Ride Comfort Analysis of Vibratory Roller via Numerical Simulation and Experiment

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Keywords: Vibration roller, Dynamic model, Ride comfort, Computer simulation, Experiment.

Abstract. In this study, based on the analysis of nonlinear geometric characteristics of wheel-deformation of soil ground contact and the weighted r.m.s acceleration responses of the vertical driver’s seat, pitch and roll angle of the cab are chosen as objective functions, a 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. Matlab/Simulink software is used to simulate the nonlinear dynamic models and calculate the objective functions according to the ISO 2631: 1997 (E) standard such as the weighted r.m.s acceleration responses of the vertical driver’s seat, pitch and roll angle of the cab. An experiment was set up to measure ride comfort for vibratory roller when vehicle compacts and moves under four different operating conditions. The numerical simulation results for ride comfort analysis were compared with the experimental results which have verified the validity of models. Ride analysis results are shown that vibratory roller ride comfort is very poor in the most of the operating conditions. The study can provide a basis for the isolation system optimum design of the off-road vehicle.

Introduction

Vibratory Roller often operate in harsh environments, the vibration excitation sources causing vehicle’s body vibration are only the excitations of the interaction between wheel and deformation ground soil, but also are the excitations of vibration drum and engine which is one of the main reasons for the driver fatigue. To evaluate the riding comfort of a vibratory roller under the different soil grounds, the nonlinear dynamics model of the single drum vibratory roller was established based on the analysis of the contact physics of the wheel with different soil grounds. Matlab/Simulink software was used to simulate the nonlinear dynamic models and calculate the values of the vertical weighted r.m.s acceleration responses of driver’s seat and cab. The nonlinear dynamics model of the whole vehicle was analyzed according to the ISO 2631: 1997 (E) standard, the influence of noise and vibration to human health which evaluates the influence of the different road conditions, operating conditions, and vehicle speeds on the driver’s ride comfort [1]. To reduce the effect of vibration to operators, identification and elimination of vibration sources are the most important tasks to achieve optimum design. Dynamic test and analysis of vibration roller, equivalent finite element model building and dynamic simulations are carried out to find out the main reason causing vibratory roller’s sloshing when it moves at low speed down road surface and the auxiliary vibration isolator for cabin to reduce vibration in low frequency range are designed[4]. To improve the ride comfort of impact roller cab, a semi-active cab suspension control model was established according to the vibration characteristics for the impact roller. Two-input single and one-output single fuzzy controller was established and the fuzzy control rules was build too based on experience and theoretical analysis. Taking a certain type of impact roller as an application object, the traditional passive rubber damping system and semi-active control suspension system based on MR damper was simulated and compared. The study results show that ride comfort of cab with semi-active suspension is very obvious[5].

To compare ride comfort of cab with hydraulic mounts between with rubber mounts, a six-degree-of-freedom model of the cab supported by hydraulic mounts with quadratic damping is set
up. And the simulation which compares performance of the hydraulic mounts and the rubber mounts used in the cab is carried out. It shows that the cab system with quadratic damping hydraulic mounts has remarkable efficiency to mitigate the vibrations and in turn to enhance the cab comfort[2]. The ride dynamics of typical North-American soil compactors were investigated via analytical and experimental methods. The 12-DOF in-plane ride dynamic model of a single-drum compactor was formulated through integrations of the models of various components such as driver seat, cabin, roller drum and drum isolators, chassis and the tires [3].

In this paper, the ride comfort of vibratory roller from the numerical simulation was compared to the experimental result. To simulate the ride comfort of cab’s isolation system, a 3D nonlinear dynamic model of a single drum vibratory roller was established which including based on Adam D. and Kopf F’s elastic-plastic soil model [7] and Bekker hypothesis of the soft soil ground [8], a quarter of the wheel- deformation soil surface contact model is established to analyze the vertical excitation force acting on the front and rear frame. Matlab/Simulink software is used to simulate the nonlinear dynamic models and calculate the objective functions such as the weighted r.m.s acceleration responses of the vertical driver’s seat, pitch and roll angle of the cab. To measure ride comforts of vibratory roller, an experiment was carried out when vehicle operates and moves under four different operating conditions. The numerical simulation results for ride comfort analysis were compared with the experimental results and the ride comfort of off-road vehicle is analyzed according to the ISO 2631: 1997 (E) standard [6].

