

Article

Study on Operating Vibration Characteristics of Different No-Tillage Planter Row Units in Wheat Stubble Fields

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Abstract: The complex field environment under conservation tillage aggravates the vibration during a planter's operation, affecting the sowing quality and fertilization depth. Studying its vibration characteristics can help to realize active vibration reduction control of planter row units. To this end, this paper took a four-row no-till planter as the research object. By establishing a field vibration model of the planter row unit, the factors affecting the vibration of the unit were clarified, and stubble height, working speed and the additional weight of the planter were used as experimental factors in carrying out field orthogonal experiments. In our experiment, we collected and analyzed vibration data on the four-row planter row units and the frame at different positions to explore the influence of various factors on the vibration characteristics of the planter. The experimental results showed that the working speed was the most important factor affecting the vibration of the planter, and the impact of stubble height and additional weight on the amplitude of the planter was more significant at low speed (1.5 m/s) than that at high speed (2.5 m/s). The difference in amplitude of each planter unit in the lateral direction was the largest, the average amplitude range of which was 1.898 m/s². The vibration energy of each planter row unit under different working conditions was mainly concentrated in the range of 10–50 Hz. However, the three-point hitch of the planter transmitted the vibration excitation of the tractor, causing 110–120 Hz high-frequency vibration of the inner row units, while the outer row units were less affected, with the vibration energy, in the range above 100 Hz, being 2.5 dB smaller than that on the inner side. The right ground wheel transmission device was abnormal, which worked together with the excitation transmitted by the three-point hitch, making the average vibration acceleration amplitude of the planter row units on the right side in the lateral direction more than 0.522 m/s² higher than that of the units on the left side. Therefore, different vibration reduction forces need to be applied according to the position of the planter row unit, so that the units can avoid the natural frequency of the frame (115 Hz) when vibrating. This study can provide a reference for active vibration reduction control and improvements in sowing quality for high-speed no-till planters.

Keywords: corn; conservation tillage; sowing quality; vibration reduction; vibration model; orthogonal experiment; time–frequency analysis



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1. Introduction

Conservation tillage technology reduces nutrient loss by reducing and avoiding tillage [1], which can reduce the workload of cleaning straw on the ground and avoid

excessive soil compaction. It has the functions of water storage and soil moisture conservation, cost saving and efficiency improvements [2,3] and has become an advanced corn planting model widely used in the wheat–corn rotation area in the Huang-Huai-Hai region. However, this complex operating environment causes large spatial differences in the soil's resistance [4,5], which aggravates the operating vibrations in planters and affects the seeding quality and fertilization depth [6]. The regulation of the pressure of a single unit on the ground is at the core of profiling control technology and is the key to building a high-quality seed furrow environment, which is directly related to subsequent seedling growth [7]. How to optimize and adjust the pressure of a single unit on the ground according to the soil's texture and the operating parameters to reduce the vibration of a single unit's operation has become a technical problem of common concern within academia and to enterprises, restricting the development of related equipment and increases in crop yields.

When the seeding vibration is uncontrollable, some scholars have improved the seeding quality by optimizing the working parameters of the seeding device [8,9]. However, relying on the optimization of the seeding device parameters can only reduce the impact of seeding operation vibrations. For this reason, some scholars consider directly reducing the vibrations during planter operation by optimizing the profiling mechanism to ensure consistency in seeding and the fertilization depth [10,11]. Since the passive profiling mechanism adjusts the ground pressure by adjusting the spring preload, once the preload is set, it cannot be adjusted in real time according to changes in the soil's resistance during operation. For this reason, Bai et al. [12] designed an electro-hydraulic active control system for planter row units, which indirectly achieved control of the seeding depth and compaction degree through real-time adjustment of the force applied to the profiling mechanism and the suppression mechanism; Jing et al. [13] designed a planter row unit electro-hydraulic downforce control system based on a PID closed-loop control algorithm, and its detection accuracy tolerance was within the range of $\pm 1.02\%$ under different field conditions. However, the above studies mostly focused on adjustment of the seeding depth and ground pressure, and there was a lack of effective control of the planter vibration. At the same time, research on the actual differences in the seeding rate between the inner and outer rows of planter units often focuses on curved operations of the planters [14] and rarely incorporates the difference in the seeding rate between rows caused by vibration when the planter row units operate in a straight line. During actual operation, the control parameters among planter row units are basically the same, which cannot reduce the difference in amplitude of each sowing row, resulting in an inconsistent amount of sowing in each row and ultimately reducing the sowing quality. To avoid the above problems, it is necessary to study suitable devices for actively reducing the vibration of the planter so that the amplitudes of the different row units tend to be consistent and the corn seeds would distribute across the rows more evenly. In order to develop a vibration reduction control system, it is first necessary to understand the vibration characteristics of the research object. For example, Zhou et al. [15] studied the optimal damping ratio for a suspension system based on an analysis of the dynamic deflection, vibration velocity and damping characteristics of the suspension system; obtained the optimal damping ratio that the required matching vibration damper should bear in the suspension system; and verified it through simulations and actual vehicle tests. Gao et al. [16] found that matching the shock absorber operating parameters based on the inherent vibration characteristics of the vehicle can help improve the performance of its power transmission system in vibration reduction. Therefore, in order to achieve precise control over the vibration of different unit rows, it is necessary to study the inter-row vibration characteristics of planter row units under different conditions so as to provide a reference for subsequent planter row unit vibration reduction control and seeding quality improvements.

In view of the above problems, this paper took a four-row no-till planter as the research object, and the specific objectives of this study were (1) to establish a field vibration model of the planter row unit and determine the factors affecting the vibration of the unit; (2) to analyze the influence of different factors on the actual operation of the planter; (3) to explore

the difference in the inter-row vibration characteristics of each planter row unit and the influence of the frame vibration on the vibration of the planter row unit; (4) to analyze the time–frequency characteristics of the planter and determine the excitation source affecting the difference in the inter-row vibrations of the planter row units; and (5) to propose improvement measures for laying a foundation for active vibration reduction control and optimization of design of the profiling mechanism in high-speed no-till planters.

2. Establishing a Vibration Model for a Planter

When a no-till planter is operating on a wheat stubble surface, it is affected by various factors and produces vibrations, which will affect the performance of the planter and the quality of the sowing, and thus has a negative impact on the seeding quality. By establishing a mathematical model of the vibration of a planter row unit, the influencing factors for the vibration of the planter row unit can be obtained, providing a theoretical basis for analyzing the vibration characteristics and parameter optimization.

