STUDY ON THE INTERACTION BETWEEN AN AGRICULTURAL TRACTOR AND FIELD TERRAIN PROFILES
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农用拖拉机与田间地面相互作用的研究

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ABSTRACT
To study the interaction between the tractor and the field terrain on which the tractor operates, the characteristics of tractor vibrations excited by the measured profiles were analyzed by simulations and experiments. The results show that the theoretically calculated values of the tractor vibrations excited by the field road profiles, including the natural frequencies of the tractor's center of gravity, acceleration RMS values of the tractor axles, and the wheel dynamic loads, were close to the measured values, with an error range of 3.5% to 10.9%. Considering the filtering effect of large tractor tires on the measured road profiles, these errors are within an acceptable range. The results confirm the validity of the methodology presented in this paper for investigating the tractor vibrations caused by measured profiles as excitations.

摘要
为了研究拖拉机与田间地面的相互作用，通过仿真分析和实验验证的方法分析了田间地面不平度激励下的拖拉机振动特性。结果表明，拖拉机在田间路面激励下的振动理论计算值包括质心处固有频率、车轴加速度均方根值和车轮动载荷，都与实测值接近，误差范围为 3.5%～10.9%，考虑到大型拖拉机轮胎对实测路面轮廓的滤波作用，这些误差在可接受的范围内。证实了本文提出由实测地面不平度作为激励源研究拖拉机振动特性的方法的有效性。

INTRODUCTION
The dynamic behavior of a tractor is strongly affected by the surface profile of the agricultural terrain upon which the tractor operates. In many cases, only the standard Power Spectral Density (PSD) of the profile is provided, not the actual measured profile *(ISO 8608, 2016)*. Carrying out profiling tests and analyses of some typical agricultural terrains and revealing the surface roughness characteristics of the measured terrains not only lays a foundation for the further study of tractor vibrations excited by different terrain profiles but also provides a realistic representation of the desired terrain for the virtual simulation and optimal design of agricultural machinery. The analysis of profile data should be targeted at an application involving excitation sources that induce dynamic response in agricultural machinery. However, most studies have focused on the acquisition of terrain profiles and modeling, which are relatively independent of the research on the interactions between vehicles and terrain *(Gialamas et al., 2016; Goodin et al., 2017; Botha et al., 2019)*. Additionally, many studies have concentrated on the simulation analysis of vehicle vibrations by using measured terrain profiles *(Kropáč et al., 2009; Wei et al., 2016; Zheng et al., 2016; Sim et al., 2017)*, but it is difficult to verify the vehicle vibrations caused by the measured profiles as excitations.

This study was designed to study the interaction between a tractor and the field terrain on which the tractor operates, the tractor vibration excited by the measured profiles was investigated by theoretical analysis and experimental research.

MATERIALS AND METHODS
As shown in Figure 1, a surface profiling apparatus (profiler) was mounted on the front counterweight of a tractor *(John Deere 904, John Deere Tianjin Co.ltd.)*, which makes it easy to further

analyze the coupling vibration between the tractor and the field terrain because the measured profiles are the sources of excitation that will cause the tractor to vibrate when the tractor wheels roll along the profile path. Previous studies have laid a foundation for the approach proposed in this study. The design and validation of the profiler was presented in detail in a previous study *(Yan et al., 2019)*. Details on the testing and analysis of agricultural terrain profiles can be found in *Wang et al (2020)*, in which the profiling tests were carried out in a harvested potato field, a grass field, a corn stubble field and a field road.

Fig. 1 - Profiling apparatus mounted on the front counterweight of a tractor

Fig. 2 - Terrain profile and tractor vibration test on field road

a) Front axle end ***b) Directly below seat*** ***c) Rear axle end***

d) Directly below the front axle ***e) Directly below the rear axle***

Fig. 3 - Sensor installation positions

To investigate the tractor vibration caused by the field terrain upon which the tractor was driven, the terrain profiles were measured on the field road, and the vibration accelerations of the tractor body and axles were measured simultaneously with a forward driving speed of 5.41 km/h. During the test, the width of each surface profiles on the parallel tracks was adjusted to be the same as the width of the tractor wheel tracks, as shown in Figure 2, hence, the measured profiles were the excitation source that caused the tractor to vibrate when the tractor wheels rolled along the profile path. Then, the tractor vibrations excited by the measured profiles were investigated. The tire pressure of the tractor was 170 kPa in the front and 160 kPa in the rear.

