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Effect of Drum's Excitation Frequency on Ride Comfort of a Single –Drum Vibratory Roller

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***Abstract*— This study proposes a half-vehicle ride dynamic model of a single-drum vibratory roller interacting with the under the ground surface deformation to analyze the effects of drum's excitation frequency on vehicle ride. The excitation frequencies for a dynamic drum affect not only soil compression efficiency, but also vehicle ride comfort. In order to evaluate the effect on vehicle ride comfort, the weighted r.m.s acceleration responses of the vertical driver's seat and pitch angle of the cab according to the ISO 2631:1997(E) standard are chosen as objective functions. The excitation frequencies for a dynamic drum are respectively analyzed based on objective functions. The results show that the effects of drum's excitation frequency on vehicle ride are very obvious especially at the range of the low excitation frequency values for a dynamic drum. The results of this are the theoretical basis for designing the dynamic systems of a single-drum vibratory roller.**

***Index Terms*— vibratory roller, drum, excitation frequency, dynamic model, ride vibration**

I. INTRODUCTION

There have been many researches on the effect of design parameters of drum's isolation system and cab's isolation system cushions on vehicle ride comfort. The design parameters of the drum's isolation system were analyzed by a tandem vibratory roller dynamic model with 7 degrees of freedom (DOF) with different operations of the vehicleto see its effects on vehicle ride comfort[1]. By using a 12 –DOF in-plane ride dynamic model of a single-drum compactor, the design parameters of both drum's isolation system and cab's isolation system were investigated to find the influences of different design parameters on the whole body vibration responses [2]. The different design parameters of cab's isolation system were examined for its effects on vehicle ride comfort by using a 3-D nonlinear dynamic model of a single drum vibratory roller[3] and comparing with experimental results[18]. Three types of cab's isolation systems such as the traditional rubber, the hydraulic and pneumatic isolation systems were analyzed and compared to investigate its effectiveness on vehicle ride comfort using a 3-D nonlinear dynamic model of the off-road vibratory roller[4]. To improve vehicle ride comfort, the design parameters of drum's isolation systems of a double-drum vibrating roller were analyzed and optimized by using a genetic algorithm[5], those of cab's isolation system of a single-drum vibrating roller were done by using a multi-objective genetic algorithm (GA)[6]. In order to reduce the cab's low-frequency shaking while vehicle operates under the different conditions, the design parameters of cab's main isolation system[7] and cab's auxiliary isolation system[8] were optimized by using the finite element method (FEM). The design parameters of cab's main isolation system were optimized by using FEM and GA[11] . The control methods for drum's isolation system and cab's isolation system such as the control strategy uses Fuzzy control[9] and a combined control method of Fuzzy and PID control[10] were proposed to improve the ride comfort of the vibratory rollers.

The effect of excitation frequencies of drum on the plastic deformation of soils was analyzed by Ario Kordestani[12], the influence of vibration frequency on the compaction of the deformation of soils was examined by C Wersäll [13]. On the other side, the influences of various kinds of soil ground on the driver's health and their working efficiency was figured out by Le V. Q, et al[14].

In this study, the excitation frequencies for a dynamic drum of a single –drum vibratory roller are respectively analyzed to recognize its effect on vehicle ride comfort. A half-vehicle ride dynamic model is established under the ground surface deformation. The weighted r.m.s acceleration responses of the vertical driver's seat and pitch angle of the cab according to the ISO 2631:1997(E) standard[16] are chosen as the objective functions. Matlab/Simulink software is used to simulate the vehicle dynamic model and calculate the values of the objective function.

II. VEHICLE DYNAMIC MODEL

A. *Half-vehicle ride dynamic model*

To analyze the effect of the excitation frequencies for a dynamic drum of a single –drum vibratory roller on the values of the objective functions according to the ISO 2631:1997(E) standard, a half-vehicle ride dynamic model is established under the ground surface deformation, as shown in Fig.1.

