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Effect of Damping Coefficient of Drum's Isolation System on Ride Comfort of a Vibratory Roller

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Abstract –*The effect of the values of the damping coefficient of drum's isolation system (c_d) on ride comfort of a vibratory roller presents is presented in this study. The c_d values are respectively analyzed* to *find out the effect on vehicle ride comfort according to the ISO 2631:1997(E). The results show that the effect of the c_d values increases, the ride comfort of a vibratory roller is significantly improved when vehicle operates at different conditions. In addition, the study results are the theoretical basis for the optimal design of drum's isolation system.*

Key Words: Vibratory roller, drum, isolation system, Ride comfort.

1. INTRODUCTION

The earth-moving machinery operates, the vibration excitation sources, such as soil ground, actuators 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 low-frequency advantages and characteristics of the hydraulic mounts were used for vibration isolation of an earth-moving machinery cab and the results indicate that the cab system with quadratic damping hydraulic mounts has remarkable efficiency in mitigating the vibrations and in turn in enhancing the cab comfort; however its nonlinear damping characteristic has almost no effect on the natural frequencies of the cab system[1]. The effects of the design parameters of drum's metal rubber isolation systems on ride comfort were analyzed by using tandem vibratory roller dynamic model with 7 degrees of freedom(DOF)[2]. The stiffness and damping coefficients of the cab's isolation system were analyzed by 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[3]. The riding comfort ofa vibratory roller under the different soil grounds was evaluated by using a nonlinear dynamics model of a single drum vibratory roller, according to the ISO 2631:1997 (E) standard [4]. An experiment was set up to measure ride comfort for vibratory roller when vehicle compacted and moved under four different operating conditions to compare with simulation results[5]. The isolation systems of vibratory roller were considered to optimize design to improve the vehicle ride comfort such as the optimal design parameters of drum's isolation systems for a double-drum vibrating roller were found out using genetic algorithm (GA)[7]; the optimal design parameters of cab's isolation system were found out using genetic algorithm NSGA-II[6],[8]; the design parameters of cab's main isolation system were optimized to reach the maximum value of the first-order natural frequency in order to avoid resonance vibration for cab at low frequency and reduce cab's low-frequency shaking[9] and cab's auxiliary isolation system were found out using the finite element method (FEM)[10].

The main objective of this study is to evaluate the effect of the values of the damping coefficient of drum's isolation system on vehicle ride comfort. A quarter-vehicle dynamic model of vibratory is established for analysis and evaluation. The weighted r.m.s acceleration response of the vertical driver's seat (a_{wb}) according to the ISO 2631:1997(E) standard[11] is chosen as objective function. The values of the damping coefficient are respectively analyzed to examine the effect on the value of objective function.

2. VEHICLE DYNAMIC MODEL

A quarter-vehicle dynamic model of vibratory roller is established to see the effect of the values of the damping coefficient of drum's isolation system on vehicle ride comfort, as shown in Fig-1.

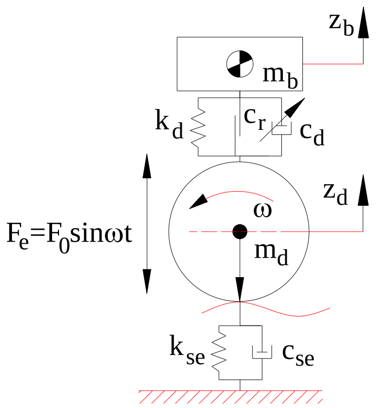


Fig-1: Quarter-vehicle dynamic model

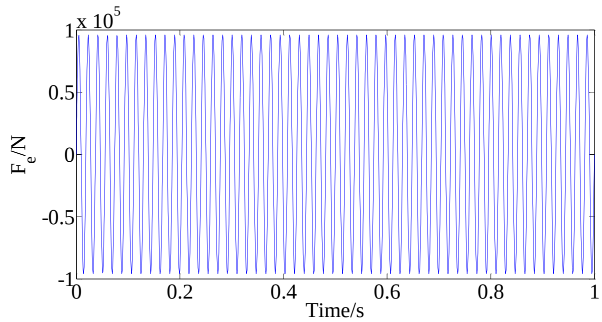
In Fig-1, m_d and m_b are drum mass and vehicle body mass; k_d and c_d are the stiffness and damping coefficients of drum's isolation system; k_{se} and c_{se} are the stiffness and damping coefficients of elastic soil ground; z_b *and* z_b are the vertical displacements of drum mass and vehicle body mass; and $F=F_0 sin(\omega t)$ is the force excitation of the vibrating drum; F_0 is the amplitude of force excitation; ω is the angular frequency of the vibrator; e is the eccentricity of the rotating mass.

The equations of motion for a quarter-vehicle dynamic model of vibratory roller using Newton's second law of motion are written as follows:

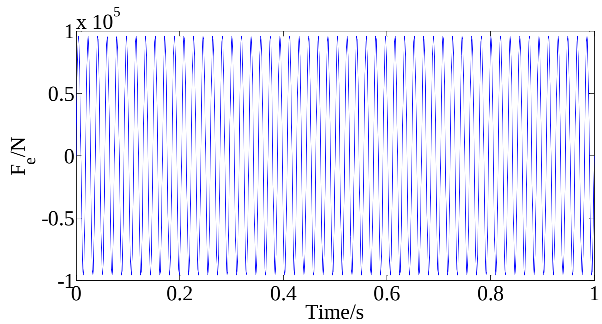
$$m_b \ddot{z}_b = k_d(z_b - z_d) + c_r(\dot{z}_b - \dot{z}_d) + c_d(\dot{z}_b - \dot{z}_d) + F_0 \sin \omega t - k_{se} z_d - c_{se} \dot{z}_d \quad (1)$$

$$m_b \ddot{z}_b = -k_d(z_b - z_d) - c_d(\dot{z}_b - \dot{z}_d) \quad (2)$$

The vertical excitation force for vibration drum is shown in Fig-2 at F_0=280000N, f=30Hz and F_0=190000N, f=35 Hz for drum