Dynamic Model of Vibratory Roller

A single drum vibratory roller with the rubber isolation systems of drum, cab and seat suspension system is selected for vehicle dynamic analysis. a 3D nonlinear dynamic model are developed based on Adam D. and Kopf F’s elastic-plastic soil model and Bekker hypothesis of the soft soil ground, as shown in Figure 1.

In Figure 1, \( m_d, m_{ff}, m_{fr}, m_c \) and \( m_s \) are the mass of the vibrating drum, frame-front, frame-rear, cab and driver's seat, respectively; \( I_{dx}, I_{ffx}, I_{frx}, I_{cx}, I_{fy}, I_{cy} \) are the moment of inertia with
respect to the x of the vibrating drum, front frame, the moment of inertia with respect to the x and y axes of rear frame and cab, respectively; \( k \) and \( c \) are the stiffness and damping of driver’s seat suspension system; \( k_{c1}, k_{c3}, k_{c2}, k_{c4}, c_{c1}, c_{c3}, c_{c2}, c_{c4} \) are the stiffness and damping of the left and right side of cab’s isolation system, respectively; \( k_{t1}, k_{t2}, c_{t1}, c_{t2} \) are the stiffness and damping of the left and right side of tires, respectively; \( l_{d1}, l_{d2}, c_{d1}, c_{d2} \) are the stiffness and damping of the left and right side of tires, respectively; \( k_{sp1} \) and \( k_{sp2} \) are the plastic stiffness of the soil; \( k_{ac1}, k_{ac2} \) and \( c_{ac1}, c_{ac2} \) are the elastic stiffness and damping of the soil; \( z_d, z_{ff}, z_f, z_c \) and \( z_s \) are the vertical displacements at centre of gravity of the drum, the frame-front, the frame-rear, cab and driver’s seat, respectively; \( \theta_d, \theta_{ff}, \theta_f, \theta_c \) and \( \theta_s \) are the roll angle displacements of the drum, the frame-front, the frame-rear and cab, respectively; \( \varphi_f \) and \( \varphi_c \) are the pitch angle displacements of the frame-rear and cab, respectively; \( q_{d1}, q_{t1} \) and \( q_{d2}, q_{t2} \) are the left and right excitation of road surface roughness, respectively; \( z_{ac1} \) and \( z_{ac2} \) are the left and right deformation of elastic soil; \( z_{t1} \) and \( z_{t2} \) are the vertical displacement of the lowest point of the left and right tires, respectively; \( l_1, l_2, l, i_1, i_2, l, l_{cf}, l, b_{ac1}, b_{ac2}, b_{d1}, b_{d2}, b_{t1}, b_{t2}, b_{c1}, b_{c2}, b_i \) are the distances; \( F = F_0 \sin(\omega t) \) is the force excitation of the vibrating drum; \( F_0 \) is the amplitude of force excitation; \( \omega \) is the angular frequency of the vibrator; \( e \) is the eccentricity of the rotating mass; \( F_p \) and \( M_{p1}, M_{p2} \) are the coupling force in the vertical direction and the coupling moments in the front-rear, left-right direction at the point of intersection, respectively; \( v \) is the vehicle speed.