In Fig-1, m_d, m_{ff}, m_{fr}, m_c and m_s are the mass of the dynamic drum, front vehicle body (frame), rear vehicle body, cab and driver's seat, respectively; I_{ff} , I_{fr}, *and* I_c are the moment of inertia of the front vehicle body, and cab, respectively; k_s and c_s are the stiffness and damping of driver's seat suspension system; k_{cf} and c_{cr} are the stiffness and damping of the front and rear cab's isolation systems, respectively; k_t and c_t are the stiffness and damping of the tire; k_d and c_d are the stiffness and damping of the drum's isolation system, respectively; z_d, z_{ff}, z_{fr}, z_c and z_s are the vertical displacements at centre of gravity of the drum, the front vehicle body, the rear vehicle body, cab and driver's seat, respectively; $\varphi_{ff,}$ φ_{fr} and φ_c are the

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pitch angle displacements of the front vehicle body and rear vehicle body and cab, respectively; q_t and q_d are the left and right excitation of road surface roughness at drum and tire, respectively; l_t, l_d, l_{t1},l_{t2}, l_{c1}, l_{c1}, l_{cf}, l_{fr}, l_s 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; ω is the angular frequency of the vibrator; e is the eccentricity of the rotating mass; F_p and M_p are the coupling force in the vertical direction and the coupling moments in the front direction at the point of intersection, respectively; v is the vehicle speed.

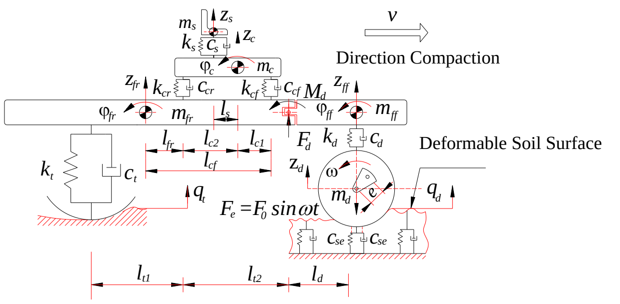


Figure 1. Half-vehicle ride dynamic model

B. Equations of motion

For the dynamic model showed in Fig-1, 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 vehicle dynamic subsystems. The equations of motion can be written as

The equation of motion for seat vertical motion is written as follows

$$m_s\ddot{z}_s = -[k_s(z_s - z_c + l_s\varphi_c) + c_s(\dot{z}_s - \dot{z}_c + l_s\dot{\varphi}_c].\qquad(1)$$

The equations of motion for the vertical and pitch motions of cab are written as follows

$$\begin{aligned} m_c\ddot{z}_c &= [k_s(z_s - z_c + l_s\varphi_c) + c_s(\dot{z}_s - \dot{z}_c + l_s.\dot{\varphi}_c)] \\ &-[k_{cf}(z_c + l_{c1}\varphi_c - z_{fr} - l_{fr}\varphi_c) + c_{cf}(\dot{z}_c + l_{c1}\dot{\varphi}_c - \dot{z}_{fr} - l_{fr}\dot{\varphi}_c)] \\ &-[k_{cr}(z_c - l_{c2}\varphi_c - z_{fr} - l_{cf}\varphi_c) + c_{cr}(\dot{z}_c - l_{c2}\dot{\varphi}_c - \dot{z}_{fr} - l_{cf}\dot{\varphi}_c)] \end{aligned}$$

(2)

$$\begin{aligned} I_c\varphi_c &= -[k_s(z_s - z_c + l_s\varphi_c) + c_s(\dot{z}_s - \dot{z}_c + l_s.\dot{\varphi}_c)]l_s \\ &-[k_{cf}(z_c + l_{c1}\varphi_c - z_{fr} - l_{fr}\varphi_c) + c_{cf}(\dot{z}_c + l_{c1}\dot{\varphi}_c - \dot{z}_{fr} - l_{fr}\dot{\varphi}_c)]l_{c1} \\ &+[k_{cr}(z_c - l_{c2}\varphi_c - z_{fr} - l_{cf}\varphi_c) + c_{cr}(\dot{z}_c - l_{c2}\dot{\varphi}_c - \dot{z}_{fr} - l_{cf}\dot{\varphi}_c)]l_{c2} \end{aligned}$$

(3)