Seven single-direction accelerometers (IEPE 1A111E) with a frequency bandwidth of 0.5-10000 Hz and a maximum acceleration of 500 m/s^2 were attached to the tractor to measure the vibration intensity of the tractor body and axles, as shown in Figure 3. The specific arrangement of accelerometers is as follows: four accelerometers were attached to the ends of the axles at the centers of the tractor's four wheels to measure the vertical acceleration of each wheel during the tractor wheel dynamic load calculation; one accelerometer was attached to the tractor cab floor below the driver's seat in the middle of the tractor body to measure the vertical acceleration of the tractor body; and two accelerometers were attached at the midpoints of the front and rear axles to measure the vertical acceleration of each axle. The measured acceleration and profiling signals were collected by the profiler's data acquisition system in tandem during the measurement process with a sample acquisition rate of 256 Hz.

MODEL ANALYSIS OF TRACTOR VIBRATION

To investigate the vibrations in an agricultural tractor produced by the interaction between the tractor and the field terrain, an equivalent model of a wheeled tractor was established, as shown in Figure 4. A John Deere 904 tractor was selected as the prototype. Referring to the relevant Chinese standard, the inherent parameters of the tractor, such as the centroid position, pitch and roll moment of inertia were measured and determined as shown in Table 1. The tractor vibration model includes the vertical displacement of the tractor's center of gravity z_c, pitch angle displacement of the tractor's center of gravity φ_c, roll angle displacement of the tractor's center of gravity θ_c, and road profile excitations q_{f1}, q_{r1}, q_{f2}, and q_{r2}.

Table 1

Some inherent parameters of the John Deere 904 tractor

Parameters	**Value**	**Variable unit**
Mass of the tractor body	M_c = 4180	[kg]
Pitch moment of inertia	J_φ = 4529.2	[kg·m^2]
Roll moment of inertia	J_θ = 2184.3	[kg·m^2]
Distance from the tractor center of gravity to the front axle	l_{cf} = 1.31	[m]
Distance from the tractor center of gravity to the rear axle	l_{cr}= 0.99	[m]
Distance from tractor centerline to the front wheel center	l_{df} = 0.82	[m]
Distance from tractor centerline to the rear wheel center	l_{dr} = 0.86	[m]
Stiffness of the front tire with a pressure of 170 kPa	k_f = 357425	[N/m]
Stiffness of the rear tire with a pressure of 160 kPa	k_r = 369840	[N/m]
Damping coefficient of the front tire with a pressure of 170 kPa	c_f = 3253	[N·s/m]
Damping coefficient of the rear tire with pressure of 160 kPa	c_r = 3674	[N·s/m]

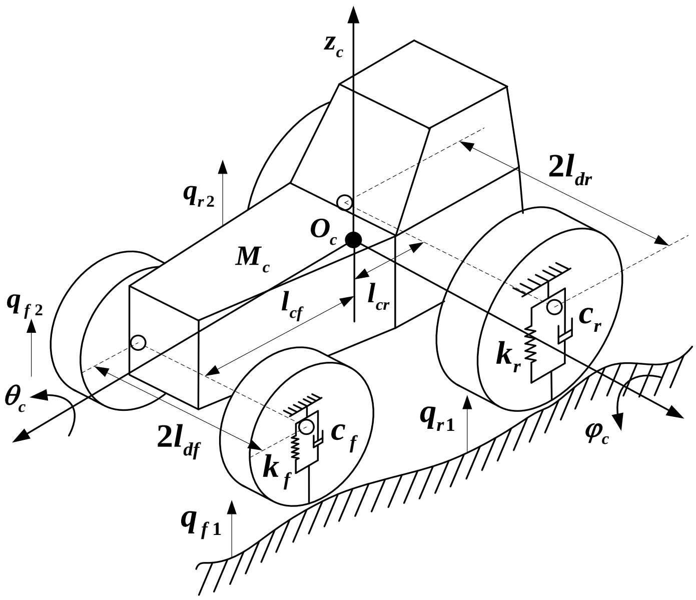


Fig. 4 - Tractor vibration model

The tractor vibration model has three DOFs with the following equations:

$$\begin{cases} M_c\ddot{z}_c + (2c_f + 2c_r)\dot{z}_c + (2k_f + 2k_r)z_c + (-2c_f l_{cf} + 2c_r l_{cr})\dot{\varphi}_c + (-2k_f l_{cf} + 2k_r l_{cr})\varphi_c \\ = c_f(\dot{q}_{f1} + \dot{q}_{f2}) + k_f(q_{f1} + q_{f2}) + c_r(\dot{q}_{r1} + \dot{q}_{r2}) + k_r(q_{r1} + q_{r2}) \\ J_\varphi\ddot{\varphi}_c + (2c_f l_{cf}^2 + 2c_r l_{cr}^2)\dot{\varphi}_c + (2k_f l_{cf}^2 + 2k_r l_{cr}^2)\varphi_c + (-2c_f l_{cf} + 2c_r l_{cr})\dot{z}_c + (-2k_f l_{cf} + 2k_r l_{cr})z_c \\ = -c_f l_{cf}(\dot{q}_{f1} + \dot{q}_{f2}) - k_f l_{cf}(q_{f1} + q_{f2}) + c_r l_{cr}(\dot{q}_{r1} + \dot{q}_{r2}) + k_r l_{cr}(q_{r1} + q_{r2}) \\ J_\theta\ddot{\theta}_c + (2c_f l_{df}^2 + 2c_r l_{dr}^2)\dot{\theta}_c + (2k_f l_{df}^2 + 2k_r l_{dr}^2)\theta_c \\ = c_f l_{df}(\dot{q}_{f1} - \dot{q}_{f2}) + k_f l_{df}(q_{f1} - q_{f2}) + c_r l_{dr}(\dot{q}_{r1} - \dot{q}_{r2}) + k_r l_{dr}(q_{r1} - q_{r2}) \end{cases} \tag{1}$$

The front axle vertical acceleration $\ddot{z}_f$ and rear axle vertical acceleration $\ddot{z}_r$ can be derived from the vertical acceleration of the center of gravity $\ddot{z}_c$ and the pitch angle acceleration of the tractor's center of gravity $\ddot{\varphi}_c$, as shown in Eq (2).

$$\begin{cases} \ddot{z}_f = \ddot{z}_c - l_1\ddot{\varphi}_c \\ \ddot{z}_r = \ddot{z}_c + l_2\ddot{\varphi}_c \end{cases} \tag{2}$$

The dynamic loads of the two front wheels in response to the excitations of terrain profiles q_{f1} and q_{f2} are determined as follows:

$$\begin{cases} F_{f1}(t) = k_f(q_{f1} - z_c + \varphi_c l_{cf} - \theta_c l_{df}) + c_f(\dot{q}_{f1} - \dot{z}_c + \dot{\varphi}_c l_{cf} - \dot{\theta}_c l_{df}) \\ F_{f2}(t) = k_f(q_{f2} - z_c + \varphi_c l_{cf} + \theta_c l_{df}) + c_f(\dot{q}_{f2} - \dot{z}_c + \dot{\varphi}_c l_{cf} + \dot{\theta}_c l_{df}) \end{cases} \tag{3}$$

The dynamic loads of the two rear wheels in response to the excitations of terrain profiles q_{r1} and q_{r2} are similarly determined as follows:

$$\begin{cases} F_{r1}(t) = k_r(q_{r1} - z_c - \varphi_c l_{cr} - \theta_c l_{dr}) + c_r(\dot{q}_{r1} - \dot{z}_c - \dot{\varphi}_c l_{cr} - \dot{\theta}_c l_{dr}) \\ F_{r2}(t) = k_r(q_{r2} - z_c - \varphi_c l_{cr} + \theta_c l_{dr}) + c_r(\dot{q}_{r2} - \dot{z}_c - \dot{\varphi}_c l_{cr} + \dot{\theta}_c l_{dr}) \end{cases} \tag{4}$$

The RMS values of the dynamic loads of the tractor wheels are as follows:

$$RMS_{F_{f1}} = \sqrt{\sum_{i=1}^{N} F_{f1}^2(t) \Big/ N} \tag{5}$$

$$RMS_{F_{f2}} = \sqrt{\sum_{i=1}^{N} F_{f2}^2(t) \Big/ N} \tag{6}$$

$$RMS_{F_{r1}} = \sqrt{\sum_{i=1}^{N} F_{r1}^2(t) \Big/ N} \tag{7}$$

$$RMS_{F_{r2}} = \sqrt{\sum_{i=1}^{N} F_{r2}^2(t) \Big/ N} \tag{8}$$

Where:

N is the number of sampling points.

Eq.(1) may be rearranged in matrix form as:

$$M\ddot{Z} + C\dot{Z} + KZ = F \tag{9}$$

Where:

$$Z = \begin{bmatrix} z_c \\ \varphi_c \\ \theta_c \end{bmatrix} \quad M = \begin{bmatrix} M_c & 0 & 0 \\ 0 & J_\varphi & 0 \\ 0 & 0 & J_\theta \end{bmatrix}$$

$$C = \begin{bmatrix} 2c_f + 2c_r & -2c_f l_{cf} + 2c_r l_{cr} & 0 \\ -2c_f l_{cf} + 2c_r l_{cr} & 2c_f l_{cf}^2 + 2c_r l_{cr}^2 & 0 \\ 0 & 0 & 2c_f l_{df}^2 + 2c_r l_{dr}^2 \end{bmatrix}$$

