

Analysis of Ride Vibration Environment of Soil Compactors

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ABSTRACT

The ride dynamics of typical North-American soil compactors were investigated via analytical and experimental methods. A 12-degrees-of-freedom 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. The analytical model was formulated for the transit mode of operation at a constant forward speed on undeformable surfaces with the roller vibrator off. Field measurements were conducted to characterize the ride vibration environments during the transit mode of operation. The measured data revealed significant magnitudes of whole-body vibration of the operator-station along the vertical, lateral, pitch and roll-axes. The model results revealed reasonably good agreements with ranges of the measured vibration data. The ride dynamic responses of the soil compactor model were subsequently analyzed to study its whole-body vibration environment while operating on undeformable random terrain surfaces. Parametric sensitivity analyses were performed to study influences of different design parameters on the whole body vibration responses, which included the drum and cab vibration isolators, vertical seat suspension and possible rear wheel-axle suspension. The simulation results indicated minor beneficial effect of a suspended rear axle, while the drum isolator and suspension seat revealed most significant potential benefits.

Implementation of a seat suspension with natural frequency in order of 1.5 Hz could lead to nearly 60% reduction in the weighted vertical acceleration value. The results also suggest that a softer platform suspension would be beneficial in reducing the vertical and pitch vibration.

INTRODUCTION

The soil and asphalt compactors are most widely used in road construction, where the compaction is achieved by the static load associated with the operating weight of the vehicle and the drum vibration generated by a vibratory system. Compared to the heavy road and off-road vehicles, a soil compactor poses far more complex ride vibration, which is attributed to the interaction of the roller drum with the deformable soil or the terrain surface, in addition to those of the rear axle tires, as in the case of various tired off-road vehicles employed in the forestry, construction and agricultural sectors. The interactions of the drum and tires with the terrain surface are known to transmit comprehensive levels of low frequency whole-body vibration (WBV) to the operator. It has been established that prolonged exposure to such low frequency and high magnitude of WBV is directly associated with various health and safety risks among the vehicle drivers [1, 2, 3, 4]. Only a few studies, however, have attempted to characterize and assess the WBV environment of soil compactors [5-6].

The WBV environment of mobile equipments is in-general strongly influenced by many design and operating factors [7]. In case of soil compactors, the WBV environment is most significantly influenced by the mode of operation. In the transit mode, the low frequency whole-body vibration responses are primarily attributable to interactions of the pneumatic tires and the non-vibrating rigid drum with mostly undeformable terrain, and to occasional drum-hop. In the compaction mode, interactions of the vibrating drum with deformable soil and the drum hop motions account as the primary source of low frequency vibration, while the engine and the drum vibrator contribute to relatively high frequency vibration. The soil compactors are generally designed with elastic vibration isolation mounts between the drum and the chassis, and between the operator platform/cabin and the chassis. Such mounts, generally provide attenuation of only higher frequency vibration arising from the drum vibrator [8], when activated. Some of the models, however, include a low natural frequency vertical suspension at the seat to limit the transmission of vertical vibration to the operator. The soil compactors may thus be considered as unsuspended vehicles in the context of low frequency vibration caused by tire/drum interactions with the terrain surface. The vibration arising from the above sources are thus mostly transmitted to the operator station. The nature of WBV transmitted to drivers of various off-road vehicles has been extensively characterized in terms of frequencies of predominant vibration and frequency-weighted vibration magnitudes in order to assess potential health and safety risks of exposure to WBV [9, 10, 11]. Only minimal data on the vibration properties of compactors, however, could be found in the literatures. Beck et al. [6] measured the vibration responses of a Caterpillar CS-433B soil compactor on a construction site in the US and showed the vector sum of whole-body vibration measured along the vertical, fore-aft and lateral axes in the order of 2.5 m/s^2 in accordance with the ISO-2631-1 guideline [12]. A recent Canadian study [5] performed measurements on 3 soil compactors and revealed average frequency-weighted rms acceleration in the dominant vertical direction of 0.91 m/s^2 , while the level in the worst machine was in the order of 1.3 m/s^2 .

The lumped-parameter and multi-body dynamic models of different off-road vehicles have been widely reported to study the influences of various designs and operating factors on the transmitted vibration and to seek designs of primary and secondary suspension systems [14, 15, 16, 17]. Such efforts for soil compactors, however, have been limited to only a few kinematic and dynamic models for motion planning of the compaction tasks [18, 19, 20]. A ride dynamic model of a soil compactor incorporating the rigid roller drum and tire/drum-soil interactions could yield considerable insight into factors affecting the transmission of vibration to the operator. Furthermore, such a model could serve as an essential tool for

design and tuning of primary and secondary suspensions for the vehicle.