Assuming that the height of the field surface relative to the reference plane is H and the wavelength of the surface undulation is l , the vertical motion of the planter row unit can be simplified as shown in Figure 1. Further, in order to establish a vibration model for the no-till planter row unit, the following assumptions were made: ① All components of the planter row unit are rigid bodies. ② The stiffness of the planter furrow opener, disc coulters, press wheels, ground wheels and gauge wheels are linearly related to the displacement. ③ The damping generated by the interaction between the furrow opener, the disc coulters, the press wheels, the ground wheels, the gauge wheels and the uneven surface of the field are linearly related to the speed, and the bounce is ignored. ④ Only the vertical action of the planter and the ground is considered.

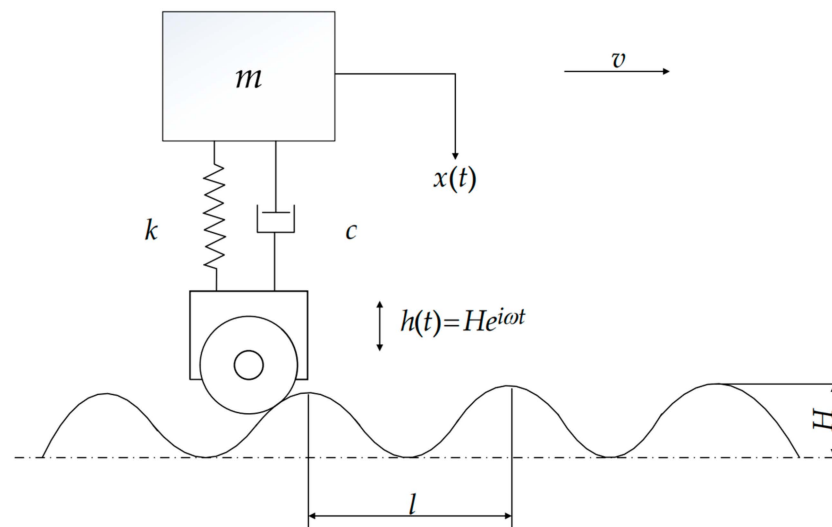


Figure 1. Simplified model of field vibration of planter row unit.

The autocorrelation function of a vibration signal from the no-till planter in the field is composed of sine and cosine functions [17,18]. Therefore, combined with Figure 1, it can be seen that the simple harmonic excitation of the planter row unit is

$$h(t) = He^{i\omega t} \quad (1)$$

where $h(t)$ is the simple harmonic excitation of the planter row unit, m ; H is the height of the field surface relative to the reference surface, m ; t is a certain moment during the seeding operation, s ; and ω is the excitation frequency, Hz .

Among them,

$$\omega = 2\pi \frac{v}{l} \quad (2)$$

where v is the working speed, m/s, and l is the wavelength of surface undulation. Then, the differential equation for the planter vibration is

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = \dot{c}h(t) + kh(t) = i\omega cHe^{i\omega t} + kHe^{i\omega t} \quad (3)$$

where m is the weight of the planter row unit, kg; k is the stiffness of the planter row unit, N/m; c is the damping coefficient of the planter row unit; and $x(t)$ is the absolute displacement of the planter row unit, m.

Let Formula (3) be

$$g = kH + i\omega cH \quad (4)$$

Substituting Formula (4) into Formula (3), then the differential equation for vibration of the planter row unit is simplified into

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = ge^{i\omega t} \quad (5)$$

The planter is damped in the field, causing the transient vibration to decay rapidly to zero. At this time, the planter only retains steady-state vibration. Therefore, in order to obtain a steady-state solution for the absolute displacement of the planter row unit, it is assumed that the specific solution for Equation (5) is

$$x(t) = Xe^{i(\omega t - \phi)} \quad (6)$$

where X is the amplitude of $x(t)$, m, and ϕ is the phase difference between the steady-state response and excitation, rad.

Substitute specific solution (6) into differential Equation (5) and divide both sides of the equation by $e^{i\omega t}$ to obtain

$$-m\omega^2 Xe^{-\phi} + i\omega cXe^{-\phi} + kXe^{-\phi} = g \quad (7)$$

Then,

$$X = \frac{ge^{\phi}}{-m\omega^2 + k + i\omega c} \quad (8)$$

Substituting Equations (4) and (8) into Equation (6), we get

$$x(t) = \frac{(kH + i\omega cH)e^{i\omega t}}{-m\omega^2 + k + i\omega c} \quad (9)$$

From Formula (9), it can be seen that in addition to its own inherent factors, such as the stiffness k and the damping coefficient c , the absolute displacement $x(t)$ of the planter row unit at time t is affected by its own weight m , the height H of the ground relative to the reference plane and the excitation frequency ω . According to Formula (2), the working speed v and the wavelength l of the surface undulation determine the magnitude of the excitation frequency ω . Therefore, for the planter row unit, the working speed and surface roughness affect the displacement of its simple harmonic motion. In simple harmonic motion, the relationship between acceleration, displacement and vibration frequency is shown in Formula (10), so the working speed and surface roughness also affect the vibration acceleration of the planter row unit.

$$a = f_0^2 s \quad (10)$$

where a is the simple harmonic motion acceleration, m/s²; f_0 is the frequency of simple harmonic motion, Hz; and s is the displacement of simple harmonic motion.

In addition to the row planter row units, a complete planter group also includes components such as the frame, the speed-change mechanism and the fertilization mechanism. Each mechanism has parts installed onto the frame, so the frame is subjected to excitation loads and generates vibrations that may also affect the planter row units connected to it.

During the seeding operation, the frame of the planter needs to be kept horizontal from front to back, and its crossbeam must also be parallel to the ground; otherwise, this will increase the resistance of the planter and cause uneven seeding depths [19]. By changing the total weight of the fertilizer in the fertilizer box in the no-till planter, the force acting on the frame can be changed, thereby affecting the vibration state of the frame and possibly affecting the planter row unit's operation.

In summary, it was determined that the vibration characteristics of the planter row unit were affected by the unit's working speed, the surface roughness and the unit's own weight.

3. Materials and Methods

3.1. The Experimental Scheme

According to the results of the theoretical analysis of the planter vibration, the experimental plan should include factors for characterizing the unevenness of the surface, the change in the weight of the planter and the working speed of the planter. Since the actual surface conditions are uncontrollable, stubble height was selected to simulate the unevenness of the ground. The wheat stubble height (A), the working speed of the planter unit (B) and the planter's operating weight (C) were selected as the experimental factors. According to the suggestions of relevant experts, a stubble height in the range of 15–25 cm was selected [20]. Considering that the working speed range of the no-till planter used in the experiment is 1.5–2.3 m/s, the working speed range of the unit was set to 1.5–2.5 m/s to ensure that it could cover the optimal working speed of the planter. The additional weight of the planter was divided from low to high according to the weight of the fertilizer poured into the fertilizer box. With the aim of simulating the actual working conditions, the total weight of the fertilizer in the two fertilizer boxes was changed to adjust the unit's operating weight. During the experiment, a situation in which no fertilizer was loaded into the two fertilizer boxes was defined as “low weight”, and the total weight of the fertilizer was 0; a situation in which half of the capacity of the two fertilizer boxes was filled with fertilizer was defined as “medium weight”, and the total weight of the fertilizer was around 220 kg; and a situation in which the two fertilizer boxes were filled with fertilizer was defined as “high weight”, and the total weight of the fertilizer was around 440 kg. Taking the root mean square (RMS) of the vibration acceleration in the X, Y and Z directions at each measuring point as the evaluation index, a 3-factor, 3-level orthogonal experiment was designed and carried out.

