

Article

Toward Sustainable Mechanization of Peanut Harvesting: Finite Element Analysis and Field Verification of a Composite Soil Cleaning Device for Use in Clay Soil

Zhaoyang Yu ^{1,†}, Yu Jin ^{1,2,†}, Hongbo Xu ¹, Zhixia Liu ², Feng Wu ¹, Zhichao Hu ¹, Fujie Ding ^{3,*} and Fengwei Gu ^{1,4,*}

¹ Nanjing Institute of Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, Nanjing 210014, China

² College of Engineering, Shenyang Agricultural University, Shenyang 110866, China

³ Linshu Dongtai Machinery Co., Ltd., Linyi 276700, China

⁴ Chinese Academy of Agricultural Sciences, Beijing 100081, China

* Correspondence: 13685490550@163.com (F.D.); gufengwei@caas.cn (F.G.)

† These authors contributed equally to this work.

Abstract

Targeting the problems of low reliability and cleaning quality during the soil cleaning process of semi-feed peanut combine harvesters under clay soil conditions, a composite soil cleaning device with “up-and-down flapping and lateral rotation” was designed. To prevent strength failure within the device, finite element static analysis, explicit dynamic simulation and modal analysis were conducted. The rationality and reliability of the structural design of the two soil cleaning devices (the up-and-down flapping type and the lateral rotation type) were verified. The simulation results show that the maximum static and dynamic stresses of the up-and-down flapping-type soil cleaning device are far lower than the material yield strength. Even under extreme impact loads from hard impurities such as stones or roots, it still maintains elastic deformation and avoids permanent failure due to plastic deformation. The maximum stress of the lateral rotary-type soil cleaning device is also far below the material yield limit. Its dynamic characteristics meet the requirements posed by the extreme working conditions in the field, and its natural frequency lies outside the vibration range experienced under working conditions, demonstrating its excellent vibration resistance performance. Meanwhile, reliability assessments and cleaning quality measurements were carried out on the device. The test results show that the soil cleaning device can meet the structural strength requirements for semi-feed peanut combined harvesting under clay soil conditions. The soil cleaning device improves the sustainability of peanut production in terms of energy consumption, soil conservation, and economic viability. These research results provide technical support for peanut harvesting and the sustainable development of the peanut industry.

Academic Editor: Sanjay Nimbalkar

Received: 19 August 2026

Revised: 2 October 2026

Accepted: 8 October 2026

Published: 16 October 2026

Copyright: © 2026 by the authors.

Licensee MDPI, Basel, Switzerland.

This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

Keywords: sustainable mechanization; reliability; strength; composite soil cleaning device; finite element analysis; semi-feed peanut harvester; clay soil

1. Introduction

Peanuts, the most internationally competitive, high-quality, and advantageous oil crop in China, are also an important source of edible vegetable oil and protein. They hold a crucial position in the national economy and social development [1,2]. The annual planting area for peanuts in China is over 4.7 million ha, with an output of over 17 million tons. The planting area and output consistently rank second and first in the world, respectively, and are showing an upward trend [3–5]. According to statistics, nearly 1.6 million ha of peanuts are planted in the hilly and mountainous areas of China, accounting for one-third of the total peanut planting area [6,7]. The peanut planting fields in these areas are mostly composed of clay soil, with small and scattered fields and complex terrain and landforms, which severely restricts the level of mechanized peanut harvesting that can be achieved [8–11]. As of 2022, the mechanized harvesting rate of peanuts in typical hilly and mountainous areas such as Sichuan and Guangdong was less than 2%, far lower than the national average rate of 53.90%. There is no specialized peanut harvesting equipment for hilly and mountainous areas with clay soil; the current harvesting equipment mainly consists of existing and improved small-scale agricultural machinery, such as small peanut single-row excavators and small picking machines. Their availability and usage are extremely limited, and their overall reliability is poor, with a high failure rate, with the harvesting quality failing to meet the requirements of local farmers [12–16].

Peanuts are underground fruit crops, and thus cleaning soil from around the pod is a necessary process during harvesting [17–19]. After nearly a decade of development in China, some soil cleaning technologies and devices for peanuts have been developed [20–22]. However, all these studies are focused on sandy or sandy-loamy soils in plains, and there are relatively few studies on soil cleaning techniques for peanuts under clay soil conditions in hilly and mountainous regions. Compared with sandy or sandy-loamy soils in plains, high-performance soil cleaning during peanut harvesting is extremely difficult in hilly and mountainous areas with clay soil. The reason for this is that in these areas, the peanut vines contain a large amount of soil, there are many hard, large soil clumps, and the clay soil easily adheres to the pods and the soil-cleaning equipment, thereby increasing their weight. These factors double the cleaning load and make it difficult to ensure the reliability of the cleaning components, which become prone to strength failure. Meanwhile, the high soil content, combined with the fact that the soil easily adheres to the pods and components, results in low-efficiency soil cleaning. At present, there are five main methods for soil cleaning commonly used in flat areas with sandy or sandy-loamy soils [23–25].

- (1) **Swing-Plate Soil Cleaning Technology.** An eccentric mechanism is used to drive two parallel vibrating plates to form a synchronous swing system. During operation, the conveying device clamps the peanut vines, enabling the pod parts to pass through the space between the two plates. Under the action of synchronous swing, soil stripping is achieved.
- (2) **Swing-Pole Soil Cleaning Technology.** An eccentric mechanism is used to drive two parallel vibrating poles. During operation, the conveying device clamps the peanut vines and transports them backward. The peanut vines pass through the space between the two poles. Through reciprocating swinging, the vibrating poles strike the vine parts, causing the vines below the conveying device to oscillate, thus dislodging the soil around the pods.
- (3) **Differential Chain Pole Soil Cleaning Technology.** A rotating-chain-pole lifting device is placed below the conveying device. The movement speed of the lifting device is lower than that of the conveying device, and the peanut pod attaches to the

chain poles. Due to the speed difference, the peanut pod will slide along the chain poles, thus removing the adhered soil.

- (4) Reciprocating Vibration Screen Soil Cleaning Technology. This technology features a vibrating screen to sieve out the soil. During operation, the peanut vines, pods, and soil are fed into the vibrating screen together. Under the reciprocating vibration of the screen, the soil is separated from the pods, with the soil falling off the screen.
- (5) Lifting Pole Soil Cleaning Technology. Similar to the reciprocating vibrating screen technology, during transportation of the peanut vines, pods, and soil, which is achieved using poles, the soil is separated from the pods and seedlings by shaking the poles.

The above-mentioned soil cleaning technologies perform well in sandy or sandy-loamy soils: the cleaning components have good reliability, and the cleaning efficiency meets the operational requirements. However, in areas with clay soil, their adaptability and reliability are poor; the failure rate of the cleaning components is high, and the quality of soil cleaning is low. Continuous cleaning operations cannot be achieved. Therefore, the above-mentioned soil cleaning technologies cannot be directly adopted in clay soil in hilly and mountainous areas, and the development of reliable and high-quality soil cleaning technologies and peanut harvester components that are suitable for these conditions is urgent.

In response to the aforementioned issues, the research object of this study is the soil cleaning device of a semi-feed peanut combine harvester. Aiming to solve the problems of its low reliability and poor soil-cleaning quality under clay soil conditions, an “up-and-down flapping–lateral rotation” soil composite cleaning scheme is proposed. The design and finite element analysis of key components of the soil cleaning device are presented. In addition, the reliability and cleaning quality of the device are evaluated through field tests, and the rationality of its design is comprehensively verified. The research results can provide a reference for the structural and strength design of soil cleaning devices for peanut combine harvesters used in clay soil.

2. Design of the Composite Soil Cleaning Device

2.1. Overall Scheme Configuration

Taking into account factors such as the soil cleaning rate, loss rate, energy consumption and cost, and in view of the clay soil working conditions, an “up-and-down flapping–lateral rotation” soil composite cleaning scheme was adopted. The structure of the composite soil cleaning device is shown in Figure 1 and mainly consists of two core working components: up-and-down flapping-type and lateral rotary-type soil cleaning devices. Combining these features combines the advantages of mechanical vibration to effectively enhance the soil cleaning efficiency and minimize peanut pod loss. The up-and-down flapping-type device efficiently cleans soil through the rapid reciprocating motion of a flexible striking plate, thereby reducing the risk of pod loss; the lateral rotary-type device utilizes the frictional force to remove the remaining soil and enhance the soil cleaning efficiency.