Since the excitation frequency due to drum vibratory system is significantly higher than the resonant frequencies of the isolation mounts in modern compactors, the low frequency whole-body vibration while performing vibratory compaction at very low speed would be expected to be lower than those encountered during transiting at higher speeds over rough terrains. This study presents the characterization of the ride vibration environment of single-drum soil compactors during the transit mode of operation using field measurements. An in-plane ride dynamic model of a soil compactor was formulated considering the rigid steel drum as an extension of the terrain, point contact of the rear-axle tire with the undeformable terrain surface, and linear visco-elastic properties of the cab and drum mounts, to predict the ride vibration responses. The variations in engine torque were not considered since vibratory compactors feature a hydrostatic transmission operating at constant engine rpm. The proposed model is validated using the measured data and applied to investigate the influences of various design modifications, such as drum isolators, cab suspension, vertical suspension seat and possible wheel-axle suspension, on the ride performance of the vehicle. The limitations of the proposed model are also discussed together with the need for additional modeling considerations.

RIDE DYNAMIC MODELING

The ride vibration responses of a soil compactor arising from interactions of the rear pneumatic tires and the front rigid drum with undeformable terrain (Fig. 1) are strongly influenced by weights and dimensions of the vehicle, in addition to many other design and operating conditions [21]. In this study, only planar dynamics of the compactor is investigated. A comprehensive 12-DOF model is formulated assuming linear stiffness and damping properties of the drum and cabin mounts, and the rear axle tires, as seen in Fig. 2. The rigid vehicle body is isolated from the ground by means of a pair of tires at the rear axle and drum suspension at the front; the drum being mounted to the front frame through a number of rubber mounts, as shown in Fig. 2. The contribution of the articulated-frame steering to the ride vibration responses during the transit mode, however, is assumed to be relatively small.

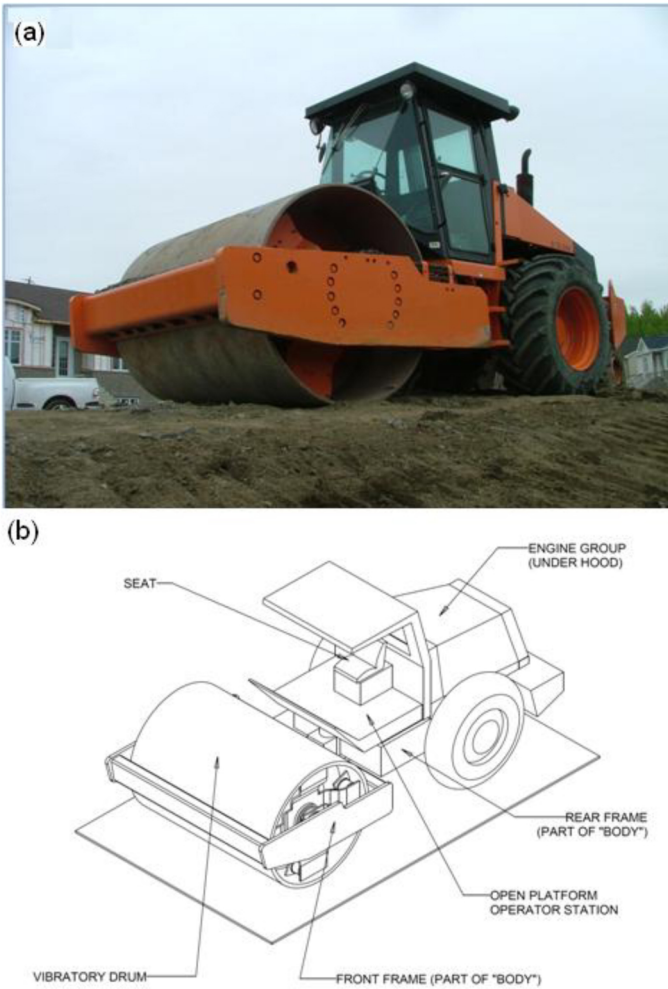


Figure 1. (a) A pictorial view of a single-drum soil compactor with enclosed driver cabin; and (b) a schematic of an open-platform soil compactor.