In order to study the influence of different factors on the vibration characteristics of the no-till planter under field operation conditions, based on a theoretical analysis and information obtained from our literature review, each experiment lasted for more than 70 s, from which 40 s of vibration experimental data under stable working conditions were extracted. Therefore, the three levels of the unit's working speed were set to 1.5, 2 and 2.5 m/s. The factor levels in the experiment are shown in Table 1, and the experimental plan is shown in Table 2. All experiments were conducted in a field environment, and the level of each group of experimental factors was set according to Tables 1 and 2. Therefore, the experimental site needed to be treated before the experiment.

Table 1. Experimental factor levels.

Level	Stubble Height A/cm	Working Speed B/m·s ^{−1}	Additional Weight C/kg
1	15	1.5	0
2	20	2	220
3	25	2.5	440

Table 2. Scheme of orthogonal experiment.

Experiment Number	Stubble Height A/cm	Working Speed B/m·s ⁻¹	Additional Weight C/kg
1	15	1.5	0
2	15	2	220
3	15	2.5	440
4	20	1.5	440
5	20	2	0
6	20	2.5	220
7	25	1.5	220
8	25	2	440
9	25	2.5	0

The stubble height was controlled by adjusting the harvester header height during mechanized wheat harvesting. During actual operation, stubble height is difficult to control accurately. For this reason, it was necessary to harvest at a certain distance first, measure the stubble height and adjust the header height until the stubble height met the requirements. Subsequently, the harvester header's height-adaptive technology was relied upon to ensure that the final stubble height fluctuated between 15 cm, 20 cm and 25 cm. The experiment site after wheat harvesting is shown in Figure 2a, and measurement of the wheat stubble's height is shown in Figure 2b.



(a) The experiment site.



(b) Measuring wheat stubble height.

Figure 2. Preparation before the experiment.

3.2. Experimental Equipment

3.2.1. The Tractor and No-Till Planter

The vibration experiment was conducted in Huimin County, Binzhou City, Shandong Province, on a wheat stubble surface in June 2024. The surface of the experimental site is generally flat, with little undulation. The distance that the no-till planter can travel in the experimental field is about 150 m. This study used a Dongfanghong LX1804 tractor (YTO Co., Ltd., Luoyang, China) mounted with a Debont 2BMG-4 no-tillage planter (Beijing Debont Technology Co., Ltd., Beijing, China). This type of planter has 4 rows of planter row units, and this type of the planter is widely used in the region; the planter and the tractor were connected by a three-point hitch.

In order to reduce the influence of additional variables on the signal acquisition, sensors were arranged in symmetrical positions within the planter row unit group as blank controls, as shown in Figure 3. A total of 7 sensors were arranged in the experiment, and each sensor was numbered, corresponding to measurement points 1 to 7. Among them, sensors 1 to 4 were installed on different planter row units from left to right to measure the

corresponding vibration signals of the units; sensors 5 and 6 were arranged on the left and right sides of the frame to obtain vibration signals at both ends of the planter; and sensor 7 was located in the center of the planter frame to obtain the frame's vibration signal. The arrangement of the measuring points is shown in Figure 3.

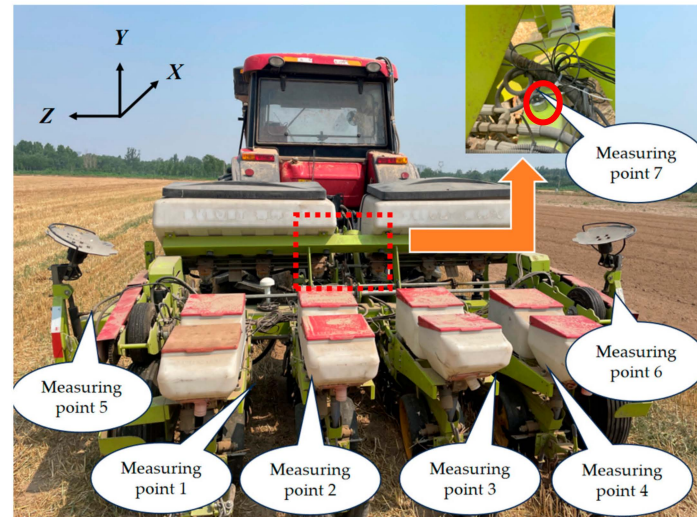


Figure 3. Arrangement of measuring points.

To facilitate experimental data processing, the X, Y and Z signal channels of the sensor cable were set to correspond to the forward, vertical and horizontal directions of the no-till planter unit, respectively, so that the experimental vibration data on various parts of the planter unit were unified in the X, Y and Z directions.

3.2.2. The Data Acquisition System

This study used a vibration signal analysis system to collect vibration data on the planter unit when it was operating on the wheat stubble surface. The vibration signal analysis system for the no-till corn planter unit mainly includes BWJ13533 three-axis acceleration sensors (Shanghai B&W Sensing Technology Co., Ltd., Shanghai, China), sensor cables, a dynamic signal analyzer, engineering data management (EDM 8.0) system software and a laptop computer. The system's working principles are shown in Figure 4, and the lines of different colors in the upper computer interface (Figure 4) represent the vibration acceleration data inputted by each channel. The main performance parameters of the vibration experiment analyzer and the sensors are shown in Table 3. The acceleration sensors were fixed in the experimental position onto the planter unit using magnetic bases. During the experiment, three-axis acceleration data generated at the sensor fixing points were collected for subsequent analysis and processing.