The soil cleaning process is generally divided into two stages: In the first stage, the up-and-down flapping-type device plays a crucial role. This device is composed of soil-striking grid plates that move up and down. Under the action of the driving device, it vibrates at a high frequency along a preset trajectory. Through various soil-dislodging mechanisms such as mechanical impact and centrifugal force, the plates can break the bond between the soil and the pods, thereby efficiently detaching the soil. The first stage is responsible for removing most of the soil from the surface of the pods. Then, in the second stage, the peanut vines are sent into the lateral rotary-type soil cleaning device. This device has multiple striking poles, which are installed on a rotating shaft and thus rotate at high speed. The centrifugal force generated by this device can break up and

remove loose soil. At the same time, with its combined friction effect, it significantly improves the cleaning efficiency and effectively prevents soil re-attachment. Due to the properties of clay soil, the peanut vines contain a large amount of soil, with many large, hard soil clumps, and the weight of the soil-cleaning components increases due to soil adhesion. This results in a significant increase in the load for processes such as high-frequency up-and-down swinging, soil squeezing and crushing, and centrifugal soil throwing. To achieve efficient soil removal under clay soil conditions, based on the results of previous experiments, the total input power of the designed composite soil cleaning device was designed to be 6 Kw.

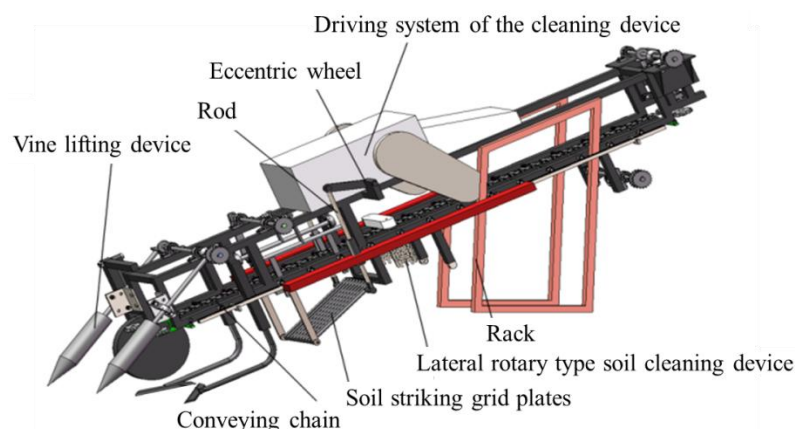


Figure 1. Overall configuration of the composite soil cleaning device.

2.2. Up-And-Down Flapping-Type Soil Cleaning Device

The up-and-down flapping-type soil cleaning device is composed of a power input module, an eccentric transmission module, a connecting rod transmission module, a soil-striking grid plate, and an auxiliary support module. The structure is shown in Figure 2. This device features an eccentric wheel–multi-link transmission system to achieve directional excitation.

The vibration transmission system is composed of an eccentric wheel, a transverse connecting rod, a steering connecting rod, a lifting connecting rod, a soil-striking grid plate, etc. During operation, power is transmitted through the eccentric wheel and the transverse and steering connecting rods to convert the planar rotation into the up-and-down reciprocating motion of the lifting connecting rod, causing the soil-striking grid plate to periodically oscillate around the lower hinge point of the adjusting connecting rod. The other side of the soil-striking grid plate was mainly used for cleaning the soil. By raising the adjusting connecting rod, the spacing between the soil-striking grid plate and the peanut pod can be dynamically adjusted to ensure smooth transportation of the peanut while maintaining the optimal excitation intensity.

The soil-striking grid plate is the core component of the device. It breaks down the soil in the peanut pod area by swinging up and down and removes the soil adhered to the pods through vibration. The overall structure of this plate is shown in Figure 3. It mainly consists of vertical grid bars welded above horizontal rods. The function of the vertical grid bars is to shake the pod clusters, while that of the horizontal rods is to connect and fix the vertical grid bars, forming a framework structure. At the same time, the ends of the horizontal rods are connected to the lifting and adjusting connecting rods, enabling the overall swinging motion. Through on-site measurement, the distribution range of the peanut pod clusters was found to be 150–250 mm. In order to ensure that the soil-striking grid plate could cover this distribution range during the transportation of peanut vines, the lateral width of the soil-striking grid plate was designed to be 400

mm. Furthermore, to ensure that the vertical grid bars make full contact with the pod cluster, the vertical grid bar spacing should be minimized as much as possible. Conversely, in order for the removed soil to pass through the gaps, the spacing of the vertical grid bars should also be maximized. To enhance the reliability of long-term shaking of pods and clay soil, the diameter of a single vertical grid bar and the spacing of the vertical grid bars need to be designed through testing.

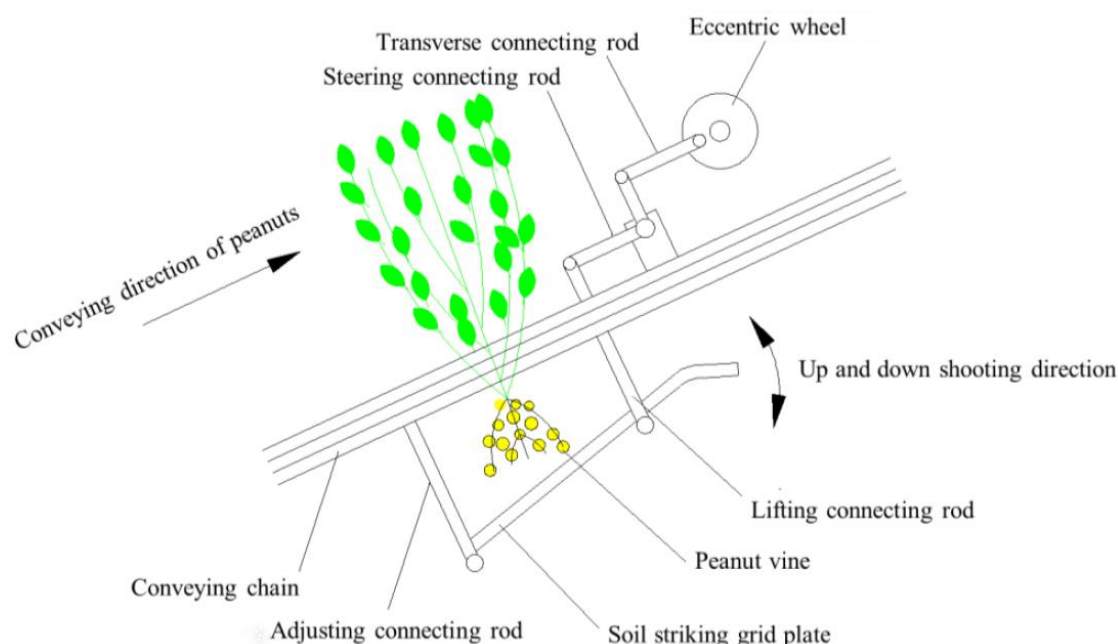


Figure 2. Schematic diagram of the up-and-down flapping-type soil cleaning device.

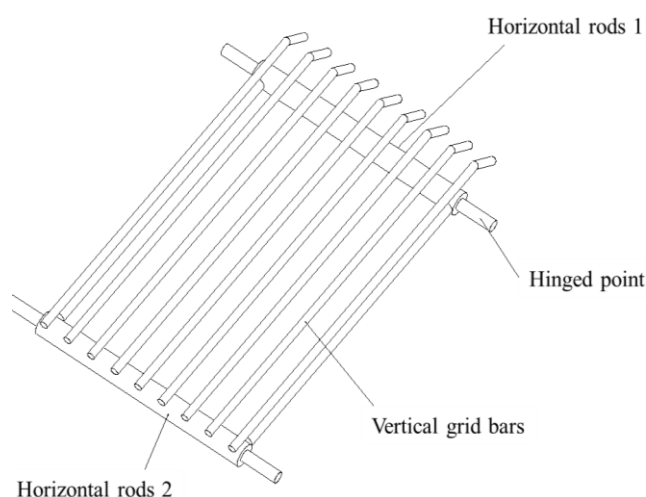


Figure 3. Schematic diagram of the soil-striking grid plate.

2.3. Lateral Rotary-Type Soil Cleaning Device

The main component of the lateral rotary-type soil cleaning device is the laterally configured squirrel-cage rotating wheels. There were two groups of such rollers: both were centered on the transportation path of the peanut pods, with one placed symmetrically on each side of the path. This device squeezes and breaks down the soil adhered to the pods through two squirrel-cage rotating wheels and removes the soil by centrifugal force.

The squirrel-cage rotating wheel consists of multiple circumferentially distributed striking poles. These poles were fixed in a framework structure by a circular ring; the overall structure is shown in Figure 4. The diameter of the squirrel-cage rotating wheels

and the distance between the two left and right rollers were designed based on the circumferential distribution range of the peanut pods. A striking pole was placed at fixed intervals along the circumference. This not only enabled the required striking frequency, but also avoided the poles being too dense, which would cause excessive damage to the peanut pods. Based on the height distribution range of the peanut pods, the height of the striking pole was set, which enabled effective striking of the soil and ensured that the cantilever structure did not deform during this process. The squirrel-cage rotating wheel was made of high-strength aluminum alloy, which has a low density and weight to reduce the rotational inertia and lower energy consumption. It has both good strength and corrosion resistance, and can be adapted to the complex working environment in the field.