$$K = \begin{bmatrix} 2k_f + 2k_r & -2k_f l_{cf} + 2k_r l_{cr} & 0 \\ -2k_f l_{cf} + 2k_r l_{cr} & 2k_f l_{cf}^2 + 2k_r l_{cr}^2 & 0 \\ 0 & 0 & 2k_f l_{df}^2 + 2k_r l_{dr}^2 \end{bmatrix}$$

$$F = \begin{bmatrix} c_f(\dot{q}_{f1} + \dot{q}_{f2}) + k_f(q_{f1} + q_{f2}) + c_r(\dot{q}_{r1} + \dot{q}_{r2}) + k_r(q_{r1} + q_{r2}) \\ -c_f l_{cf}(\dot{q}_{f1} + \dot{q}_{f2}) - k_f l_{cf}(q_{f1} + q_{f2}) + c_r l_{cr}(\dot{q}_{r1} + \dot{q}_{r2}) + k_r l_{cr}(q_{r1} + q_{r2}) \\ c_f l_{df}(\dot{q}_{f1} - \dot{q}_{f2}) + k_f l_{df}(q_{f1} - q_{f2}) + c_r l_{dr}(\dot{q}_{r1} - \dot{q}_{r2}) + k_r l_{dr}(q_{r1} - q_{r2}) \end{bmatrix}$$

The natural frequencies of the tractor's center of gravity, such as the vertical vibration frequency f_z, pitching vibration frequency f_φ and roll vibration frequency f_θ, were determined by the undamped vibration equations, for which the eigenvalues of $H = M^{-1}K$ need to be derived. Then, the natural frequencies were calculated as follows:

$$\begin{cases} f_z \approx 2.88 \ \text{Hz} \\ f_\varphi \approx 3.38 \ \text{Hz} \\ f_\theta \approx 3.45 \ \text{Hz} \end{cases}$$

Simulation of tractor vibration accelerations

To simulate the tractor vibrations caused by the measured profiles as excitations in the MATLAB Simulink environment, Eq (1) can be written in terms of state-space equations:

$$\begin{cases} \dot{X} = AX + BU \\ Y = CX + DU \end{cases} \tag{10}$$

Where:

A is the system matrix, B is the input matrix, C is the output matrix, D is the feed-forward matrix, X is known as the state vector, Y is the output vector, and U is the input vector.

$$A = \begin{bmatrix} -\frac{2c_f + 2c_r}{M_c} & -\frac{-2c_f l_{cf} + 2c_r l_{cr}}{M_c} & 0 & -\frac{2k_f + 2k_r}{M_c} & -\frac{-2k_f l_{cf} + 2k_r l_{cr}}{M_c} & 0 \\ -\frac{-2c_f l_{cf} + 2c_r l_{cr}}{J_\varphi} & -\frac{2c_f l_{cf}^2 + 2c_r l_{cr}^2}{J_\varphi} & 0 & -\frac{-2k_f l_{cf} + 2k_r l_{cr}}{J_\varphi} & -\frac{2k_f l_{cf}^2 + 2k_r l_{cr}^2}{J_\varphi} & 0 \\ 0 & 0 & -\frac{2c_f l_{df}^2 + 2c_r l_{dr}^2}{J_\theta} & 0 & 0 & -\frac{2k_f l_{df}^2 + 2k_r l_{dr}^2}{J_\theta} \\ 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \end{bmatrix}^{\mathrm{T}}$$

$$B = \begin{bmatrix} \frac{c_f}{M_c} & \frac{c_r}{M_c} & \frac{c_f}{M_c} & \frac{c_r}{M_c} & \frac{k_f}{M_c} & \frac{k_r}{M_c} & \frac{k_f}{M_c} & \frac{k_r}{M_c} \\ -\frac{c_f l_{cf}}{J_\varphi} & \frac{c_r l_{cr}}{J_\varphi} & -\frac{c_f l_{cf}}{J_\varphi} & \frac{c_r l_{cr}}{J_\varphi} & -\frac{k_f l_{cf}}{J_\varphi} & \frac{k_r l_{cr}}{J_\varphi} & -\frac{k_f l_{cf}}{J_\varphi} & \frac{k_r l_{cr}}{J_\varphi} \\ \frac{c_f l_{df}}{J_\theta} & \frac{c_r l_{dr}}{J_\theta} & -\frac{c_f l_{df}}{J_\theta} & -\frac{c_r l_{dr}}{J_\theta} & \frac{k_f l_{df}}{J_\theta} & \frac{k_r l_{dr}}{J_\theta} & -\frac{k_f l_{df}}{J_\theta} & -\frac{k_r l_{dr}}{J_\theta} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} = D$$