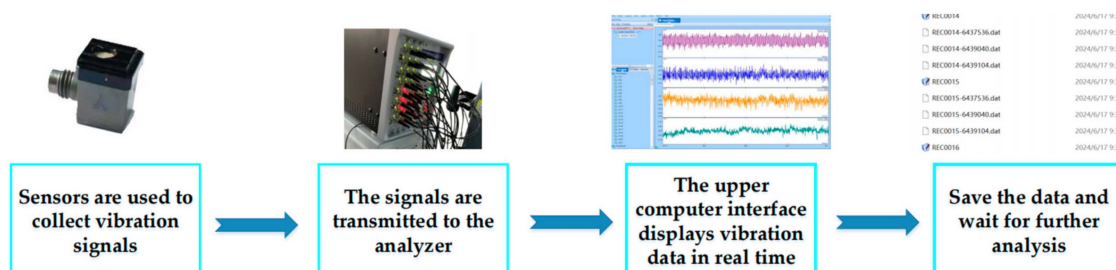


Figure 4. Vibration signal analysis system.

Table 3. Main performance parameters of experimental equipment.

Device Name	Performance Indicators	Parameter Value	Manufacturer
Spider-80Xi dynamic signal analyzer	Input channels	32	Crystal Instruments Corporation, Santa Clara, CA, USA
	Input dynamic range/dB	150	
	Maximum sampling rate/kHz	102.4	
BWJ13533 sensor	Range/g	±100	Shanghai B&W Sensing Technology Co., Ltd., Shanghai, China
	Sensitivity/(mV·g ^{−1})	50	
	Lateral sensitivity/%	≤5	
	Frequency range/Hz	1~6000	

During the experiment, the vibration data were sampled continuously. According to the sampling theorem [21], the sampling frequency should be at least twice the analysis frequency. Therefore, the sampling frequency of the dynamic signal analyzer was set to 2560 Hz and the analysis frequency to 1152 Hz. In order to reduce errors caused by other factors, the vibration signals of the unit were collected in each experiment, and the signals were extracted under stable conditions.

3.3. Experimental Data Processing

3.3.1. Time Domain Analysis and Processing

During the experiment, the sensors collected the vibration acceleration in the X, Y and Z directions of the measuring point and calculated the RMS of the acceleration in the three directions as an evaluation index. The calculation formula for the RMS is shown in Formula (10). Then, the experimental results were intuitively analyzed and variance-analyzed based on the RMS values in the X, Y and Z directions, and the order of primary and secondary factors and the significance of each factor could be determined so as to arrange the influence of the three factors on the vibration characteristics of the planter in order.

$$\text{RMS} = \sqrt{\frac{1}{N} \sum_{k=1}^N x_k^2} = \sqrt{\frac{x_1^2 + x_2^2 + x_3^2 + \cdots + x_k^2}{N}} \quad (11)$$

where x_k is the vibration acceleration amplitude, m/s², and N is the number of signals collected.

3.3.2. Time–Frequency Analysis and Processing

In the experiment, four acceleration sensors were fixed onto the four rows of planter row units, and the remaining three sensors were located on the left and right ends and at the center of the frame. By analyzing the vibration signals of each planter row unit, the vibration characteristics of each planter row unit and the frame could be obtained, and the similarities and differences in the vibration between units could be compared. The vibration data measured in this paper are non-stationary signals. The time–frequency analysis method is an effective means of processing non-stationary vibration signals [22,23], which can simultaneously reflect the time domain and frequency domain characteristics [24,25]. In the experiment, Matlab 2022a software was used to perform time–frequency analysis of the collected vibration acceleration signal. In Matlab software, the pspectrum function is used to analyze signals in the frequency domain and the time domain, and short-time Fourier transform (STFT) is performed in combination with the use of the spectrum function to obtain time–frequency analysis diagrams for the signals. Assuming the vibration signal is $g(t)$, the definition of STFT is as shown in Formula (11). On this basis, the persistent spectrum of the signal can be obtained, which is a time-frequency view that shows the percentage of the time that a given frequency is present in a signal. The longer a particular frequency persists in a signal as the signal evolves, the higher its time percentage and thus the brighter or “hotter” its color in the display [26].

$$\text{STFT}(t_0, f) = \int_{-\infty}^{+\infty} g(\tau)w(\tau - t)e^{-j2\pi f\tau}d\tau \quad (12)$$

where t_0 is the starting time, s; f is the frequency, Hz; τ is the time, s; $w(t)$ is the window function; and $\text{STFT}(t_0, f)$ is the frequency spectrum of the signal component with frequency f at time t_0 .

4. Results and Discussion

4.1. Experimental Results

The time domain vibration data on the seven measuring points in each experiment were recorded, and the RMS values of acceleration were calculated. The results are shown in Figure 5.