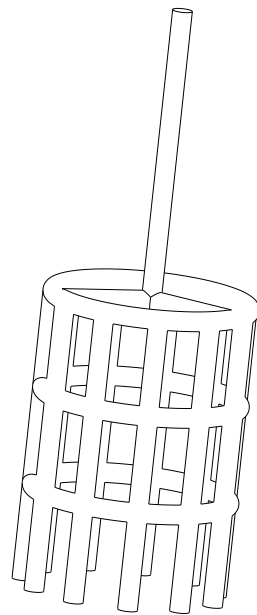


Figure 4. Schematic diagram of the squirrel-cage rotating wheel.

3. Finite Element Simulation Analysis of the Structural Mechanics of the Composite Soil Cleaning Device

All finite element analyses in this study were conducted within a single unified software environment, ANSYS Workbench (2020, ANSYS, Inc., Canonsburg, PA, USA), which features multiple solver modules. Different modules were employed according to the physical nature of each problem: the Static Structural module (implicit solver) was chosen for the quasi-static strength assessment under steady loads; the LS-DYNA module ((2020, ANSYS, Inc., Canonsburg, PA, USA)) was chosen for the transient impact analyses involving large deformation, high strain rates and complex contact; the Modal module was chosen for the extraction of natural frequencies and mode shapes; and the Harmonic Response module was chosen for the frequency-domain evaluation of resonant responses. Since these modules shared the same geometry, material database, meshing tools and post-processing environment within Workbench, the consistency of the finite element models was guaranteed.

The finite element meshing strategy was as follows: A uniform meshing strategy was adopted for all finite element analyses. All models were discretized with hexahedral-dominant meshes generated with quadratic hexahedral solid elements (SOLID186). For the soil-striking grid plate and the brackets, the geometries of which are regular prismatic structures, the sweep method was used; for the lateral rotary-type soil cleaning device, which has a relatively complex geometry, the multizone (partition) method was applied. The global element size was selected according to the dimensions of each component, and local refinement was applied at the contact regions, around the keyway and at the hinge points to

capture possible stress concentrations. The mesh quality was verified against the engineering acceptance criteria (skewness and orthogonal quality) before the solution. The mesh quality of all models satisfied the engineering acceptance criteria for hexahedral meshes. The modal (Section 3.2.2) and harmonic response (Section 3.2.3) analyses retained the same mesh as the explicit dynamics model to ensure model consistency. In the explicit dynamic analyses (LS-DYNA module), hourglass control was enabled with the corresponding element formulation to suppress non-physical zero-energy modes.

3.1. Finite Element Simulation Analysis of the Up-And-Down Flapping-Type Soil Cleaning Device

3.1.1. Static Analysis of the Soil-Striking Grid Plate

The Static Structural module of Workbench software was used to conduct static simulation of the structure of the soil-striking grid plate [26,27]. It was assumed that the soil-striking grid plate was made of No. 45 steel, and only the stress and strain within the yield limit were considered; therefore, the material property was set to linear elasticity. The material density was 7.85 g/cm^3 , the elastic modulus was 210 GPa, the Poisson's ratio was 0.31, and the yield limit was 355 MPa. The model was divided into a hexahedral-dominant mesh using the sweep method, and the steel frame of the model base was fixed in a plane. Peanuts are grown in ridges, with two rows planted on each ridge. During harvest, two rows of peanuts are harvested at a time. Under clay soil conditions, after excavation, a large number of soil clumps are adhered to the peanut pods. According to measurements, the total mass of two rows of peanut pods and soil in the area of the soil-striking grid plate was approximately 8–20 kg. Taking into account the dynamic amplification effect caused by the high-frequency oscillation (the eccentric wheel rotates at 300 r/min, equivalent to a maximum soil velocity of 700 mm/s; the cleaning frequency is 4.0 Hz), as well as the potential impact of hard soil clumps, a conservative dynamic amplification factor of 4 was adopted. As a result, the maximum design load was determined to be 800 N. This load was applied perpendicular to the ground. The loading model and mesh model are shown in Figure 5.

After the numerical simulation was completed, the stress distribution cloud diagram and strain field distribution characteristics of the soil-striking grid plate were obtained through post-processing analysis. The stress–strain cloud diagram analysis, shown in Figure 6, revealed that the maximum equivalent stress value in the model was 3.2 MPa, which is lower than the material's yield strength of 355 MPa. Under static working conditions, the structural stress level remains within the elastic deformation range, meeting the load-bearing capacity requirements of the design.

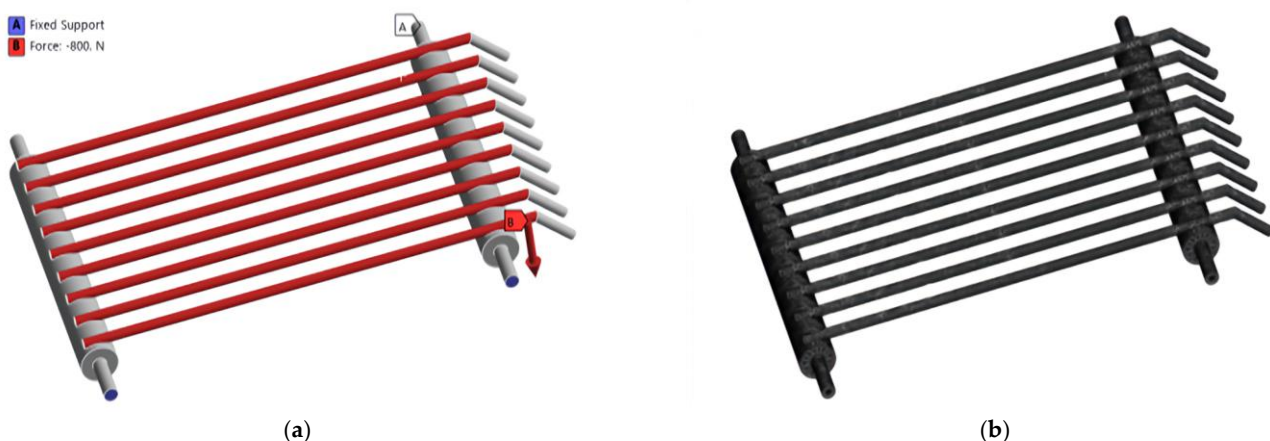


Figure 5. Finite element model of the soil-striking grid plate. (a) Finite element loading model. (b) Mesh model.

In the static analysis, the fixed constraint was applied to the plane of the steel frame of the model base, which corresponds to the mounting surface on the combine harvester frame, while the connections between the soil-striking grid plate and the lifting connecting rods were modeled as hinged joints with rotational degrees of freedom. It should be noted that, even if the fixed boundary partially over-constrains the real hinged connections and leads to a somewhat lower computed stress (3.2 MPa), the explicit dynamic analyses with frictional contacts (Sections 3.1.2 and 3.1.3) yielded maximum stresses of 48 MPa and 81.24 MPa, respectively, still far below the yield limit of 355 MPa. The structural safety conclusion is therefore not affected by this boundary treatment.

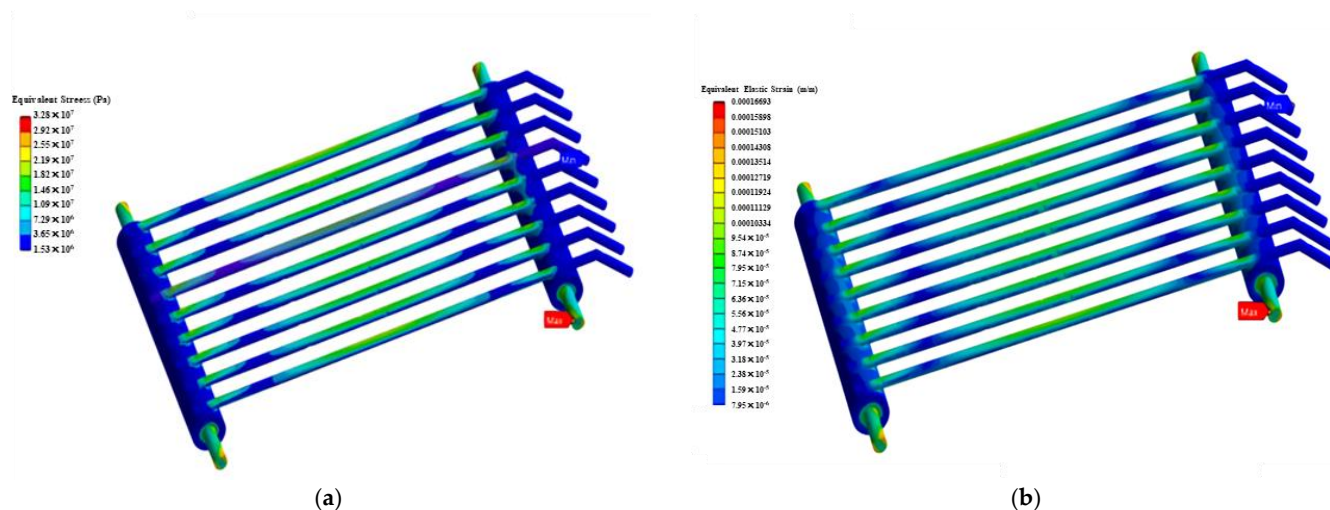


Figure 6. Stress–strain cloud diagram of the soil-striking grid plate. (a) Stress cloud diagram. (b) Strain cloud diagram.