The compactor model integrates: (i) a 2-DOF model of the driver suspension seat as reported by Allen [22] in order to investigate the potential ride performance benefits of a suspension seat; (ii) a virtual rear-axle suspension with linear stiffness and damping constants (k_a^z , c_a^z) for investigating the potential ride performance benefits of a suspension axle; (iii) drum mounted to the front frame (vehicle body) through a number of rubber mounts represented by lumped linear stiffness and damping constants in the vertical (k_d^z , c_d^z) and longitudinal (k_d^x , c_d^x) directions, as pictorially shown in Fig. 3(a); and (iv) operator station (cabin) and engine group supported on the rear frame (vehicle body) through a number of rubber mounts with linear visco-elastic properties. The engine and cab mounts are represented by their vertical stiffness (k_{ej}^z , k_{cj}^z) and damping constants (c_{ej}^z , c_{cj}^z), as shown in the Figure. In a similar manner, the shear properties of the engine and cab mounts are represented by linear

stiffness (k_{ej}^x , k_{cj}^x) and damping (c_{ej}^x , c_{cj}^x) constants, where subscript $j=f, r$ denotes the front and rear mounts, respectively. The tire interaction with the terrain surface in the vertical direction is modeled by linear stiffness (k_t^z) and damping (c_t^z) elements and by linear stiffness (k_t^x) and damping (c_t^x) elements in the shear direction. The rear axle suspension representation by the vertical spring and damping elements (k_a^z , c_a^z) is supported by an independent suspension design proposed by Timoney Technologies [23] shown in Fig. 3(b).

The DOFs of the model include: the vertical, pitch and longitudinal motions of the vehicle body (z_b , ϕ_b , x_b) cabin mass (z_c , ϕ_c , x_c) and the engine mass (z_e , ϕ_e , x_e); vertical motion of the rear wheels (z_w); and vertical motion of the suspension seat mass (z_s) and the operator mass (z_h). The unsprung mass m_w is assumed to be constrained along the vertical axis. In the model m_b and I_b represent the mass and pitch mass moment of inertia of the vehicle body, and m_c and I_c , and m_e and I_e are those of the operator-station and the engine, respectively. The suspension seat and the operator's seated mass are represented by m_s and m_h , respectively.

The heights h_a , h_t and h_d are the vertical distances between the rear-axle center, tire-soil contact and the drum centre from the vehicle body cg, respectively. Similarly, h_{cm} and h_{em} are the vertical distances of the cab and engine mass cg from their isolation mounts, respectively. The visco-elastic properties of the seat suspension are represented by lumped linear stiffness and damping constants in the vertical (k_s^z , c_s^z) axis.

The equations of motion of the vehicle model are derived using the Lagrange's energy method and summarized below.

Unsprung Mass:

$$m_w \ddot{z}_w + F_t^z - F_a^z = 0; \quad (1)$$

Vehicle Body:

$$\begin{aligned} (m_b + m_w) \ddot{z}_b + F_d^x + F_t^x - F_e^x - F_c^x &= 0; \\ m_b \ddot{z}_b + F_d^z + F_a^z - F_{ef}^z - F_{er}^z - F_{cf}^z - F_{cr}^z &= 0; \\ I_b \ddot{\phi}_b + h_d F_d^x + h_t F_t^x - l_d F_d^z + l_t F_a^z \\ + (h_e - h_{em}) F_e^x - (l_e - l_{ef}) F_{ef}^z - \\ (l_e + l_{er}) F_{er}^z + (h_c - h_{cm}) F_c^x + (l_c + l_{cf}) F_{cf}^z \\ + (l_c - l_{cr}) F_{cr}^z &= 0; \end{aligned} \quad (2)$$