$$C = \begin{bmatrix} -\frac{2c_f + 2c_r}{M_c} & -\frac{-2c_f l_{cf} + 2c_r l_{cr}}{M_c} & 0 & -\frac{2k_f + 2k_r}{M_c} & -\frac{-2k_f l_{cf} + 2k_r l_{cr}}{M_c} & 0 \\ -\frac{-2c_f l_{cf} + 2c_r l_{cr}}{J_\varphi} & -\frac{2c_f l_{cf}^2 + 2c_r l_{cr}^2}{J_\varphi} & 0 & -\frac{-2k_f l_{cf} + 2k_r l_{cr}}{J_\varphi} & -\frac{2k_f l_{cf}^2 + 2k_r l_{cr}^2}{J_\varphi} & 0 \\ 0 & 0 & -\frac{2c_f l_{df}^2 + 2c_r l_{dr}^2}{J_\theta} & 0 & 0 & -\frac{2k_f l_{df}^2 + 2k_r l_{dr}^2}{J_\theta} \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}^{\mathrm{T}}$$

$$X = \begin{bmatrix} \dot{z}_c & \dot{\varphi}_c & \dot{\theta}_c & z_c & \varphi_c & \theta_c \end{bmatrix}^{\mathrm{T}}$$

$$U = \begin{bmatrix} \dot{q}_{f1} & \dot{q}_{r1} & \dot{q}_{f2} & \dot{q}_{r2} & q_{f1} & q_{r1} & q_{lf2} & q_{lr2} \end{bmatrix}^{\mathrm{T}}$$

$$Y = \begin{bmatrix} \ddot{z}_c & \ddot{\varphi}_c & \ddot{\theta}_c & z_c & \varphi_c & \theta_c \end{bmatrix}^{\mathrm{T}}$$

According to the state-space expression (10), the Simulink model was established as shown in Figure 5 and was to analyze the vertical, pitch and roll vibrations of the tractor’s center of gravity, as well as the vibrations of the front and rear axles excited by the terrain profiles.

The values of the elements in matrices A, B, C and D determined in the state-space expression (10) were calculated according to the parameters in Table 1.

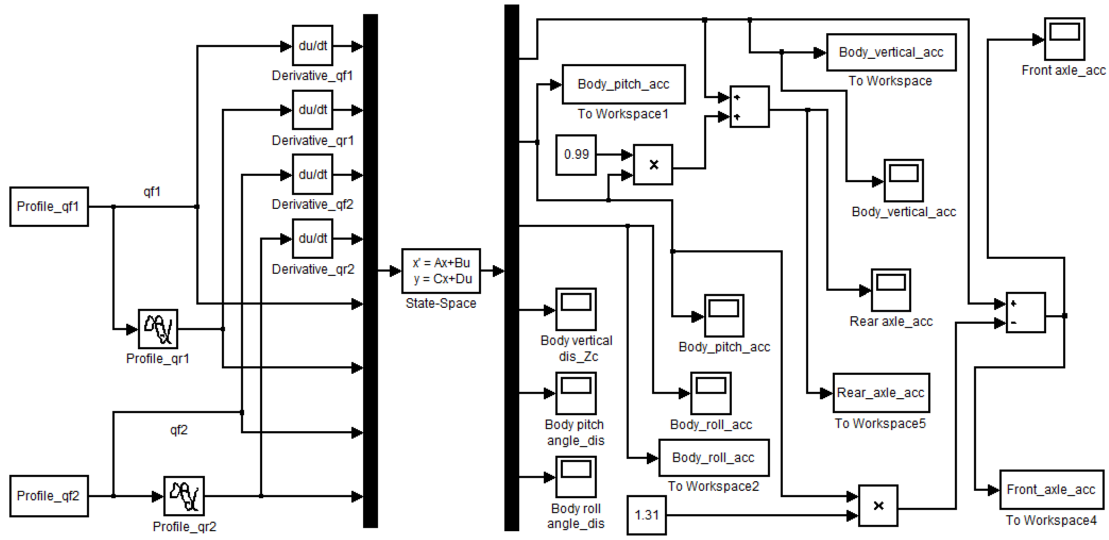
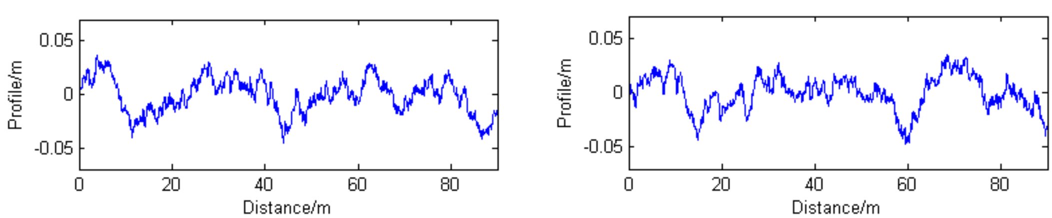


Fig. 5 - Simulink model of the tractor's center of gravity and axle vibrations.