To set the differential equation describing the motion of objects, a combined method of the multi-body system theory and D’Alembert’s principle is chosen in this paper. 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 set up force and moment equations to describe the differential equations of motion for the front frame, rear frame, cab and driver’s seat of the vehicle subjects such as

The differential equations of motion for front frame are formulated as follows

\[
m_{ff} \ddot{z}_{ff} = -\sum_{i=1}^{2} F_{di} \tag{1}
\]

\[
I_{fx} \ddot{\theta}_{ff} = F_{d2}b_{d2} - F_{d1}b_{d1} \tag{2}
\]

The differential equations of motion for rear frame are formulated as follows

\[
m_{fr} \ddot{z}_{fr} = \sum_{i=1}^{4} F_{ci} - \sum_{i=1}^{2} F_{ni} - F_p \tag{3}
\]

\[
I_{fr} \ddot{\theta}_{fr} = \sum_{i=1}^{2} F_{c1}l_{c1} + \sum_{i=1}^{2} F_{c1}l_{cf} + \sum_{i=2}^{3} F_{c1}l_{ef} - F_p l_{t1} - M_{p1} \tag{4}
\]

\[
I_{fr} \ddot{\theta}_{fr} = \sum_{i=1}^{3} (-1)^{i} F_{c1}l_{c1} + \sum_{i=1}^{3} (-1)^{i} F_{c1}l_{d1} + \sum_{i=1}^{3} F_{c1}b_{c1} - \sum_{i=2}^{3} F_{c2}b_{c2} - M_{p2} \tag{5}
\]

The differential equations of motion for cab are formulated as follows

\[
m_{ce} \ddot{z}_{ce} = F_i - \sum_{i=1}^{4} F_{ci} \tag{6}
\]
The differential equation of motion for driver’s seat is formulated as follows:

\[ m_s \ddot{z}_s = -F_s \]  

(9)

The coupling force in the vertical direction and the coupling moments in the front-rear, left-right direction at the point of intersection are formulated as:

\[ F_p = \sum_{i=1}^{3} F_{di} \]  

(10)

\[ M_{p1} = \sum_{i=1}^{3} F_{di} l_d \]  

(11)

\[ M_{p2} = F_d b_{d2} - F_d b_{d1} \]  

(12)

where, \( F_{c1}, F_{c2}, F_{c3}, F_{c4} \) and \( F_c \) are the vertical reaction forces of cab’s isolation system and driver’s seat suspension system; \( F_{d1}, F_{d2} \) and \( F_{d3} \) are the vertical reaction forces of the right- and left-tires and right- and left-vibrating drum and it will be determined based on the analysis of nonlinear geometric characteristics of wheel-deformation of soil ground contact in the section below.

**Wheel-deformed Soil Ground Contact Model[1]**

**Tires-elasto and Plastic Soil Ground Contact Model**

When vibratory roller moves and operates elastic-plastic soil surface which becomes elastic after each compaction, and the rear tires are gradually changed from elastic-plastic to elastic soil surface. In this paper, the tire-deformation soil surface contact model was established based on Bekker’s hypothesis of the soft soil ground to analyze the vertical excitation force acting on the rear frame. Tires-elasto and plastic soil ground contact model is shown in Figure 2.

![Figure 2. Tires-elasto and plastic soil ground contact model.](image)

Loading phase, the vertical reaction force of tire \( F_{g1} \) can be determined as
\[ F_{g1} = B \left[ \int_{0}^{\theta} \rho(\theta) R \cos \theta d\theta + \int_{0}^{\theta} \tau(\theta) R \sin \theta d\theta \right] \]  

Unloading phase, the vertical reaction force of tire \( F_{g2} \) can be determined as

\[ F_{g2} = B \left[ \int_{0}^{\theta} \rho(\theta) R \cos \theta d\theta + \int_{0}^{\theta} \tau(\theta) R \sin \theta d\theta \right] \]  

The total vertical reaction force of the tire-deformation soil surface contact is defined as

\[ F_g = F_{g1} + F_{g2} \]  

After obtaining Eq. (15), the equations of the contact between the left and right tire and the deformed soil are defined as

\[
\begin{aligned}
F_{l1} - F_{g1} + m_{1}g &= 0 \\
F_{l2} - F_{g2} + m_{2}g &= 0
\end{aligned}
\]  

where, \( m_{g} \) is the mass of tire; \( F_{g1} \) and \( F_{g2} \) is the vertical reaction forces of the left and right tire-deformation soil surface contact.