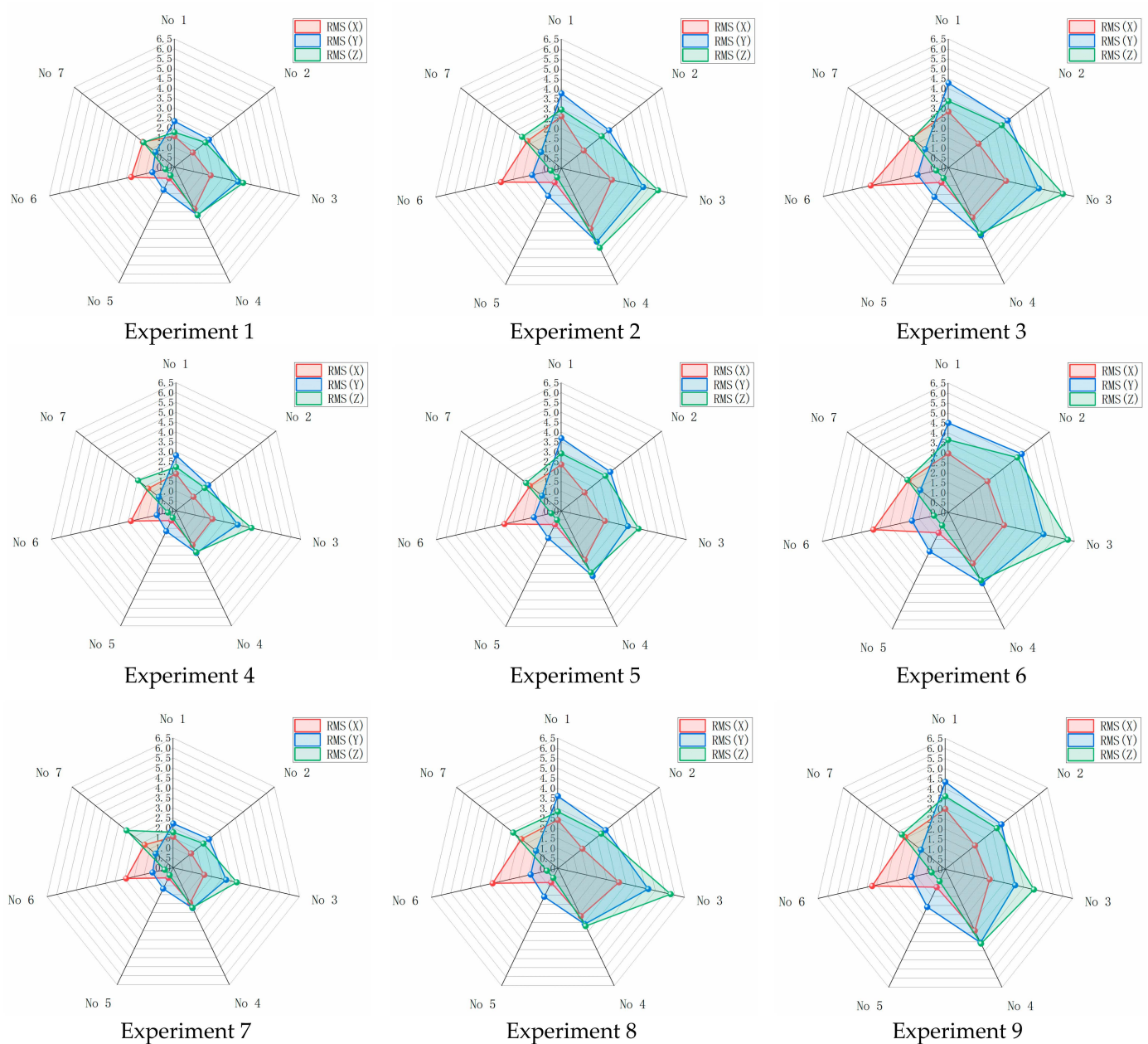


Figure 5. RMS values of each measuring point in the X, Y and Z directions in 9 experiments.

As shown in Figure 5, among the four rows of planter row units, the units in which measuring points 1 and 2 were located always had the largest amplitude in the Y direction; the units where measuring points 3 and 4 were located had a larger amplitude in the Z direction. The trend in the change in amplitude at measuring point 5 on the left end of the frame was similar to that at measuring points 1 and 2; the amplitude at measuring point 6 on the right end was the largest in the X direction; and the change in amplitude of measuring point 7, located in the center of the frame, was similar to that of measuring points 3 and 4. According to the amplitude size of the measuring points in each direction, it can be inferred that measuring points 2, 3 and 7 were close to the three-point hitch and were affected by the force of the tractor transmitted by the hitch, and the amplitude in the Z direction increased; measuring points 5, 6 and 7 were located on the frame and were directly affected by the tractor's traction, which produced a large vibration acceleration in the X direction.

4.2. Effects of Different Factors on the Vibration Characteristics of the Planter

In order to explore the influence of different experimental factors on the vibration of the planter, according to the results of calculating the RMS values of the seven measuring points in each direction, the RMS in the X, Y and Z directions was used as an evaluation indicator, and the range of each factor in the orthogonal experiment was calculated to obtain its weight of influence on the experiment results [27]. The range results are shown in Table 4.

Table 4. The range of each factor in the experiments.

Measuring Point	Factors	X Direction/ $\text{m} \cdot \text{s}^{-2}$	Y Direction/ $\text{m} \cdot \text{s}^{-2}$	Z Direction/ $\text{m} \cdot \text{s}^{-2}$
1	Stubble height	0.0819	0.2793	0.2334
	Working speed	1.2683	1.9211	1.6241
	Additional weight	0.0676	0.1153	0.0373
2	Stubble height	0.1970	0.3511	0.3814
	Working speed	0.9608	2.5319	1.7841
	Additional weight	0.1721	0.3722	0.2957
3	Stubble height	0.1551	0.4266	0.3001
	Working speed	0.9131	1.7132	1.9456
	Additional weight	0.5370	0.7376	1.1821
4	Stubble height	0.3277	0.4063	0.4334
	Working speed	0.9228	1.4795	1.4403
	Additional weight	0.3715	0.3827	0.4607
5	Stubble height	0.0660	0.1294	0.0232
	Working speed	0.3951	1.1864	0.2312
	Additional weight	0.1082	0.1730	0.0490
6	Stubble height	0.1142	0.0411	0.0196
	Working speed	1.5339	2.0698	0.2443
	Additional weight	0.2659	0.1492	0.0320
7	Stubble height	0.1093	0.0422	0.5753
	Working speed	0.6065	1.0309	0.0930
	Additional weight	0.0344	0.0731	0.3687

As shown in Table 4, the range of working speed was generally greater than the range of stubble height and additional weight, so working speed had the greatest experimental impact on the vibration of the planter. For the planter unit, the range of the RMS in the X direction was relatively consistent, while the range of the RMS in the Y and Z directions was quite different; except for at measuring point 3, the effects of stubble height and additional weight on the unit's vibration were similar. For the planter frame, the effect of stubble height on its amplitude was greater than that of additional weight. In general, the change in

amplitude of the measuring points on the right side of the planter was greater than that for those on the left side. However, given it is a symmetrical structure, the vibration of the left and right sides of the planter should also be similar [28]. Therefore, it was inferred that the right-side components of the planter were subject to greater excitation, resulting in more violent vibration of the right-side components, especially the unit on which measuring point 3 was located.

According to the results of the range analysis, working speed was the most important factor affecting the vibration of the planter. The stubble height and the weight of the implement also had different effects on the vibration of different parts. Therefore, observing the amplitude changes of each measuring point in Figure 5 when the speed was the same and the other factors were at different levels. It was found that under low-speed conditions (experiments 1, 4 and 7), when the stubble height was the highest (25 cm), the amplitude of each component of the planter increased significantly; when the stubble height was 20 cm, the amplitude of each measuring point was similar to the amplitude when the stubble height was 15 cm, which was speculated to be related to the increase in the weight of the implement. As the working speed increased, the proportion of the influence of the stubble height and the weight of the implement on the vibration of the planter decreased, and the difference in the amplitude of the planter at the highest speed (experiments 3, 6 and 9) was small. Therefore, when the planter operated at a low speed, the lower the stubble height, the lower the amplitude of the planter, and the weight of the implement increased the gravity on the planter, suppressing its vertical vibration.

In summary, before seeding, the stubble height can be reduced by lowering the height of the wheat harvester's header, thereby suppressing excessive vibration of the planter. Excessive mechanical vibration will increase the missed seeding rate of the planter [29]. Considering the impact of vehicle speed on planter vibration, various monitoring sensors were used to monitor the seeding qualification rate, the missed seeding rate and the reseeding rate in real time during the operation of the planter. Adjusting the working speed of the planter based on the seeding quality parameters obtained could reduce the impact of severe mechanical vibrations on its seeding quality.