3.1.2. Explicit Dynamics Analysis of the Soil-Striking Grid Plate

The LS-DYNA module of Workbench software was used to conduct simulation analysis on the soil-striking grid plate under the loaded impact. The established finite element model of the load impact is shown in Figure 7. Clay soil was added to the soil-striking grid plate and treated as a whole. An initial vertical downward velocity was applied to the soil to study the strength limit of the soil-striking grid plate under impulse.



Figure 7. Finite element model of load impact.

In the explicit dynamics analyses, the clay soil was modeled as a rigid body serving as the loading medium for the impact load. Since the aim of this study is to determine the structural strength of the cleaning components rather than the mechanical response of the soil itself, the clay soil was simplified as a rigid body, and only its density (1.7 g/cm³) was required for the material definition. The effect of impact on the structure was characterized by the initial downward velocity and the applied load imposed on the soil

body. The model was divided into a hexahedral-dominant mesh using the sweep method, and the model base plane was fixed. A load of 800 N perpendicular to the ground was applied to the upper surface of the soil. The eccentricity of the eccentric wheel was 70 mm, and the maximum stroke of the soil-striking grid plate was 140 mm. The eccentric wheel rotates at 300 r/min, taking 0.2 s to complete one cycle of motion. This is equivalent to a maximum soil velocity of 700 mm/s; this velocity was assigned to the soil. The contact between the clay soil and the soil-striking grid plate was modeled using the body interaction setting of the ANSYS Workbench LS-DYNA module, which was solved as the penalty-based automatic general contact (AUTOMATIC_GENERAL) in LS-DYNA. The penalty value used for the contact algorithm was default. All potential contact pairs were automatically detected between the clay soil and the soil-striking grid plate. The response time was 0.001 s. The loading model and mesh model are shown in Figure 8.

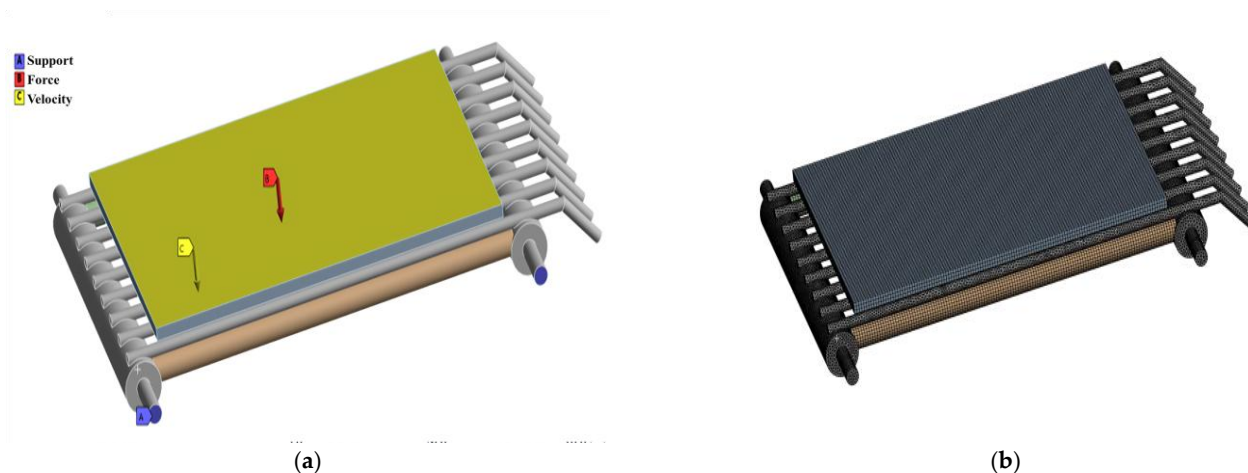


Figure 8. Finite element loading model of load impact. (a) Loading model. (b) Mesh model.

After the simulation was completed, the stress field distribution characteristics of the model were extracted through the post-processing program, as shown in Figure 9. The numerical simulation results showed that the maximum equivalent stress of the model under dynamic loading was 48 MPa. There was a significant safety difference between this stress and the material's yield limit of 355 MPa.

Through the stress–strength comparison, it was shown that under impulse load excitation, the dynamic response of the structure was always elastic deformation, verifying its load-bearing stability.

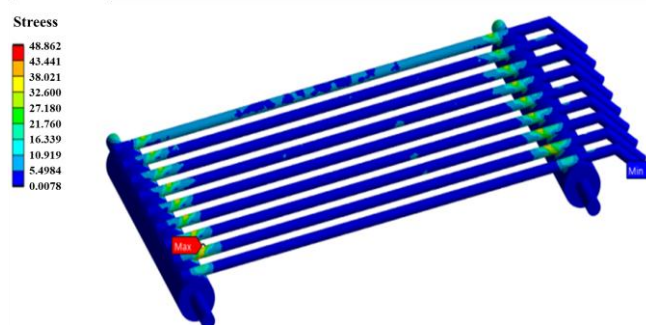


Figure 9. Load impact stress cloud diagram.

3.1.3. Explicit Dynamic Analysis at the Connection Points of the Brackets

The LS-DYNA module of Workbench software was used for the simulation of the brackets, and the stress conditions at the connection points were analyzed. The finite element model of the brackets is shown in Figure 10.

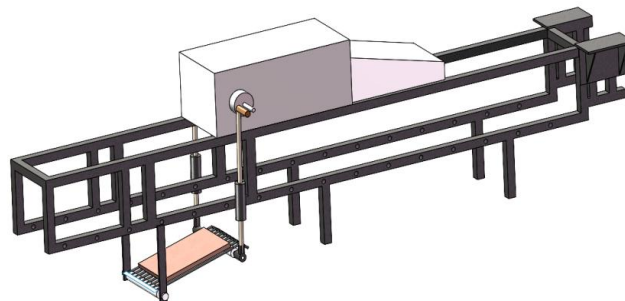


Figure 10. Finite element model of the brackets.

The density of clay soil was set as 1.7 g/cm³. The bracket material was stainless steel, with a density of 7.80 g/cm³, an elastic modulus of 200 GPa, a yield limit of 360 MPa, and a Poisson's ratio of 0.3. The model was divided into a hexahedral-dominant mesh using the sweep method, and the base of the brackets was fixed. A load of 800 N perpendicular to the ground was added to the upper surface of the soil, and its circumferential displacement was constrained. The eccentricity of the eccentric wheel was set to 70 mm, and the maximum stroke of the soil-striking grid plate was 140 mm. The eccentric wheel rotates at 300 r/min, taking 0.2 s to complete one cycle of motion. This is equivalent to a maximum soil velocity of 700 mm/s; this velocity was assigned to the soil. The contacts between the soil-striking grid plate and the front and rear brackets were defined as frictional contacts, and were solved as a penalty-based AUTOMATIC_SINGLE_SURFACE with a Coulomb friction coefficient of 0.15. The interaction between the clay soil and the soil-striking grid plate was modeled with a body interaction, solved as a penalty-based automatic general contact (AUTOMATIC_GENERAL). The response time was 0.001 s. The loading model and mesh model are shown in Figure 11.

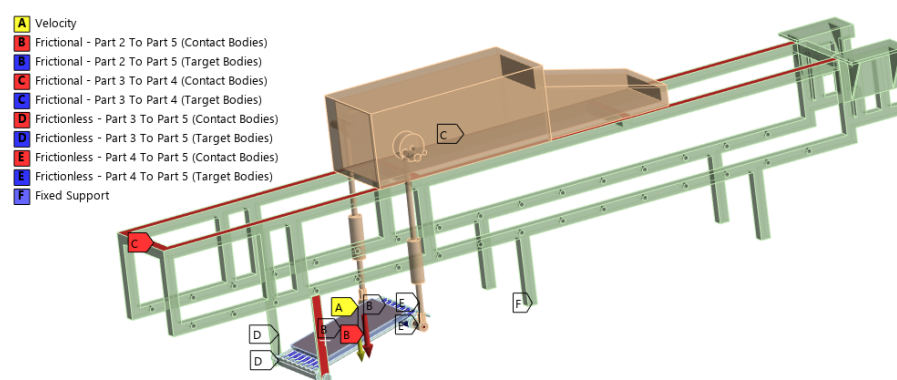


Figure 11. Finite element loading model of the brackets.