**Vibrating Drum and Elastic-plastic Soil Ground Contact Model**

In order to describe the vertical excitation force acting on the front frame generated by a vibratory drum with elastic-plastic soil, a 3-DOF vibration model is established based on Adam D. and Kopf F’s elastic-plastic soil ground, as shown in Figure 3.

The motion of the vibratory drum on a soil patch of given density may exhibit, over each cycle of the drum vibrator, two or more often three distinct phases, which are described below:

1. **Loading phase**

   \[ (m_{f} + m_{d} \dot{z}) g \geq F_{s_{max}} \text{ and } \dot{z}_{d} \geq 0 \]

   In this case, the vibrating drum moves downward in the vertical direction and a compressive load is applied to the soil. Both the plastic and elastic stiffness of soil surface \( k_{sp} \) and \( k_{se} \) are increased while the elastic damping of the soil surface \( c_{se} \) is decreased and the soil layers becomes more elastic. The third-order differential equation describing the vertical motion of the vibrating drum is given by Ario Kordestansi et al. [3] as follows:

\[
\varepsilon m_{d} \ddot{z}_{d} + m_{d} \dot{z}_{d} = \varepsilon F_{d} + F_{d} - k_{sp} z_{d} + k_{sp} z_{d} + k_{se} z_{d} + k_{se} z_{d} + \varepsilon F_{0} \cos \alpha x + F_{0} \sin \alpha x
\]  

where, \( A \) damD and Kopf.F have proposed two ratios below

The plasticity ratio \( \varepsilon \) is given by
The damping to plasticity ratio $\gamma$ is given by

$$\gamma = \frac{c_{se}}{k_{sp}}$$  \hspace{1cm} (19)

(2) Unloading phase $(m_f + m_d)g \geq F_{s\text{max}}$ and $\dot{z}_d < 0$

In this case, the vibrating drum moves upward while a contact between the vibrating drum and soil surface is retained. The differential equation describing the vertical motion of the vibrating drum is given by

$$m_d \ddot{z}_d = F_d - k_{se} z_d - c_{se} \dot{z}_d + F_0 \sin \omega t$$ \hspace{1cm} (20)

(3) Drum-Hop phase $F_d=0$

In this case, the drum with its continued upward motion tends to lose contact with the soil surface. The differential equation describing the vertical motion of the vibrating drum is given by

$$m_d \ddot{z}_d = F_0 \sin \omega t$$ \hspace{1cm} (21)

After obtaining Eq. (17), Eq.(20) and Eq.(21), the vertical reaction forces of the right- and left-vibrating drum ($F_{d1}$ and $F_{d2}$) can also be determined through Eq.(10), Eq.(11) and Eq.(12).

Experiments and Simulations

In this section, a single drum vibratory roller ride experiment was carried out when vehicle compacts and moves on the elastic-plastic soil ground at a speed of $v=3\text{km/h}$ according to ISO 2631: 1997 (E) standard, the influence of noise and vibration to human health which analyzes the influence of the different operating conditions on the driver’s ride comfort. Belgim LMS dynamic test and analysis system with ICP® three-direction acceleration sensors were used to measure acceleration vibration, the multi-point measurement method for vehicle ride comfort analysis are used for analysis and comparison. Arrangement of measuring points and acceleration sensors are shown as Figure 4.

![Figure 4. Arrangement of measuring points and acceleration sensors.](image-url)

In this paper, the acceleration responses of the pitch and roll angles of the cab were derived from the measured vertical accelerations using kinematic relations, while assuming small angular motions and negligible contribution due to structure flexibility, such that

The acceleration of cab pitch angle is given by
The acceleration of cab roll angle is given by

$$\ddot{\phi}_c = \frac{\ddot{z}_3 - \ddot{z}_4}{L_c}$$  \hfill (22)

where, $L_c$, $B_c$ are the distance between measurement points.