Simulation of tractor wheel dynamic loads was used to analyze the dynamic load of the front wheel in response to the terrain due to excitation of terrain profile q_{f1} and q_{f2} excitation. The analytical model of the dynamic load characteristics of the tractor rear wheels in response to the terrain is similar to that of the dynamic load characteristics of the tractor front wheel.

RESULTS AND DISCUSSION

The field road profiles from the left and right tractor wheel tracks measured by the profiler at a speed of 5.41 km/h are shown in Figure 6.



a) Left track ***b) Right track***

Fig. 6 - Measured field road profiles

According to the geometric relationship between the tractor's center of gravity and the positions of the accelerometers attached to the tractor body, the measured vertical acceleration of the center of gravity $\ddot{z}_c$ can be derived from the vertical acceleration $\ddot{z}_s$ of the accelerometer attached to the tractor body below the driver's seat with the following equation:

$$\ddot{z}_c = \ddot{z}_s - l_{cs}\ddot{\varphi}_c \tag{11}$$

Where:

l_{cs} is the distance from the tractor's center of gravity to the position of the accelerometer attached to the tractor cab floor below the driver's seat in the middle of the tractor body.

The measured vertical, pitching and rolling vibration accelerations of the tractor's center of gravity in the time domain while the tractor was driving on the field road at a speed of 5.41 km/h are shown in Figures 7-9. These figures show that the amplitude of the vertical vibration of the tractor's center of gravity was larger than the amplitudes of the pitching and rolling vibrations.

The measured vibration accelerations of the front and rear axles in the time domain are shown in Figure 10 and Figure 11, respectively, indicating that the vibration acceleration amplitude of the front axle was larger than that of the rear axle. The dynamic loads of the front and rear wheels are shown in Figure 12 and Figure 13, respectively.