The overall stress cloud diagram of the brackets is shown in Figure 12. The overall peak stress of the brackets was 81.24 MPa. The stress cloud diagram at the brackets' connection point is shown in Figure 13; the stress peaks at the hinge joints of the front and rear brackets were 0.36 MPa and 5.35 MPa, respectively. The above equivalent stress values were significantly safer compared to the allowable stress limit of the material: 355 MPa. A comparison of the measured stress data in all directions with the standard value

of material strength indicated that the structural system fully met the strength requirements under the actual working conditions in the three-dimensional force state.

The bracket was made from 304 steel and had a yield strength of 360 MPa. The reliability assessment in this study was based on the static yield strength criterion. The maximum stress of 81.24 MPa accounted for only 22.6% of the yield strength. Even considering the lower allowable stress of the welded joints (typically 60% of the parent material) in the connection zones, the strength requirement was still satisfied with a sufficient margin. The stresses at the bolt joints were extremely low, and the effect of bolt preload did not affect the strength conclusions. Since the maximum stress was far below the yield strength, the structure remains in the linear elastic range, and no local plasticity occurs. Fatigue life assessment was beyond the scope of this study and was identified as future work. Nevertheless, the low stress levels (well below the typical fatigue limit range of 304 stainless steel), together with the field test results (continuous operation of more than 3 mu without deformation or fracture), provided strong engineering evidence of the bracket's reliability.

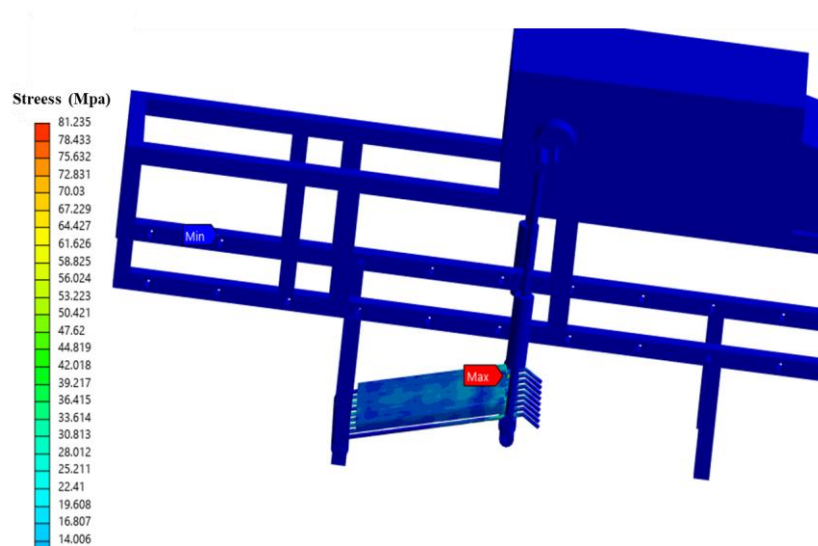


Figure 12. Overall stress cloud diagram of the brackets.

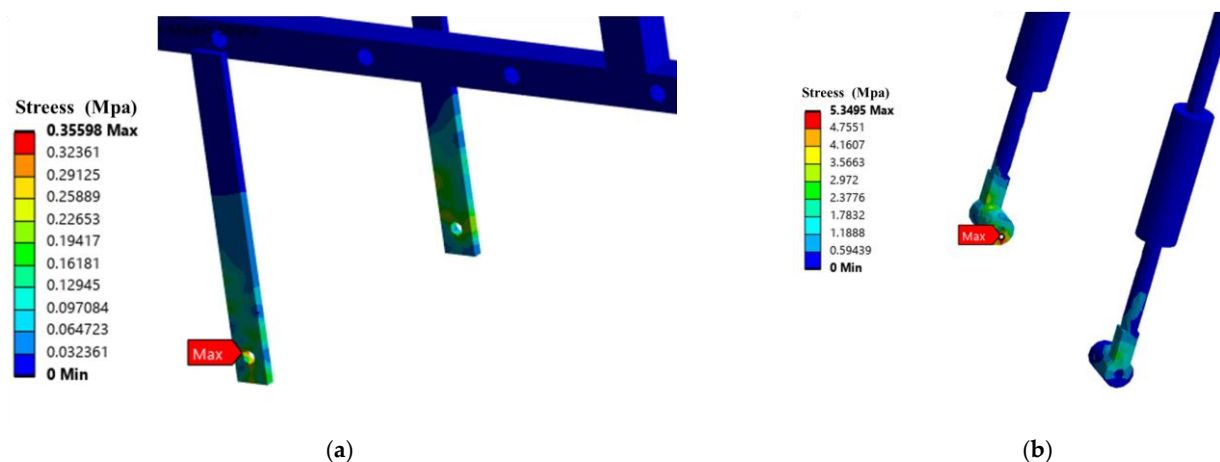


Figure 13. Stress cloud diagram at the connection point of the brackets. (a) The connection point of bracket A. (b) The connection point of bracket B.

3.2. Finite Element Simulation Analysis of the Lateral Rotary-Type Soil Cleaning Device

3.2.1. Dynamic Simulation of the Lateral Rotary-Type Soil Cleaning Device

The strength of the component under dynamic loads was analyzed using the LS-DYNA module of the Workbench software. The model consisted of an axle sleeve, a key, and a squirrel-cage rotating wheel.

The material of the axle sleeve was 45 steel, with a density of 7.85 g/cm³, an elastic modulus of 210 GPa, a Poisson's ratio of 0.31, and a yield limit of 355 MPa, while the material of the key was 40Cr alloy steel, with a density of 7.85 g/cm³, an elastic modulus of 200 GPa, a Poisson's ratio of 0.28, and a yield limit of 785 MPa and the material of the squirrel-cage rotating wheel was 7075-type aluminum alloy, with a density of 2.81 g/cm³, an elastic modulus of 73 GPa, a Poisson's ratio of 0.33, and a yield limit of 455 MPa. The model was divided into hexahedral-dominant grids using the partitioning method; the grid model is shown in Figure 14.



Figure 14. Mesh model of the lateral rotary-type soil cleaning device.

The rotational speed of 600 r/min applied in the simulation was the maximum rotational speed of the drive system's axle sleeve. This maximum speed was deliberately selected to cover the most unfavorable inertial loading condition. The contact between each part was set as frictional contact; these contacts were solved as a penalty-based AUTOMATIC_SINGLE_SURFACE with a Coulomb friction coefficient of 0.15. The calculation time was 0.001 s; the loading model is shown in Figure 15.

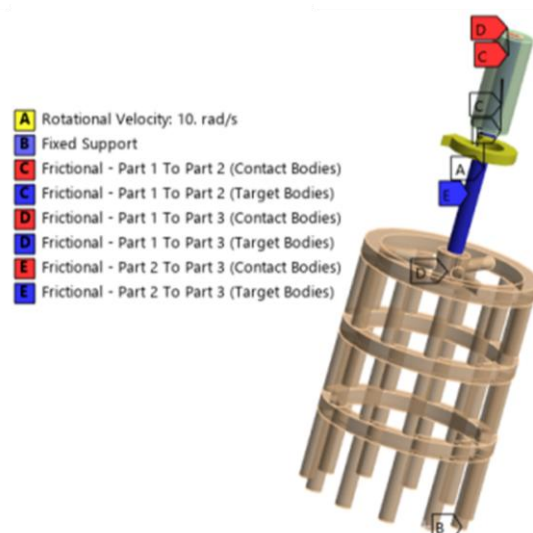


Figure 15. Finite element loading model of the lateral rotary-type soil cleaning device.

The stress field distribution of the lateral rotary-type soil cleaning device is shown in Figure 16 and demonstrates the evolution of the stress gradient of the overall structure. The local stress fields of key components were visualized using multi-level visualization analysis, as shown in Figure 17. Through this multi-scale display approach, not only could the overall stress gradient trend be observed, but also the stress concentration at key components could be accurately identified. The stress distribution of the model under the ultimate working condition is shown in the stress cloud diagrams of Figures 16 and 17. From these diagrams, the following characteristics were observed: The peak stress of the overall lateral rotary-type soil cleaning device structure was 4.43 MPa, where the maximum stresses of the key, axle sleeve and squirrel-cage rotating wheel were 0.14 MPa, 0.36 MPa and 4.43 MPa, respectively. By comparing the yield limits of the materials, it can be seen that there were order-of-magnitude differences between the stress values of each component and their strength thresholds. The stress values of each component were far below their individual yield limits. The stress level of the overall structure under extreme loads remained within the elastic deformation range, and the structure was reliable.

It should be noted that this analysis mainly evaluated the inertial strength of the rotating wheel at maximum speed; the external impact loads from the soil, pods and soil clods were not directly evaluated in this model. Nevertheless, the structural safety conclusion is not affected for the following reasons: the maximum stress of 4.43 MPa accounts for only about 1% of the yield strength of 7075 aluminum alloy (455 MPa); the explicit dynamic analyses considering the clay soil impact loads (Sections 3.1.2 and 3.1.3) yielded stress levels of 48 MPa and 81.24 MPa, respectively, indicating that the stress magnitude under soil impact is still far below the yield limits; and the field tests (continuous operation without deformation or fracture) directly verified the reliability of the rotating wheel.