4.3. Analysis of the Vibration Characteristics of Planter Row Units in Different Rows

In order to explore the influencing factors of the vibration characteristics of the planter so as to reasonably set the seeding parameters and improve the sowing quality, it was necessary to analyze the vibration data for each row of the planter row units. As shown in Figure 5, for all the planter row units, the amplitude in the X direction was always the smallest. Therefore, only the amplitudes in the Y and Z directions, where there were large differences among the units, were studied. The average values of the RMS in the Y and Z directions at measuring points 1–4 in nine groups of experiments were calculated and used to characterize the vibration intensity of each measuring point in the Y and Z directions. The results are shown in Table 5.

Table 5. Average value of RMS of planter row units in Y and Z directions.

Measuring Points	Y Direction/ $\text{m} \cdot \text{s}^{-2}$	Z Direction/ $\text{m} \cdot \text{s}^{-2}$
1	3.503	2.790
2	3.116	2.799
3	3.857	4.688
4	3.327	3.321

As shown in Table 6, the amplitudes of the units in the Y direction were similar, but the amplitudes in the Z direction were quite different. Among them, the amplitudes of measuring points 1 and 2 in the Z direction were almost the same, while measuring point 3 had the largest amplitude in the Z direction, followed by measuring point 4. The data in Figure 5 and Table 4 show that the amplitudes of the components on the right side of the planter row unit group were greater than those of the components on the left side,

which was particularly significant in the Z direction. It was speculated that the right side of the planter was affected by more excitation sources [30], resulting in a large increase in the amplitude of the units on the right side in the Z direction. According to the structural characteristics of the ground-wheel-driven planter, it is speculated that the long-term operation of the planter row unit may cause abnormalities in the ground wheel and the chain drive on the right side [31], resulting in an increase in the amplitude of the units on the right side of the planter row unit and the right half of the frame. In response to the problems with the aforementioned ground-wheel-driven planter, the traditional ground wheel drive can be replaced by an electric motor drive or a hydraulic motor drive to avoid the disadvantages of the mechanical drive method.

Table 6. *p*-values of ANOVA of measurement points on the frame.

Measuring Points	Factors	<i>p</i> Values		
		X Direction	Y Direction	Z Direction
5	Stubble height	0.806	0.728	0.904
	Working speed	0.104	0.087	0.091
	Additional weight	0.596	0.581	0.653
6	Stubble height	0.370	0.872	0.881
	Working speed	0.003	0.028	0.046
	Additional weight	0.101	0.353	0.725
7	Stubble height	0.810	0.951	0.022
	Working speed	0.127	0.151	0.462
	Additional weight	0.978	0.844	0.055

4.4. Analysis of the Vibration Characteristics of the Planter Frame

The planter frame was directly connected to the planter row unit and was subject to the forces exerted by the three-point hitch, the ground wheel and other components. Therefore, studying the changes in its RMS values in all directions is helpful to analyze the excitations received by the planter row units inside the planter group. A variance analysis of the RMS values in all directions at measuring points 5, 6 and 7 was performed, and the *p* values were obtained, as shown in Table 6.

As shown in Table 6, in most cases, the *p* value of each factor was greater than 0.05. Only the working speed had a significant effect on the vibration at measuring point 6 ($p < 0.05$), and the stubble height had a significant effect on the vibration at measuring point 7 in the Z direction ($p < 0.05$). From the results of the orthogonal experiment, it could be seen that the vibration data at measuring points 5 and 6 on the left and right sides of the planter were very different. Compared with measuring point 5, the RMS value generated by measuring point 6 in the X direction was more than $1.5 \text{ m}\cdot\text{s}^{-2}$ higher, while the RMS values of the two in the Y and Z directions were relatively close, and this situation occurred in all nine groups of experiments. According to the results of the variance analysis, the working speed had a very significant effect on the vibration of measuring point 6 in the X direction ($p < 0.01$). It can be inferred that the excitation affecting the right end of the frame acted in the X direction and was closely related to the speed of the planter. Combined with the previous inference in Section 3.3, this excitation was likely to come from the ground wheel on the right side of the planter. It is necessary to check the operation of the ground wheel and its chain drive process to prevent it from affecting the seeding of a planter row unit, thereby reducing the rates of missed seeding and repeated seeding [32]. In comparison, working speed was also a factor that has a greater impact on the frame's vibration; meanwhile, stubble height and additional weight only had a greater impact on the Z-direction vibration at the center of the frame, and their impact on the left and right ends was not significant. In these results, the impact of additional weight on the frame's vibration was found to be slightly greater than that of stubble height. The *p* values of both at measuring point 7 were greater in the X and Y directions, and the lower impact

should be related to the fact that measuring point 7 was mainly subjected to the force of the three-point hitch.

Comprehensive analysis of the vibration characteristics of the seeding unit and the frame showed that there was a high probability that the power transmission process in the right ground wheel of the planter was abnormal, resulting in a significant increase in the RMS value of the right planter row units in the Z direction, while the units farther away from the right ground wheel were almost unaffected. Therefore, in addition to its operating status, the impact of the ground wheel on the vibration of the planter row unit was also related to the distance between the unit and the ground wheel. For multi-row mechanically driven planters with large operating widths, the installation position of each ground wheel should be carefully considered during their design.

4.5. Analysis of Time–Frequency Characteristics of Planter Vibration

In Sections 4.1–4.4, it was determined that there were more excitation sources on the right side of the planter, resulting in more intense vibration amplitudes in the right parts. In order to obtain the characteristics of the aforementioned vibrations in the frequency domain, time–frequency analysis was performed on the data at each measuring point. Considering the normal operating speed of the 2BMG-4 no-till planter, a travel speed of 2 m/s was selected for all experimental groups, and then the experimental group with the lowest amplitude was selected. Combined with the inference within the analysis of the inter-row vibration in the planter row units in Section 3.3, it was ultimately determined that the Z-direction vibration signal was the most representative for each measuring point in the fifth group of experiments, and a time–frequency analysis was performed on these data.

According to the literature [17,33] and the results of Fast Fourier Transformation (FFT) analysis of the time domain data from our experiment, the distribution of the vibration energy of the planter was concentrated in the lower-frequency range. Therefore, the frequency range of the time–frequency analysis was set to 0–200 Hz. Matlab 2022a software was used to perform a time–frequency analysis on the time domain signals of all the measuring points in the experiment, and a time–frequency diagram and a persistent spectrum of the vibration signals of the four planter row units and the frame in each group of experiments were obtained. A time–frequency diagram for each measuring point in the Z direction in the fifth group of experiments is shown in Figure 6, and corresponding persistent spectrum diagrams are shown in Figure 7.