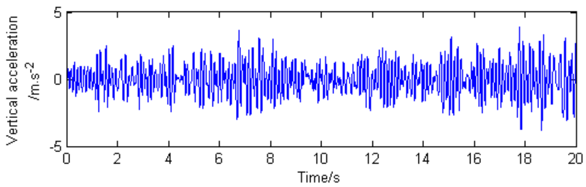


Fig. 7 - Vertical vibration acceleration of the tractor's center of gravity

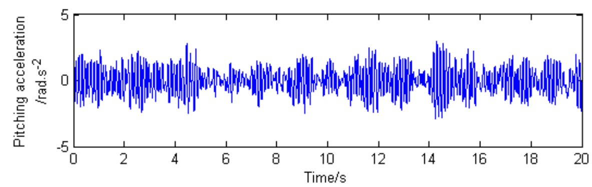


Fig. 8 - Pitching vibration acceleration of the tractor's center of gravity

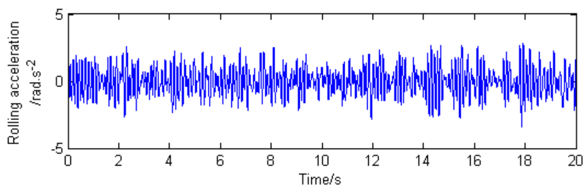


Fig. 9 - Rolling vibration acceleration of the tractor's center of gravity

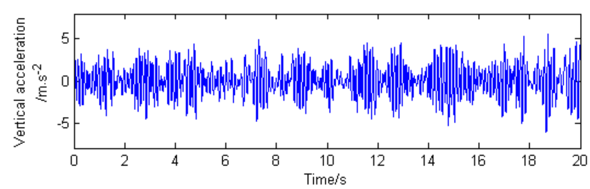


Fig. 10 - Vertical vibration acceleration of the front axle

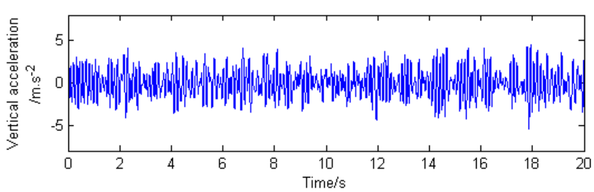


Fig. 11 - Vertical vibration acceleration of the rear axle

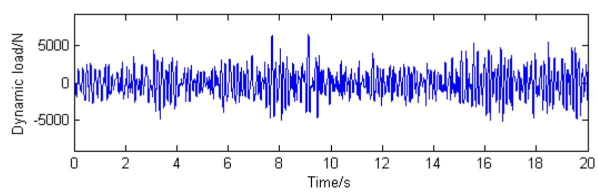


Fig. 12 - Dynamic load of the tractor front wheel

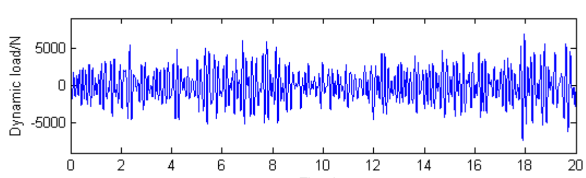


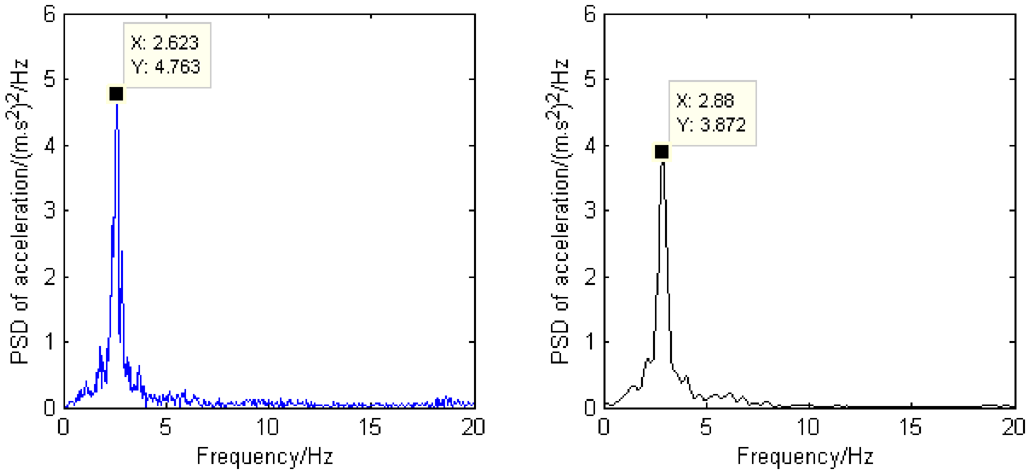
Fig. 13 - Dynamic load of the tractor rear wheel

The *PSD*s of the vertical, pitching and rolling vibration accelerations of the tractor's center of gravity corresponding to Figures 7-9 are shown in Figure 14(a), Figure 15(a) and Figure 16(a), respectively.

These figures display the vibration acceleration *PSD* in the range of 0 ~ 20 Hz, which covers the main frequency range of the tractor's natural frequencies and vibration response energy.

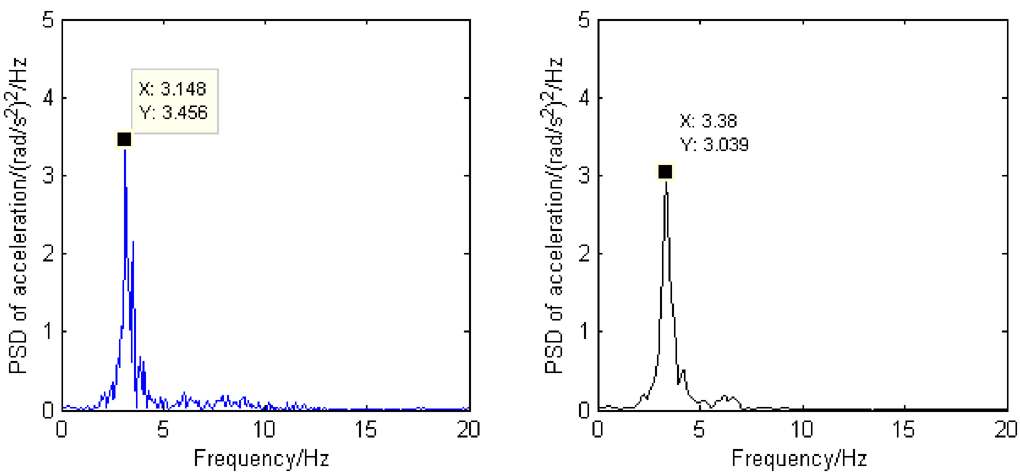
The natural frequencies of the tractor's center of gravity can be obtained from the peak frequencies of the vertical, pitching and rolling vibration acceleration *PSD*s of the tractor body's responses to the wheel excitations *(Zhu et al., 2016)*.

The natural frequencies of the vertical, pitching and rolling vibrations were 2.623 Hz, 3.148 Hz and 3.575 Hz, as shown in Figure 14(a), Figure 15(a) and Figure 16(a), respectively.