The coupling force in the vertical direction and the coupling moments in the front direction at the point of intersection are formulated as

$$F_d = [k_d(z_{ff} - z_d) + c_d(\dot{z}_{ff} - \dot{z}_d].\qquad(4)$$

$$M_d = [k_d(z_{ff} - z_d) + c_d(\dot{z}_{ff} - \dot{z}_d]l_d.\qquad(5)$$

The equation of motion for the vertical and pitch motions of the rear vehicle body are written as follows

$$\begin{aligned} m_{fr}\ddot{z}_{fr} &= [k_{cf}(z_c + l_{c1}\varphi_c - z_{fr} - l_{fr}\varphi_c) + c_{cf}(\dot{z}_c + l_{c1}\dot{\varphi}_c - \dot{z}_{fr} - l_{fr}\dot{\varphi}_c)] \\ &+[k_{cr}(z_c - l_{c2}\varphi_c - z_{fr} - l_{cf}\varphi_c) + c_{cr}(\dot{z}_c - l_{c2}\dot{\varphi}_c - \dot{z}_{fr} - l_{cf}\dot{\varphi}_c)] \\ &-[k_t(z_{fr} - l_{t2}\varphi_{fr} - q_t) + c_t(\dot{z}_{fr} - l_{t2}\dot{\varphi}_{fr} - \dot{q}_t)]\text{-}F_d \end{aligned}$$

(6)

$$\begin{aligned} I_{fr}\ddot{z}_{fr} &= [k_{cf}(z_c + l_{c1}\varphi_c - z_{fr} - l_{fr}\varphi_c) + c_{cf}(\dot{z}_c + l_{c1}\dot{\varphi}_c - \dot{z}_{fr} - l_{fr}\dot{\varphi}_c)]l_{fr} \\ &+[k_{cr}(z_c - l_{c2}\varphi_c - z_{fr} - l_{cf}\varphi_c) + c_{cr}(\dot{z}_c - l_{c2}\dot{\varphi}_c - \dot{z}_{fr} - l_{cf}\dot{\varphi}_c)]l_{cf} \\ &-[k_t(z_{fr} - l_{t2}\varphi_{fr} - q_t) + c_t(\dot{z}_{fr} - l_{t2}\dot{\varphi}_{fr} - \dot{q}_t)]l_{t2}\text{-}F_d l_{t1} - M_d \end{aligned}$$

(7)

The equation of motion for the dynamic drum is written as follows

$$m_d\ddot{z}_d = m_e e(2\pi f)^2 \sin 2\pi ft + F_d - k_{se}z_d - c_{se}\dot{z}_d\qquad(8)$$

III. VEHICLE RIDE COMFORT CRITERIA[1],[3],[15]

A number of methods can be applied to evaluate the vehicle ride comfort, for examples, frequency-domain and time-domain methods.

$$a_w = \left[\frac{1}{T}\int_0^T a_w^2(t)dt \right]^{1/2}\qquad(9)$$

In this formula, $a_w(t)$ 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 likely reactions to various magnitudes of overall vibration in the public transport and vehicle, a synthetic index-called weighted r.m.s acceleration, aw can be calculated from formula Eq.(9); besides, the r.m.s. value of the acceleration in vehicle would be compared with the values in Table1.

Table 1. Comfort levels related to a_w threshold values[16]

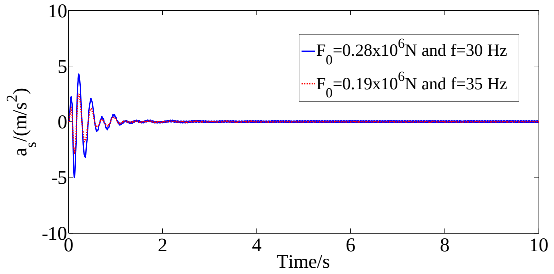
$a_w/(m.s^2)$	Comfort level
< 0.315	Not uncomfortable
0.315÷0.63	A little uncomfortable
0.5 ÷ 1.0	Fairly uncomfortable
0.8 ÷ 1.6	Uncomfortable
1.25 ÷ 2.5	Very uncomfortable
> 2	Extremely uncomfortable

IV. SIMULATION AND DISCUSSION

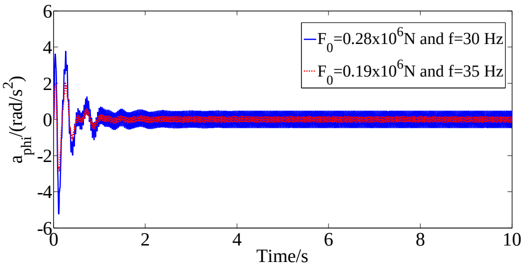
In order to criticise the effect of the excitation frequencies for a dynamic drum on vehicle ride comfort, Matlab/simulink software is applied to solve the differential equations in Part II with a set of simulation parameters of a single –drum vibratory roller[11] when vehicle compacts under two cases including the compacts on original place (Case 1), and compacts and moves on the elastic soil ground and rear wheel moves on the road surface conditions as ISO level E according to the International Standards Organization (ISO) 8608 [17] at the speed of v=3km/h (Case 2).