As shown in Figure 6, the vibration energy of all the planter row units in the experiment was concentrated in the low-frequency range of 5–45 Hz, but there were differences in the frequency range above 90 Hz. Measurement points 2 and 3, located on both sides of the central axis of the planter, had different vibration energy distributions from measurement points 1 and 4 in the range of 105–120 Hz, while measurement point 7, located at the connection between the frame and the three-point hitch, had the most significant vibration energy distribution in this range. Considering that the engine load will affect the mounted equipment when a tractor is mounted with equipment [34], it is speculated that the reason for this may be that the three-point hitch transmitted the power of the tractor, which, in turn, caused the planter frame and the planter row units close to the frame to produce vibrations of different intensities in the same frequency range. However, because measurement points 1 and 4 were far away from the three-point hitch, they were far less affected than measurement points 2 and 3, making their vibration energy distribution in the range above 50 Hz more stable than that at measurement points 2, 3 and 7; measurement point 7 also had a distribution of vibration energy in multiple frequency domain ranges, which may have been related to the frame being subjected to multiple external excitations, such as that of the three-point hitch, the planter row units and the planter transmission components. The signal energy at measuring points 5 and 6 on the left and right ends of the frame was mostly concentrated within the 0–9 Hz range. The vibration energy at higher frequencies may be transmitted after the forced vibration of the planter row unit and the center of the frame. Further, the persistent spectrum was analyzed to obtain the frequency domain

distribution curve of the vibration signal. The brighter the signal line at a certain frequency in the persistent spectrum, the longer the signal duration of the frequency was, as shown in Figure 7.

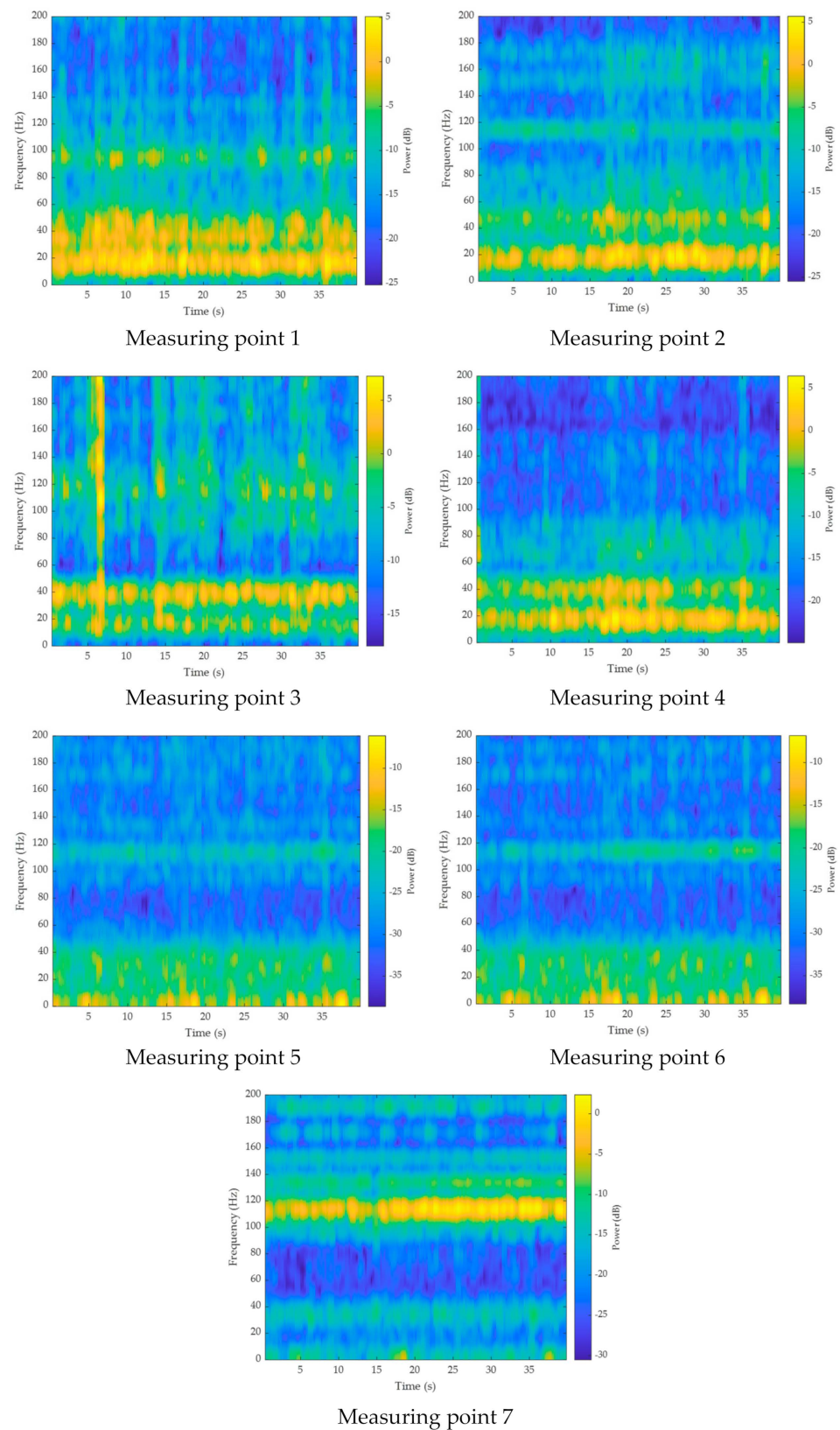


Figure 6. Time–frequency diagrams in the Z–direction for each planter row unit and the frame.