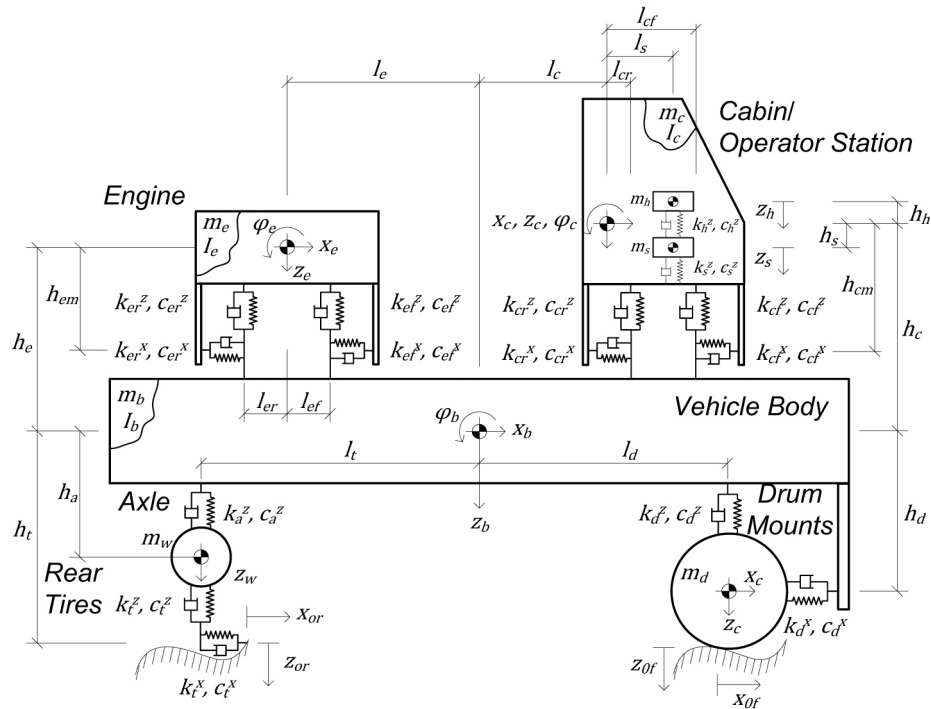


Figure 2. Twelve-DOF pitch-plane model of a single-drum soil compactor with a virtual rear axle suspension.

Engine:

$$\begin{aligned} m_e \ddot{x}_e + F_e^x &= 0; \\ m_e \ddot{z}_e + F_{ef}^z + F_{er}^z &= 0; \\ I_e \ddot{\phi}_e + h_{em} F_e^x - l_{ef} F_{ef}^z + l_{er} F_{er}^z &= M_e(t) \end{aligned} \quad (3)$$

Cabin:

$$\begin{aligned} m_c \ddot{x}_c + F_c^x &= 0; \\ m_c \ddot{z}_c + F_{cf}^z + F_{cr}^z - F_s^z &= 0; \\ I_c \ddot{\phi}_c + h_{cm} F_c^x - l_{cf} F_{cf}^z + l_{cr} F_{cr}^z + l_s F_s^z &= 0 \end{aligned} \quad (4)$$

Seat/Occupant:

$$\begin{aligned} m_s \ddot{z}_s + F_s^z - F_h^z &= 0; \\ m_h \ddot{z}_h + F_h^z &= 0 \end{aligned} \quad (5)$$

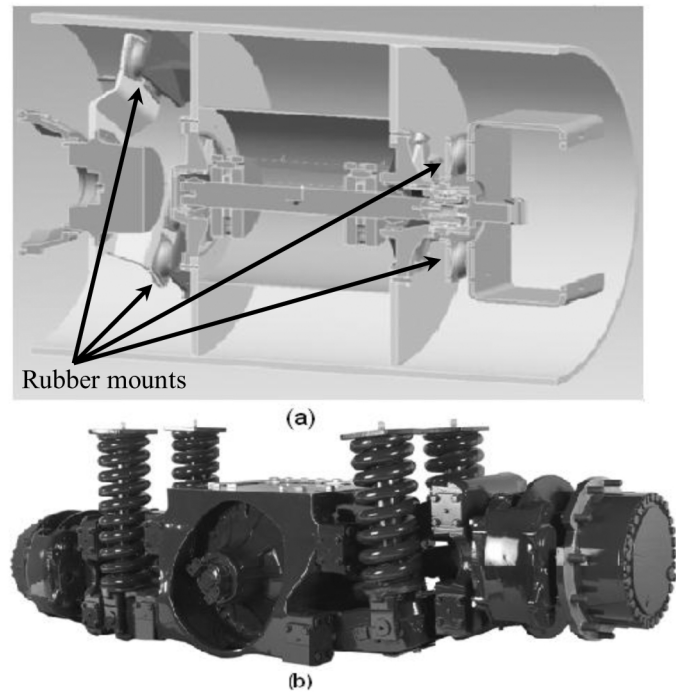


Figure 3. Pictorial views of: (a) the vibratory roller drum with rubber mounts on the drum drive (left) and on the drum bearing bracket (right); and (b) the axle suspension developed by Timoney Technologies [23].

In the above equations, F_t^x and F_d^x are the longitudinal forces due to tires and drum mounts, F_t^z , F_a^z and F_d^z are the vertical forces developed by the tires, axle suspension and the drum mounts, respectively. F_e^x and F_c^x represent the total longitudinal forces developed by the front and rear engine and cab mounts. The vertical forces due to engine and cab mounts are represented by F_{ej}^z and F_{cj}^z , where $j = f, r$ refers to the front and rear mounts, and $M_e(t)$ is the engine torque excitation. The forces and the moments developed due to the tire, drum mounts, engine, cab mounts and suspensions are described in details in [Appendix I](#).