To solve the nonlinear differential equations presented in this paper for the ride comfort analysis of vibratory roller, Matlab/Simulink software is used with a set of parameters of the single drum vibratory roller and elastic-plastic soil ground by the references[3,9]. Simulation of vibratory roller ride comfort carried out under four different conditions such as low/high frequency vibration (28Hz/35Hz) when vibratory compacts on original place (case 1 & case 2), low/high frequency vibration when vibratory compacts and moves on the elastic-plastic soil ground at the speed of $v$=3km/h, (case 3 & case 4). The simulation results are compared with the experimental results and the acceleration responses and power spectral density (PSD) acceleration responses of the vertical driver’s seat, pitch and roll angles of cab are compared with the measured accelerations, case 1 results are shown in Figure 5.

$$\ddot{\theta}_c = \frac{\ddot{z}_1 - \ddot{z}_3}{B_c}$$  \hfill (23)

Figure 5. The acceleration and PSD acceleration responses of the vertical driver’s seat, pitch and roll angles of cab are compared with the measured data.

Comparison of simulation and experimental results from Figure 5 are shown that (1) the peak value range and the variation law of the time domain response curve are basically the same; (2) the vibration energy distribution on the frequency response curve, the peak of the power spectrum and the peak frequency are similar. It is proved that the mathematical model is correct and feasible.
The values of the weighted r.m.s acceleration responses of the vertical driver's seat, pitch and roll angle of the cab ($a_{wza}$, $a_{wc}$, and $a_{wa}$) are determined according to ISO 2631: 1997 (E) standard and compared with the corresponding measured values, as shown in Table 1.

<table>
<thead>
<tr>
<th>Cases</th>
<th>$a_{wza}$ (m/s²)</th>
<th>$a_{wc}$ (rad/s²)</th>
<th>$a_{wa}$ (rad/s²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Simulation</td>
<td>Measured</td>
<td>Deviation</td>
</tr>
<tr>
<td>1</td>
<td>0.8966</td>
<td>0.9740</td>
<td>7.95%</td>
</tr>
<tr>
<td>2</td>
<td>1.1066</td>
<td>1.2490</td>
<td>11.40%</td>
</tr>
<tr>
<td>3</td>
<td>0.7498</td>
<td>0.7992</td>
<td>6.18%</td>
</tr>
<tr>
<td>4</td>
<td>0.9961</td>
<td>1.1120</td>
<td>10.42%</td>
</tr>
</tbody>
</table>

The results presented in Table 1 suggest good correlations between the values of the model and the measured data. The maximum deviation between the simulation and measured results are shown that the weighted r.m.s acceleration responses of the vertical driver's seat is 11.40%, the weighted r.m.s acceleration responses of pitch angle of cab is 12.03%, and the weighted r.m.s acceleration responses of roll angle of cab is about 12.21%. From the results in Table 1 and ISO 2631: 1997 (E) standard, both model and measured values are shown that a driver subjectively feels is very uncomfortable when the vehicle operates and moves under four different conditions. And in a low frequency region, vehicle ride comfort becomes worse when there are vibratory compacts and moves on plastic soil ground.

Conclusions

In this study, a 3D nonlinear dynamic model of a single drum vibratory roller was developed for the numerical simulation based on Adam D. and Kopf F’s elastic-plastic soil model and Bekker hypothesis of the soft soil ground. The numerical simulation results for ride comfort analysis were compared with the experimental results. The numerical results are proved that the mathematical model is correct and feasible and are presented suggest good correlations between the values of the model and the measured data. Finally, both model and measured values are shown that vibratory roller ride comfort is very poor in the most of the operating conditions, and the study can provide a basis for isolation system optimum design of the off-road vehicle in futures studies.

Acknowledgement

This research was financially supported by Jiangsu province research Project, China (No. BY2011151); Jiangsu province science, technology support program, China (No. BE2010047) and Thai Nguyen University research project, Viet Nam (DH2015-TN02-01).

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