The acceleration responses of the vertical driver's seat and pitch angle of the cab when vehicle compacts in Case 1 and Case 2 at low and high excitation frequencies for a drum (f=30Hz and f=35Hz) are shown Fig.2 and Fig.3.

From the results of Fig.2a, we can determine the values of the weighted rms acceleration of the vertical driver's seat (a_{ws}) in Case 1 at low and high excitation frequencies for a drum ($F_0=0.28x10^6N$, f=30Hz and $F_0=0.19x10^6N$, f=35Hz) are 0.3492 m/s^2 and 0.6418 m/s^2. The a_{ws} value at the high excitation frequency for a drum $F_0=0.19x10^6N$ and f=35Hz reduce by 83.79% in comparison with at the low excitation frequency for a drum $F_0=0.28x10^6N$, f=30Hz. Those values of a_{ws} are the uncomfortable conditions for driver comfort (according to Tab.1). Vehicle's ride comfort is improved significantly when the vehicle compacts in Case 1 at the excitation high frequency for a drum.



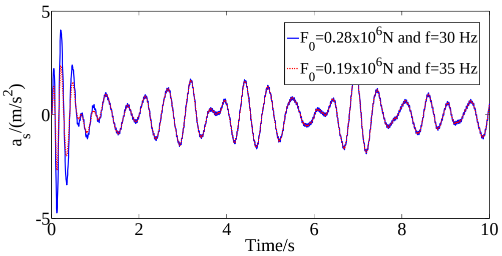
(a) Vertical driver's seat



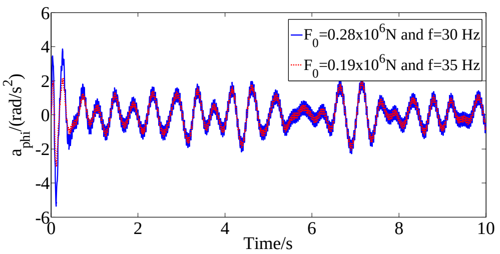
(b) Pitch angle of the cab

Figure2. The acceleration responses in Case 1

From the results of Fig.2b, we can determine the values of the weighted rms acceleration of pitch angle of cab (a_{wphi}) in Case 1 at low and high excitation frequencies for a drum (F_0=0.28x10^{6}N, f=30Hz and F_0=0.19x10^{6}N, f=35Hz) are 0.3535 rad/s^2 and 0.6742 rad/s^2. The a_{wphi} value at the excitation high frequency for a drum F_0=0.19x10^{6}N and f=35Hz reduce by 90.72% in comparison with at the excitation low frequency for a drum F_0=0.28x10^{6}N, f=30Hz. Cab's pitch angle of cab significantly is reduce by the excitation high frequency for a drum.



(a) Vertical driver's seat

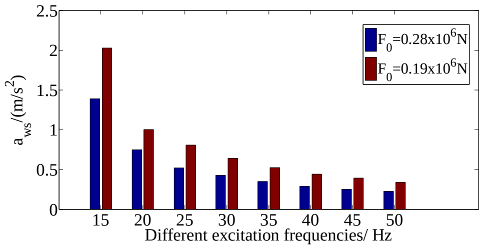


(b) Pitch angle of the cab

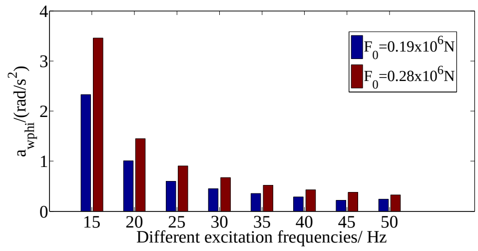
Figure 3. The acceleration responses in Case 2