The coupled differential equations of motion for the ride dynamic model of the compactor could be expressed in the matrix form as:

$$[M]\{\ddot{Z}\} + [C]\{\dot{Z}\} + [K]\{Z\} = \{F(t)\} \quad (6)$$

Where $[M]$, $[C]$ and $[K]$ are $(n \times n)$ mass, damping and stiffness matrices, respectively; n is the number of DOF and $\{F(t)\}$ is the $(n \times 1)$ forcing vector and $\{Z\}$ is the vector of generalized coordinates.

FIELD MEASUREMENTS AND DATA ANALYSIS

Field measurements were undertaken to characterize the vibration transmission properties of soil compactors under transit mode (vibration off) on undeformable surfaces at constant forward speeds. Consequently, the experiments were performed under controlled speed condition on the test courses at an equipment manufacturer's site. The results attained from the measurements performed on the test track are thus not considered suitable for assessment of WBV exposure that necessitates measurements under conditions encountered in actual jobsites. These were, however, considered vital for determining the vibration properties of the vehicle, particularly the dominant frequencies and relative magnitudes of vibration. The experiments involved two different vehicles. Two typical soil compactors were selected with their gross weight in the range of 10 tons, which represents the greatest population of this type of equipment in North America. The selected machines were powered by an in-line 4 or 6 cylinder engines. Both the vehicles were equipped with an open platform (operator-station) featuring an operator seat without a vertical suspension.

Compactors were driven on tracks longer than 100 m on an undeformable road surface. The vibrator was set to the "off position. Apart from the transit on the tracks, the transient responses of the vehicles were also measured while operating over discrete obstacles. The test program involved a total of 27 and 23 runs with the two compactors, which are denoted

by "6-cylinder compactor" and "4-cylinder compactor", respectively. These also included runs with engine idling for verifications of the sensors signals. Each test vehicle was instrumented to measure the vibration of the responses of the four primary bodies, namely the seat, operator-station, drum, and the vehicle body. In the current design, soil compactors wheel axles are rigidly bolted to the rear-frame; the vertical vibrations of the rear frame at the axle location were acquired.

The vehicle was instrumented with accelerometers at different locations. These included: (i) a 3-axis seat accelerometer per ISO recommendations [24] installed on the seat cushion for measurements of fore-aft (x), lateral (y) and vertical (z) vibration at the operator/seat interface; (ii) a rate gyro coupled with a 3-axis accelerometer installed on the platform floor to measure the translational and rotational vibration of the operator-station structure; (iii) a rate gyro together with a 3-axis accelerometer attached to the rear frame, close to the center of mass of the vehicle body for measurements of translational and rotational vibration of the vehicle body; (iv) a 3-axis accelerometer mounted on the rear frame above the bolted joint of the rigid axle, although the vertical acceleration only was acquired, since the horizontal vibrations were expected to be similar that of the vehicle body; (v) a 3-axis accelerometer attached to the drum-propelling side of the front frame (hydrostatic transmission) in-plane with the drum axis for measurements of translational vibrations transmitted by the drum to the front frame (vehicle body) through the elastic mounts; and (vi) a 3-axis accelerometer attached to the vibrator motor bracket of the drum to measure the drum radial accelerations along the longitudinal- and vertical-axes. [Fig. 4](#) illustrates the locations of some of the accelerometers.

The measured analog data were processed through a 0.7 Hz cut-off frequency high-pass filter to eliminate the signal bias, if any, and digitized using Brüel & Kjær PULSE Analyzer Platform. The time history of each signal was recorded through the Time capture analyzer. The FFT analyzer processed each data segment at a sampling rate of 256 with a resolution of 0.125 Hz over a range of 100 Hz. The analysis also utilized a 75% overlap and a Hanning time-weighting function. The CPB analyzer was also used to derive one-third-octave bands acceleration spectra with center frequencies ranging from 0.5 Hz to 100 Hz. The acquired data were analyzed to derive the acceleration spectra corresponding to each test compactor in each segment corresponding to the different operating modes considered. The results were obtained in terms of PSD and -octave band rms values of accelerations along all the translational and rotational axes. The results attained for the related segments or repeated trials for each operating mode were subsequently grouped and averaged.