(a) At F_0=280000 N and f=30Hz



(b) At F_0=190000N and f=30Hz

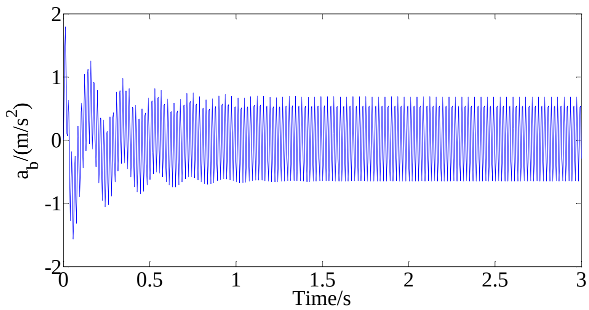
Fig- 2: Vertical excitation force for vibration drum.

3. SIMULATION AND DISCUSSION

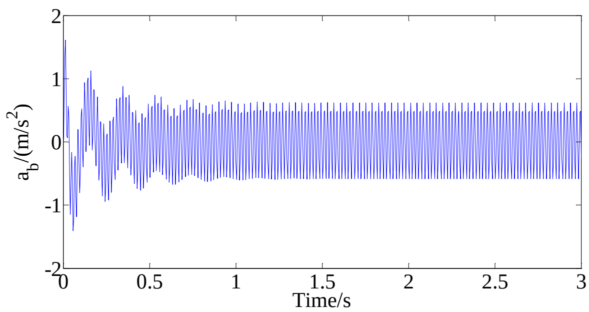
In order to solve the general dynamic differential equation of a vibratory roller presented in section 2, Matlab/Simulink software is used with a set of parameters of vehicle in Tab.1. The simulation results of the time domain acceleration responses of the vertical vehicle body (a_b) when vehicle compacts on original place at the low excitation frequency for drum (F_0=280000N và f=30Hz) and at the low excitation frequency for drum (F_{02}=190000 và f=35Hz) are shown in Fig.3

Table- 2: Parameters of a vibratory roller

Parameters	Values
m_d/kg	4.378 x10^3
m_b/kg	2.822 x10^3
c_r/ (N.s/m)	2.900x10^3
k_d/ (N/m)	3.98x10^6
c_{se}/(N.s/m)	21x10^4
k_{se}/ (N/m)	10x10^6
F_0/N	(0.28/0.19)x10^6
f/Hz	30/35



(a) At F_0=280000 N and f=30Hz



(b) At F_0=190000N and f=30Hz

Fig- 3: Time domain acceleration responses of the vertical vehicle body

From the results of the acceleration responses (in Fig.3), we could determine the values of the weighted r.m.s acceleration of the vehicle body according to the ISO 2631:1997(E)[11] such as a_{wb}= 0.5144 m/s^2 at F_0=280000 N and f=30Hz and a_{wb}=0.4630 m/s^2 at F_0=190000N and f=30Hz. The a_{wb} value at the low excitation frequency for drum reduces by 11.10% in comparing with at the high excitation frequency for drum. The vehicle ride comfort is improved.

In order to analyze the effect of the values of the damping coefficient of drum's isolation system (c_d) on ride comfort of a vibratory roller, the c_d values such as c_d=1.0c_r, c_d=2c_r and c_d=3c_r are selected for analysis. The simulation results of the time domain acceleration responses of the vertical vehicle body (a_b) when vehicle compacts on original place at the low excitation frequency for drum (F_0=280000N và f=30Hz) is shown in Fig.4

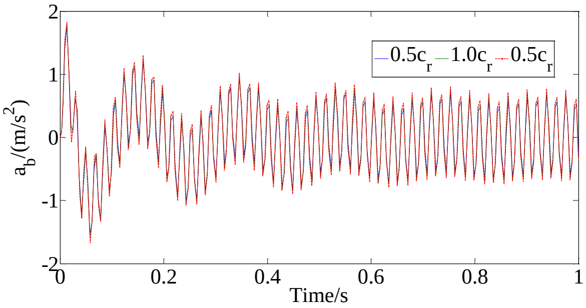


Fig- 4: Time domain acceleration responses of the vertical vehicle body with three c_d values

From the results in Fig.4, we could determine the values of the values of the weighted r.m.s acceleration of the vehicle body with $0.5c_r$, $1.0c_r$ and $1.5c_r$ such as 0.4837 m/s^2, 0.5144 m/s^2, and 0.5469 m/s^2. The increases in both c_d and a_{wb} values makes vehicle ride comfort reduces and conversely, vehicle ride comfort improves.

4. CONCLUSION

The study focuses on effect of the values of the damping coefficient of drum's isolation system (c_d) on ride comfort of a vibratory roller. A quarter-vehicle dynamic model of vibratory is established for analysis and evaluation. The weighted r.m.s acceleration responses of the vertical driver's seat (a_{ws}) according to the ISO 2631:1997(E) standard[11] is chosen as objective function. The major conclusions can be drawn from the analysis and evaluation results as follows: (1) The a_{wb} value at the low excitation frequency for drum reduces by 11.10% in comparison with at the high excitation frequency for drum. The vehicle ride comfort improves and (2) The cd and awb values increases leads to the reduction in vehicle ride comfort and conversely improvement in vehicle ride comfort.

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