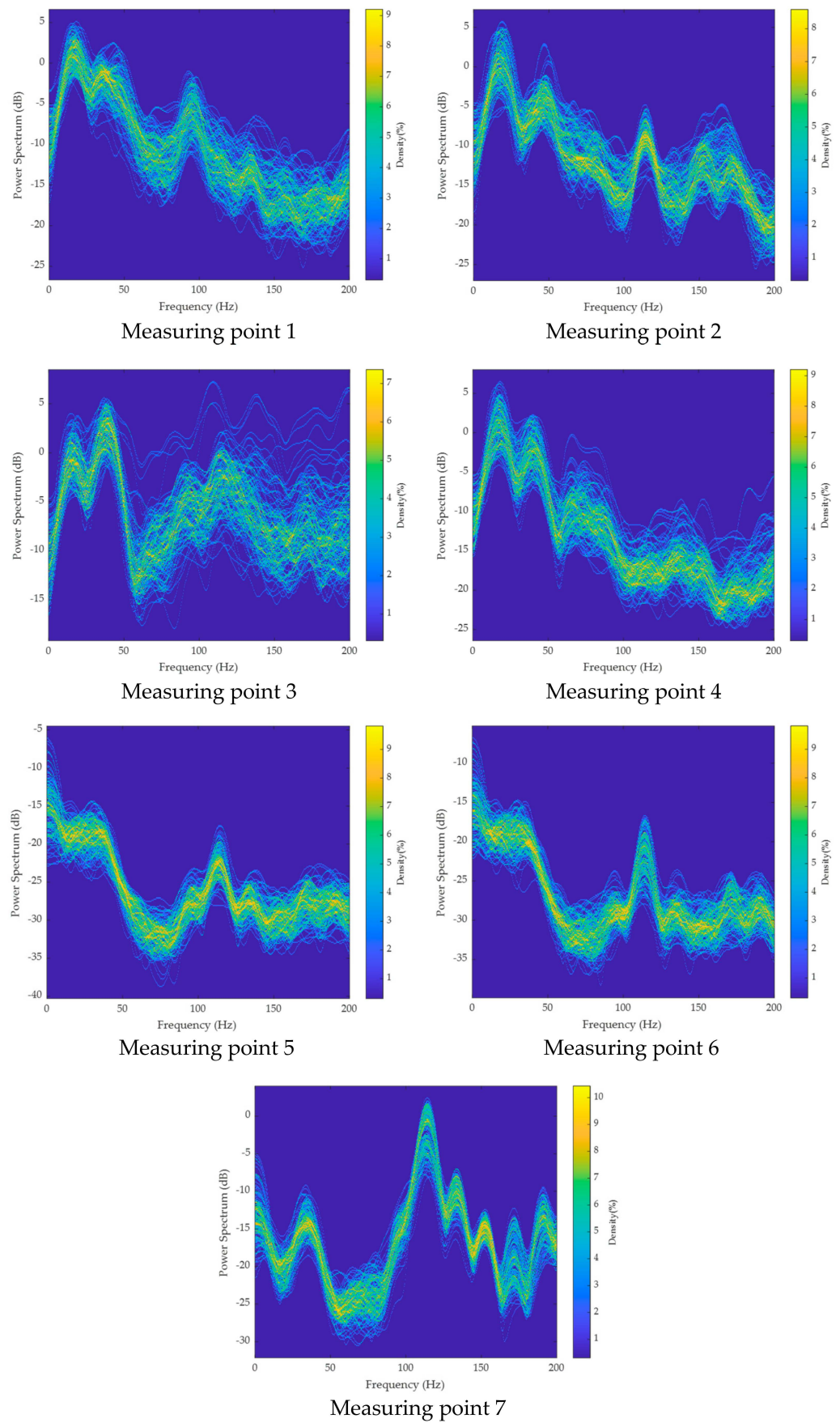


Figure 7. Persistent spectra in the Z-direction of each planter row unit and the frame.

As shown in Figure 7, the two amplitude peaks of the planter row unit in the low-frequency range (below 100 Hz) were near 20 Hz and 45 Hz, respectively, and the peak duration was relatively long. In the frequency domain range above 100 Hz, the frame and the inner row planter row units generated new peaks near 115 Hz, while the outer planter row units did not have peaks at this frequency, which might have been related to their distance from the center of the frame. The vibration signals of the given frequency at measuring points 1 and 4 were more concentrated than those at measuring points 2, 3 and 7. The reason for this might be that the intensity and number of external excitations received at measuring points 1 and 4 close to the two sides of the planter were less and fewer than those received at the center of the planter. Therefore, the planter row units at measuring points 1 and 4 vibrated more steadily, and the vibration had less impact on their seeding quality. Measuring points 5, 6 and 7 were located at different positions on the frame, and the magnitude and direction of the excitations were different, but the peak frequency of their signals was the same. Near 115 Hz, measuring points 5, 6 and 7 all generated peaks, and it was speculated that the frame resonated near 115 Hz.

Combined with the analysis of the time–frequency diagrams and the persistent spectrum diagrams, it can be seen that the three-point hitch transmitted the excitation from the tractor, which led to differences in the vibration of the inner and outer planter row units. Therefore, in actual field operations, it is necessary to consider individual control of the operating parameters of each unit. Since the driving force of the ground-wheel-driven planter comes from the same set of gears [35], it is unable to control a row of units individually. Therefore, electronic control of seeding is carried out on the basis of changing the drive mode of the planter to achieve variable seeding, which can effectively reduce inconsistency in seed spacing, row spacing and the soil penetration depth caused by this difference [36]. Since the three-point hitch that transmits the tractor's driving force will inevitably affect the inner planter row units, it is necessary to consider installing different passive profiling mechanisms onto planter row units in different positions on the planter or setting the active profiling control parameters according to the position of the planter row unit to ensure that the effect of vibration on seeding is controlled as required and the problems of re-seeding and missed seeding caused by severe vibrations are avoided.

5. Conclusions

- (1) This paper calculated the absolute displacement of the planter in steady-state operation based on the kinematic relationship between the planter row units and the field surface in the vertical plane, which clarified the factors affecting the vibration of the planter, such as the working speed, the total weight of the unit and the uneven surface of the field. Based on the results of our theoretical analysis, an experimental plan for the vibration characteristics of the planter was proposed.
- (2) The results of the orthogonal experiment showed that the working speed was the most important factor affecting the vibration of the planter in all directions, while the stubble height and additional weight of the planter had little effect on the vibration of the planter, being much lower than that of the working speed. Therefore, in studying reducing planter vibrations, maintaining a suitable working speed is another key point in addition to the vibration reduction system.
- (3) The average amplitudes of the four planter row units were similar, except in the lateral direction. Meanwhile, in the lateral direction, the range of the average amplitude of each planter row unit reached 1.898 m/s^2 , and the average amplitude of the center of the frame in the lateral direction was more than 2 m/s^2 higher than that of the two ends, indicating that violent vibration of the center of the frame in the lateral direction drove the increase in the amplitude of the planter row units in this direction.
- (4) The vibration energy of different planter row units was mainly distributed around 10–50 Hz. Meanwhile, the inner planter row units generated a large amount of vibration energy, in the range of 110–120 Hz, which coincided with the resonant frequency interval of the center of the frame. It was speculated that the vibration of the frame

in this frequency range drove the vibration of the inner planter row units. Finally, it was determined that the main excitation sources affecting the difference in inter-row vibration of the planter row units include the three-point hitch and the ground wheel transmission device.

- (5) To reduce the difference in the vibration among the planting row units, this paper posits that active profiling control technology could be used in the future to set different control parameters according to a unit's position or to set different preloads for the profiling springs of different planter row units so as to stabilize the sowing parameters for each row and improve the sowing quality.

Based on the results of our theoretical analysis, this paper studied the influence of various factors on the vibration characteristics of a four-row no-till planter and provided a preliminary idea on setting the active vibration reduction control parameters for planter units under different operating environments, as well as independent control of the operating parameters for each row of planters. Future research should focus on integrating the above control methods into the planter to achieve real-time regulation of various parameters during the planter's operation.

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