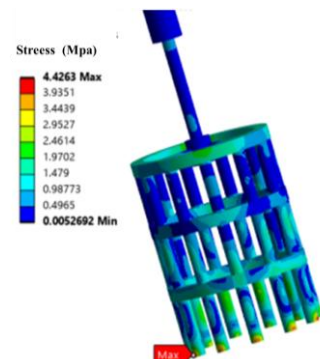


Figure 16. Overall stress cloud diagram of the lateral rotary-type soil cleaning device.

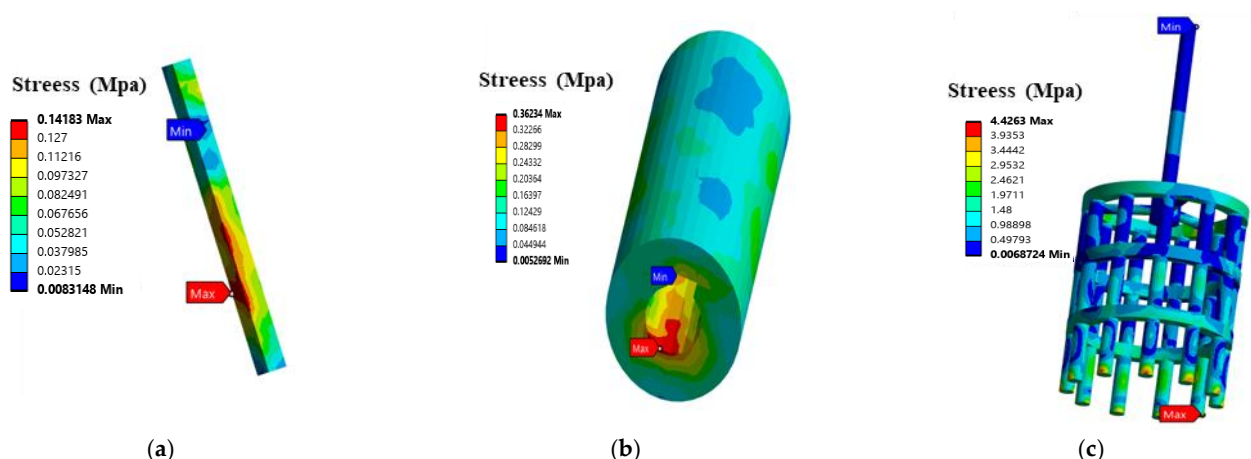


Figure 17. Stress cloud diagram of key components of the lateral rotary-type soil cleaning device. (a) Key. (b) Axle sleeve. (c) Squirrel-cage rotating wheel.

3.2.2. Modal Analysis of the Squirrel-Cage Rotating Wheel

Due to the vibration of the squirrel-cage rotating wheel, it might suffer fatigue failure under the combined action of the natural frequency and alternating load. Therefore, it was subjected to modal analysis using the Modal module of Workbench software [28,29]. The material of the squirrel-cage rotating wheel was 7075-type aluminum alloy, with a density of 2.81 g/cm³, an elastic modulus of 73 GPa, a Poisson's ratio of 0.33, and a yield limit of 455 MPa. The modes up to the 25th order were calculated. According to industry consensus, vibrations of equipment are centered on natural frequencies below 2000 Hz.

The modal analysis was performed under the free–free condition (no fixed constraints) to determine the intrinsic natural frequencies of the squirrel-cage rotating wheel independent of the support configuration. As shown in Table 1, the first six modes exhibit near-zero frequencies (0 to 3.2×10^{-3} Hz), which correspond to the six rigid-body degrees of freedom of the free–free modal analysis; these rigid-body modes represent rigid translations and rotations of the whole structure rather than elastic deformation and therefore do not pose any vibration risk. The modal analysis results indicate that the third- and ninth-order modes exhibit distinct vibration pattern characteristics. The former reflect the axial oscillation of the transmission shaft, while the latter reflect the oscillation response of the outer end face of the wheel. Both of these vibration responses might pose a risk of structural resonance. The vibration mode characteristics are shown in Figure 18. The amplitude values corresponding to both modes approach the safety threshold, and they should be paid special attention during structural optimization.

Table 1. Natural frequencies ranging from 0 to 2000 Hz.

Mode Order	Natural Frequency
1	0
2	0
3	1.0867×10^{-3}
4	2.1825×10^{-3}
5	2.7108×10^{-3}
6	3.1952×10^{-3}
7	322.72
8	361.96
9	1350.5
10	1350.6
11	1731.5
12	1881
13	1917.8

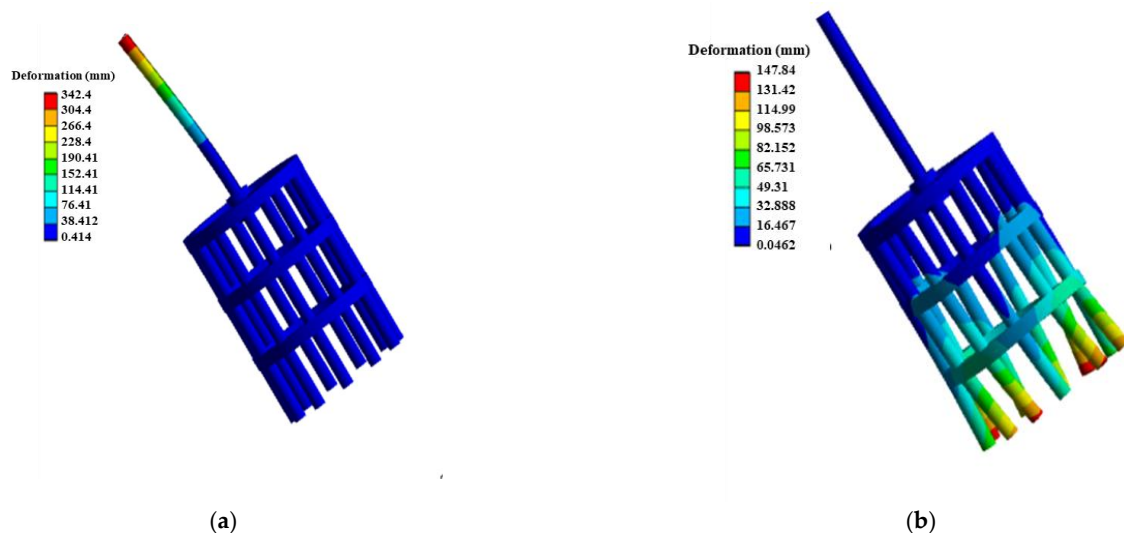


Figure 18. Vibration mode characteristics of the squirrel-cage rotating wheel. (a) Third-order mode. (b) Ninth-order mode.

3.2.3. Harmonic Response Analysis of the Natural Frequency of the Squirrel-Cage Rotating Wheel

Due to the risk of potential structural resonance at the two natural frequencies mentioned above, a harmonic response analysis after loading was conducted. The harmonic response analyses listed in Table 1 were conducted at the seventh (322.72 Hz) and ninth (1350.5 Hz) natural frequencies, the first two flexible modes with significant deformation. The near-zero rigid-body modes (modes 1–6) were excluded from the harmonic response analysis because these modes possess no elastic stiffness and produce no meaningful stress response; the frequency sweep ranges (300–400 Hz and 1300–1400 Hz) also exclude the near-zero frequency region.

(1) Analysis at the natural frequency of 322.72 Hz

The calculation results from the above were imported into the Harmonic Response module of the Workbench software. Then, the superposition method was replaced by the global method, and the damping coefficient was corrected to 0.075. When the unremoved soil clumps and other materials are separated from the up-and-down flapping-type soil cleaning device, they enter the lateral rotary-type soil cleaning device. At this moment, all materials are present in the squirrel-cage rotating wheel, while none are acting upon the up-and-down flapping-type soil cleaning device. The maximum rotational speed of the wheel is 600 r/min (as shown in Section 3.2.1). Assuming that all the materials were acting on the wheel, using the torque calculation formula $T = 9550 P/N$, the torque of the squirrel-cage rotating wheel at this time was calculated to be 95.5 N·m. Taking into account safety redundancy, the torque was rounded, and a torque load of 100 N·m was applied to the shaft to calculate the corresponding spectrum at frequencies ranging from 300 to 400 Hz. Figure 19 shows the stress curve of the squirrel-cage rotating wheel at these frequencies, while Figure 20 presents the phase curve corresponding to the maximum stress.