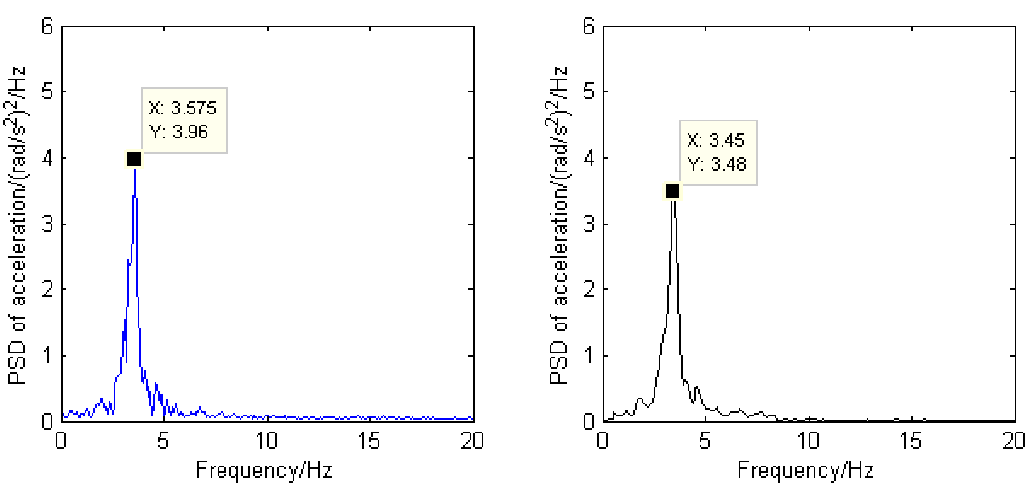
a) Tested PSD *b) Simulated PSD*

Fig. 14 - Vertical vibration acceleration PSDs of the tractor's center of gravity



a) Tested PSD *b) Simulated PSD*

Fig. 15 - Pitching vibration acceleration PSDs of the tractor's center of gravity



a) Tested PSD *b) Simulated PSD*

Fig. 16 - Rolling vibration acceleration PSDs of the tractor's center of gravity

The simulated *PSD*s of the vertical, pitching and rolling vibration accelerations of the tractor's center of gravity excited by the measured field road profiles of the left and right tractor wheel tracks are shown in Figure 14(b), Figure 15(b) and Figure 16(b), respectively.

The measured and theoretically calculated values of tractor vibrations excited by the field road profiles, including the natural frequencies of the tractor's center of gravity, the *RMS* values of the tractor axle accelerations and the wheel dynamic loads, are concluded in Table 2. A comparison between the measured values and the theoretically calculated values of tractor vibrations shows that the theoretical values are close to the measured values with an error range of 3.5% to 10.9%. Considering the filtering effect of large tractor tires on the measured road profiles, these errors are within an acceptable range. This conclusion confirms the validity of the developed tractor vibration model and the proposed methodology for investigating the tractor vibrations caused by measured profiles as excitations.

Table 2

Experimental values and theoretically calculated results of the tractor vibration tests on a field road (test speed: 5.41 km/h)

Value	Measured values	Theoretical values	Error
Natural frequency of the tractor's center of gravity vertical vibration (Hz)	2.623	2.88	9.8%
Natural frequency of the tractor's center of gravity pitching vibration (Hz)	3.148	3.38	7.4%
Natural frequency of the tractor's center of gravity rolling vibration (Hz)	3.575	3.45	3.5%
RMS value of the vertical vibration acceleration of the tractor's center of gravity (m/s^2)	1.18	1.08	8.5%
RMS value of the pitching vibration acceleration of the tractor's center of gravity (rad/s^2)	1.01	0.93	7.9%
RMS value of the rolling vibration acceleration of the tractor's center of gravity (rad/s^2)	1.06	0.97	8.5%
RMS value of the front axle acceleration (m/s^2)	1.88	1.69	10.1%
RMS value of the rear axle acceleration (m/s^2)	1.65	1.55	6.1%
RMS value of the tractor front wheel load (N)	2750	2941	6.9%
RMS value of the tractor rear wheel load (N)	3142	3484	10.9%

CONCLUSIONS

In this paper, a surface profiling apparatus mounted on the front counterweight of a tractor was designed to measure agricultural terrain profiles along parallel tracks. To investigate the interaction between a tractor and the field terrain on which the tractor operates, the terrain profiles were measured on the field road, and the vibration accelerations of the tractor body and axles were measured simultaneously with a forward driving speed of 5.41 km/h. An equivalent model of a wheeled tractor was established. Then, natural frequencies of the tractor's center of gravity, acceleration *RMS* values of the tractor axles, and wheel dynamic loads were investigated by theoretical analysis and experimental research. The results show that the theoretically calculated values of tractor vibrations excited by the field road profiles were close to the measured values, with an error range of 3.5% to 10.9%. This conclusion confirms the validity of the developed tractor vibration model and the proposed methodology for investigating the tractor vibrations caused by measured profiles as excitations.

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