicle speed of 15 km/h.

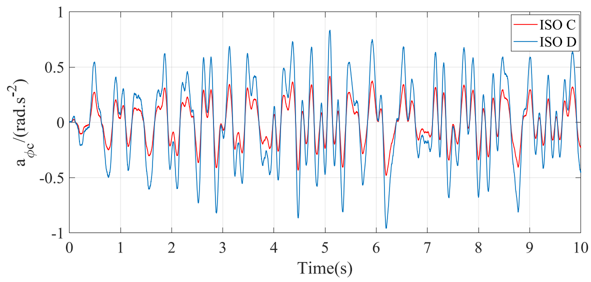
Based on the results shown in Fig. 3, the root mean square (RMS) values a_{zs}, a_{zc}, $a_{\varphi c}$, and $a_{\theta c}$can be evaluated according to ISO standard 2631-1:1997 [14]. When the wheel loader travels on an ISO Class C road surface, the RMS values of a_{zs}, a_{zc}, $a_{\varphi c}$, and $a_{\theta c}$are be determined to be 0.6974 m/s², 0.7754 m/s², 0.2316rad/s², and 0.2155rad/s², respectively.In contrast, when the vehicle operates on an ISO Class D road surface, which represents a rougher terrain, the RMS values for the same parameters increase. Specifically, The RMS values of a_{zs}, a_{zc}, $a_{\varphi c}$, and $a_{\theta c}$are determined to be 1.3944m/s², 1.5504 m/s², 0.4632rad/s², and 0.4308rad/s², respectively. These results indicate a clear increase in vibrational impact on both the cab and driver seat as road roughness escalates from ISO Class C to ISO Class D, meaning that road surface roughness significantly affects the dynamic behavior of the wheel loader. The rougher the surface, the greater the vibrational load experienced by the operator, which can lead to discomfort and potentially contribute to long-term health issues such as lower back pain and fatigue. These findings highlight the crucial role of road surface quality in the vibrational dynamics of heavy machinery like wheel loaders.



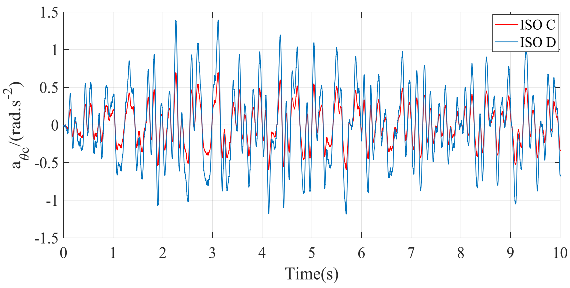
a.*The vertical acceleration of the driver's seat*



b.*The vertical acceleration of the cab*



c.*The pitching acceleration of the cab*



d.*The rolling acceleration of the cab*

Fig. 3. *The results of the acceleration responses under random road level C and D*

IV CONCLUSIONS

In this study, a dynamic model of a wheel loader with 7-DOFisestablished to analyze its vibrational behavior under various road surface conditions. The key conclusions drawn from the evaluation results are as follows: (i) The RMS values of a_{zs}, a_{zc}, $a_{\varphi c}$, and $a_{\theta c}$ increase when the wheel loader operates on an ISO Class D road surface compared to an ISO Class C road. This indicates that rougher terrain amplifies the vibration levels experienced by the loader, directly affecting ride comfort. (ii) For wheel loaders, which lack suspension systems and frequently operate in harsh environments, the implementation of effective cab isolation systems is critical. Specifically, optimizing and developing advanced cab vibration isolation structures is an essential strategy to enhance operator comfort and reduce fatigue during extended operations.

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