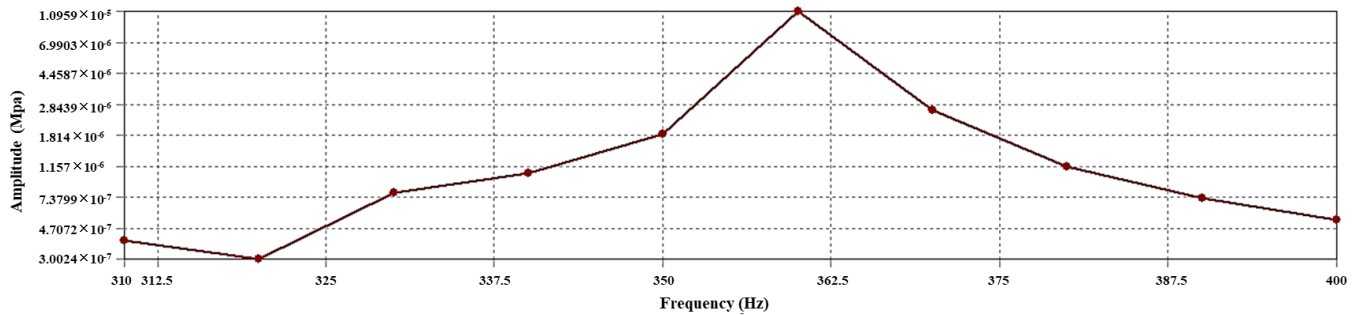


Figure 19. Stress curve of the squirrel-cage rotating wheel at frequencies ranging from 300 to 400 Hz.

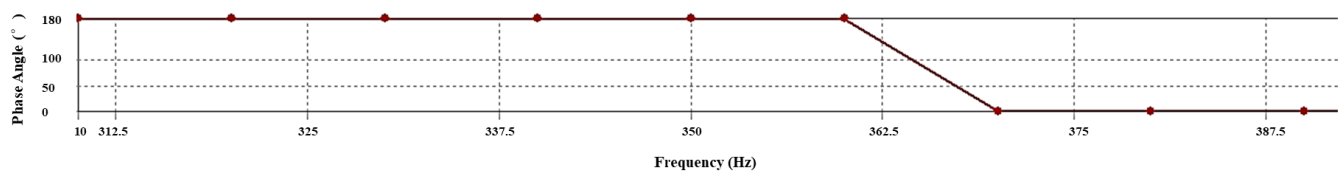


Figure 20. Phase curve of the squirrel-cage rotating wheel at 360 Hz.

The dynamic analysis data indicated that when the excitation frequency reached 360 Hz and the phase parameter was 180°, the system exhibited the maximum stress response characteristics. The stress cloud diagram under this specific frequency–phase combination is shown in Figure 21, showing the gradient distribution and the peak region of the stress field under the critical parameters. The maximum equivalent stress of 0.12 MPa is significantly lower than the material's yield strength of 455 MPa. This structure has sufficient strength reserve under the rated working conditions, effectively avoiding failure due to fracture.

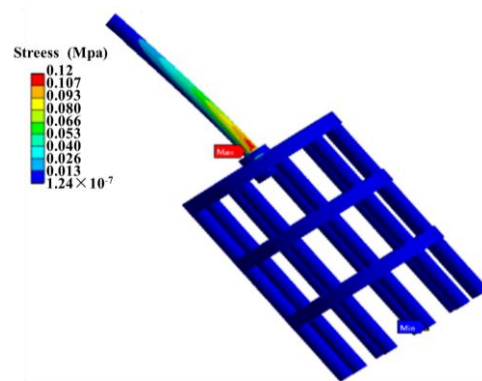


Figure 21. Stress cloud diagram of the squirrel-cage rotating wheel at 360 Hz.

(2) Analysis at the natural frequency of 1350.5 Hz

The calculated results from the above were again imported into the Harmonic Response module of the Workbench software, and the superposition method was replaced by the global method. A constant damping ratio of 0.075 was applied in the harmonic response analysis. This value lies within the typical range of 0.01–0.1 (percentage of critical damping) suggested for engineering structures in the ANSYS documentation [30]. It should be noted that for metallic materials, the damping ratio is generally below 0.05 [31]. However, the squirrel-cage rotating wheel is an assembled structure containing a keyed connection, friction contacts and bearings. The interface friction of jointed connections can contribute up to about 90% of the total damping of assembled structures [32].

The equivalent damping ratio of assembled structures was significantly higher than the value of pure metallic materials. The value of 0.075 was therefore adopted as a reasonable estimate at the upper end of the typical range for such an assembled structure. A torque load of 100 N·m was applied to the shaft and the corresponding spectrum at frequencies ranging from 1300 to 1400 Hz was calculated. Figure 22 shows the stress curve of the squirrel-cage rotating wheel at these frequencies, while Figure 23 presents the phase curve corresponding to the maximum stress.

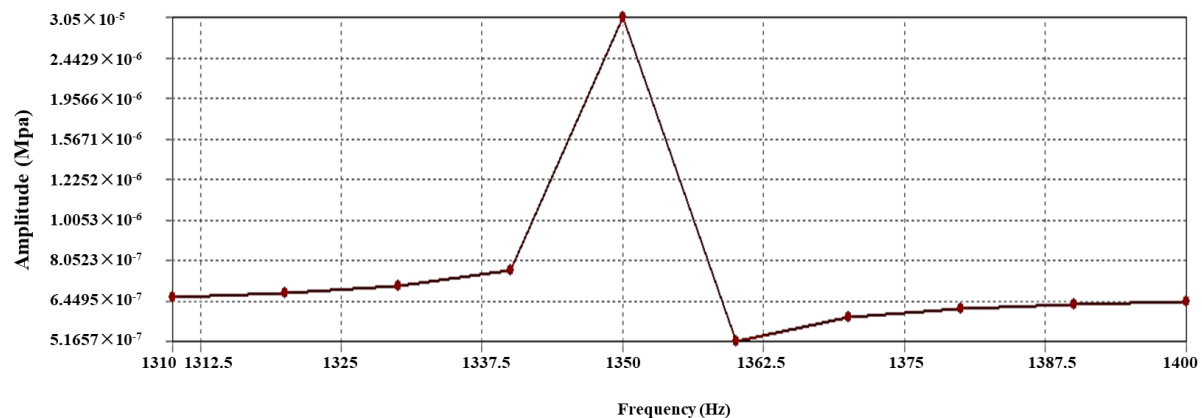


Figure 22. Stress peak curve of the squirrel-cage rotating wheel at frequencies ranging from 1300 to 1400 Hz.

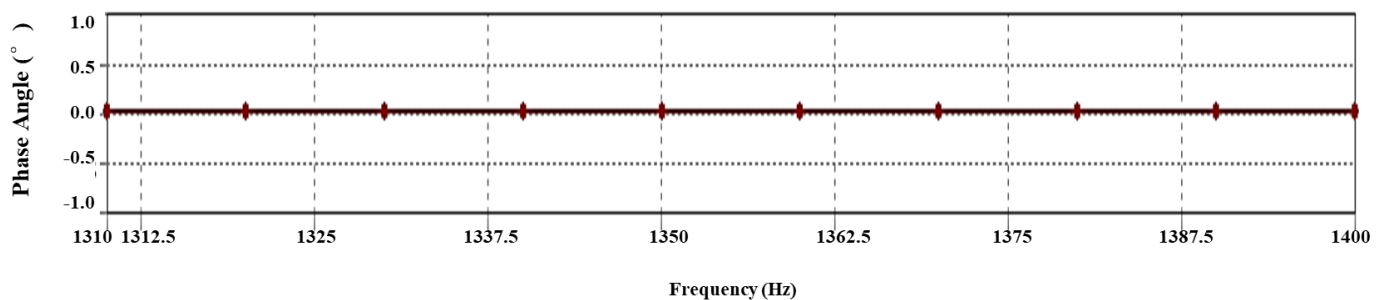


Figure 23. Phase peak curve of the squirrel-cage rotating wheel at 1350 Hz.

The dynamic analysis data indicate that when the excitation frequency reached 1350 Hz and the phase parameter was 0°, the system exhibited the maximum stress response characteristics. The stress cloud diagram under this specific frequency-phase combination is shown in Figure 24, showing the gradient distribution and the peak region of the stress field under the critical parameters. The maximum equivalent stress of 0.44 MPa was significantly lower than the material's yield strength of 455 MPa. Therefore, this structure had sufficient strength reserve under the rated working conditions, effectively avoiding failure due to fracture.

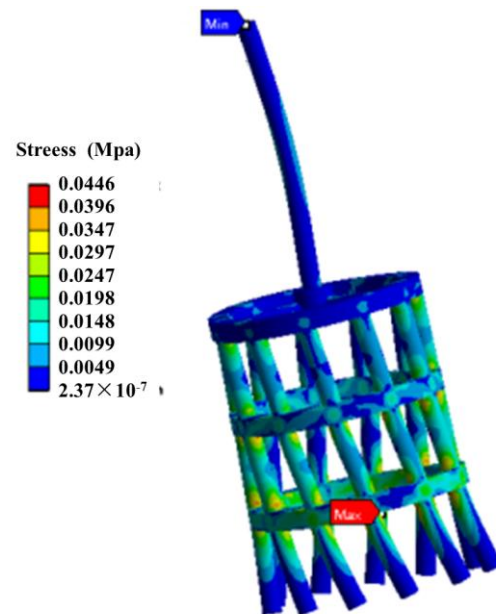


Figure 24. Stress cloud diagram of the squirrel-cage rotating wheel at 1350 Hz.