From the results of Fig.3a, we can obtain the values of the weighted rms acceleration of the vertical driver's seat (a_{ws}) in Case 2 at low and high excitation frequencies for a drum (F_0=0.28x10^{6}N, f=30Hz and F_0=0.19x10^{6}N, f=35Hz) are 0.8013 m/s^2 and 0.9393 m/s^2. The a_{ws} value at the high excitation frequency for a drum F_0=0.19x10^{6}N and f=35Hz reduce by 17.22% in comparison with at the low excitation frequency for a drum F_0=0.28x10^{6}N, f=30Hz. Those values of a_{ws} are the uncomfortable conditions for driver comfort (according to Tab.1). Similarly, from the results of Fig.3b, it also indicates that the a_{wphi} value at the high excitation frequency for a drum F_0=0.19x10^{6}N and f=35Hz reduce by 17.32% in comparison with at the low excitation frequency for a drum F_0=0.28x10^{6}N, f=30Hz.

In order to evaluate the effect of the excitation frequencies for a dynamic drum on the a_{ws} and a_{wphi} values, the excitation frequencies for a dynamic drum f=[15, 20, 25, 30, 35, 40, 45, 50] are selected for evaluation. The a_{ws} and a_{wphi} values with the different excitation frequencies for a dynamic drum in Case 1 and Case 2 are shown in Fig.4 and Fig.5.

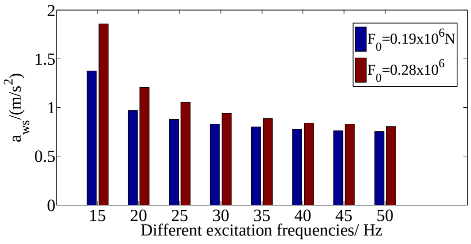


(a) Vertical driver's seat

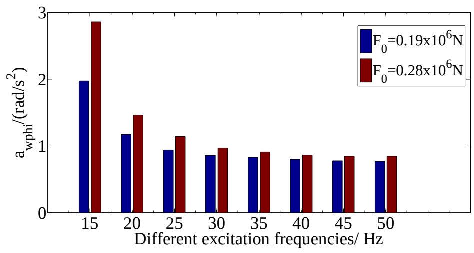


(b) Pitch angle of the cab

Figure 4. a_{ws} and a_{wphi} values with the different excitation frequencies for a dynamic drum in Case 1



(a) Vertical driver's seat



(b) Pitch angle of the cab

Figure 5. a_{ws} and a_{wphi} values with the different excitation frequencies for a dynamic drum in Case 2

From the results of Fig.4 with Case 1, it is pointed out that the f value increases, the a_{ws} and a_{wphi} values with the amplitude of excitation force for a drun F_0=0.28x10^{6}N reduce by 35.17%, 24.75%, 20.27%, 13.24%, 10.55%, 8.7%, 9.14%, 6.73% and 44.98%, 25.49%, 21.40%, 13.26 %, 10.05%, 8.09%, 8.82%, 9.77% respectively in comparison with the amplitude of excitation force for a drun F_0=0.28x10^{6}N. The a_{ws} and a_{wphi} values significantly reduce at the range of the low excitation frequency values. Similarly, from the results of Fig.5 with Case 2, the a_{ws} and a_{wphi} values are indicated to reduce significantly at the range of the low excitation frequency values. The effect of the excitation frequencies for a dynamic drum can be seen obviously.

V. CONCLUSION

In this study, a half-vehicle ride dynamic model under the ground surface deformation is established for analyzing and evaluating the effects of drum's excitation frequency on vehicle ride. The major conclusions can be drawn from the analysis results as follows:

i) The a_{ws} value with Case 1 at the high excitation frequency for a drum F_0=0.19x10^6N and f=35Hz reduce by 83.79% in comparison with at the low excitation frequency for a drum F_0=0.28x10^6N, f=30Hz and the a_{wphi} value reduce by 90.72% in comparison with at the excitation low frequency for a drum F_0=0.28x10^6N, f=30Hz.

ii) The a_{ws} value with Case 2 at the high excitation frequency for a drum F_0=0.19x10^6N and f=35Hz reduce by 17.22% in comparison with at the low excitation frequency for a drum F_0=0.28x10^6N, f=30Hz and the a_{wphi} value reduce by 17.32% in comparison with at the low excitation frequency for a drum F_0=0.28x10^6N, f=30Hz.

iii) The a_{ws} and a_{wph} values with Case1 and Case 2 significantly reduce at the range of the low excitation frequency values. The effect of the excitation frequencies for a dynamic drum is very obvious.

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