4. Field Test of the Composite Soil Cleaning Device

In order to evaluate the reliability and cleaning efficiency of the composite soil cleaning device in the field, first, based on the theoretical design and simulation analysis results above, a prototype of the device was fabricated. Then, using the operation parameter combinations determined in the previous pre-tests as the basis, a large-scale reliability assessment test was carried out to provide the necessary basis for improving the design of the composite soil cleaning device.

In the previous pre-tests, the composite soil cleaning device was configured as a semi-feed peanut combine harvester. We selected four key operational parameters of the device as experimental factors: the cleaning frequency, the cleaning angle, the conveying speed, and the rotation speed of the rotating wheel. The cleaning frequency refers to the vibration frequency of the up-and-down flapping-type soil cleaning device, while the cleaning angle refers to the angle between the soil-striking grid plate and the ground when it is at its lowest position, and the conveying speed is the conveying speed of the clamping device. Finally, the rotation speed is the rotation speed of the squirrel-cage rotating wheel. A single-factor experiment was conducted using the soil cleaning rate and the total loss rate as test indicators, and the data were analyzed using the fuzzy comprehensive evaluation method. For each experimental factor, three levels were selected. The soil cleaning rate and the total loss rate of each level of the factor were measured. Since the soil cleaning rate is a positively skewed indicator (the higher the better) and the total loss rate is a negatively skewed indicator (the lower the better), it was necessary to convert them into indicator membership values. The weight values for the two indicators were both set at 0.5. Finally, a comprehensive score was given for the two indicators; the specific method for comprehensive scoring was taken from reference [33]. When the cleaning frequency was 4.0 Hz, the cleaning angle was 23°, the conveying speed was 1.0 m/s, and the rotation speed of the rotating wheel was 160 r/min, the overall score was the highest.

This section detailed large-scale reliability assessments of the composite soil cleaning device, conducted in Linyi City, Shandong Province, where the soil type is clay soil. Based on the results of the previous parameter optimization tests, the key parameters of the composite soil cleaning device were set as follows: the cleaning frequency was 4.0 Hz, the cleaning angle was 23°, the conveying speed was 1.0 m/s, and the rotation speed

of the rotating wheel was 160 r/min. Under the above-mentioned parameters, the changes in the structural strength of the composite soil cleaning device were studied. Meanwhile, operation indicators such as the soil cleaning rate and total loss rate were measured. The test process is shown in Figure 25. (The test method refers to the agricultural industry standard of the People's Republic of China—"Peanut Harvester Operating Quality (NY/T502-2016)".)



Figure 25. Field test pictures.

The aim of field testing the composite soil cleaning device was to comprehensively evaluate its performance under actual conditions. The test results show that under the operating conditions of clay soil, when harvesting continuously for more than 3 acres, none of the hinge points, fixed points, welding points, and possible stress concentration points of the composite soil cleaning device, as well as component profiles, exhibited any signs of strength failure such as deformation or fracture. The soil cleaning rate of the device was over 90%, and its total loss rate was less than 5%. These indicators met the agricultural industry standard of the People's Republic of China—"Peanut Harvester Operating Quality (NY/T502-2016)". Therefore, the composite soil cleaning device performed well and met the strength requirements for large-scale harvesting operations in clay soil conditions, as well as the quality requirements for soil cleaning.

5. Discussion

5.1. Sustainability Implications of This Design

In addition to its structural reliability and cleaning performance verified above, the proposed composite soil cleaning device offers distinct sustainability benefits for peanut production in hilly and mountainous areas with clay soil in terms of energy consumption, soil conservation and the economic viability for smallholder farms, which are discussed below.

- (1) **Energy consumption.** After the digging and soil cleaning stages, the peanut combine harvester also carries out picking, screening and air sorting. The composite soil cleaning device effectively removes the soil adhered to the peanut vines before these stages. Consequently, the operational load of the picking, screening and air sorting components is substantially reduced, which improves their working efficiency and quality and reduces their power consumption. In conventional harvesters, a large amount of soil can easily accumulate in the working components, which increases their weight and operational load and degrades both efficiency and quality; use of the composite soil cleaning device largely avoids this adverse effect. In addition, by effectively removing the soil and impurities from the harvested material, the device reduces the labor intensity of post-harvest manual soil removal, decreases the water

consumption of de-soiling through washing, and lowers the energy consumption of secondary mechanical de-soiling after harvesting. Since the composite cleaning device improves the soil cleaning rate, the operating time of the combine harvester per unit area is shortened, which also contributes to reducing its overall energy consumption. It should be noted that quantitative energy measurements were not carried out in this study; the direct measurement of the power consumption of the cleaning components under field conditions is planned as future work.

- (2) Soil conservation. The soil removed by the composite soil cleaning device falls directly back onto the field, reducing the soil content within the harvested material and reducing soil loss during harvesting, which is of particular importance in hilly regions with thin topsoil. Furthermore, since the composite cleaning device is integrated into the peanut combine harvester, digging, soil cleaning, picking, screening and air sorting are accomplished in a single pass, effectively reducing the number of times single-function implements are passed over the field and potentially alleviating soil compaction caused by repeated operations.
- (3) Economic viability for smallholder farms. The mechanized harvesting rate of peanuts in the hilly and mountainous regions of China is currently less than 2%, mainly due to the lack of suitable machinery and the shortage of rural labor in these areas. By presenting a reliable and efficient cleaning device that can be integrated into small-scale peanut harvesters, this study supports the mechanization of peanut harvesting in hilly regions, which could reduce labor and labor costs and increase the marketable yield through improved cleaning and reduced loss rates. The simple structure and low power requirements of the device also imply that the acquisition and operating costs will be affordable for smallholders with limited investment capacity.

5.2. Analysis of Study Reliability

The structural safety conclusions of this study are not affected by the finite element mesh. The purpose of the finite element analyses in this study is to verify the structural safety of the cleaning device, rather than to provide high-accuracy absolute stress values.

The computed maximum stresses (3.2 MPa static; 48 MPa and 81.24 MPa explicit dynamic) are far below the yield strengths (355–360 MPa), with a minimum safety factor of approximately 4.4 (81.24 MPa vs. 360 MPa). In the present case, even a mesh-induced variation of 20% would only reduce the safety factor from 4.4 to approximately 3.5, which remains well above the common engineering threshold of 2–3 adopted in agricultural machinery design [34].

The reliability of the design is corroborated by multiple independent lines of evidence. Beyond the finite element analyses, three independent facts support the same conclusion: (i) the loading assumptions were deliberately conservative (e.g., the equivalent vertical load of 800 N derived using a dynamic amplification factor of 4, and the maximum rotational speed of 600 r/min used in the rotating wheel analysis); (ii) the stress levels were consistent across different analysis types (static, explicit dynamic and harmonic response analyses); and (iii) the results of field tests were supportive under real clay soil conditions (continuous operation without any deformation or failure of the components). The convergence of these independent lines of evidence provides stronger assurance of the structural reliability than a single mesh convergence check would.

As emphasized in the NAFEMS guidance [35], the purpose of establishing mesh convergence is to confirm that the computed quantity of interest is stable with respect to the mesh; here, the above analysis shows that the strength margin itself provides this robustness, and the conclusions are mesh-insensitive by construction.

The convergence of these independent lines of evidence provides robust assurance of the structural reliability of the designed cleaning device. The factor-of-safety criterion is

widely adopted as the primary strength verification in finite element analyses of agricultural machinery, and a safety factor of 2–3 is commonly considered sufficient in such designs [34].

A systematic mesh sensitivity analysis, if considered necessary for the completeness of the verification, will be reported in a subsequent study.

5.3. Corrosion/Wear Mitigation Strategy

One more point needs to be clarified. The connection between the 45-steel axle sleeve and the 7075-aluminum alloy squirrel-cage wheel involves contact between different metals, which, under repeated exposure to wet clay soil, could in principle lead to galvanic corrosion (since aluminum is more anodic than steel) and wear at the keyed connection caused by abrasion from soil particles. The following mitigation measures were adopted in the design: (I) the aluminum components were anodized to form a hard, corrosion-resistant oxide film on the surface; (II) anti-corrosion grease was applied at the assembly interfaces to isolate the galvanic pair and reduce friction; and (III) the connection was designed with proper fit tolerances and sealed against soil ingress to limit abrasive wear. A periodic maintenance schedule (regreasing, cleaning and inspection) is also recommended for field operation. In addition, the field tests confirmed that no corrosion- or wear-induced failure occurred at the connection during continuous operation. Nevertheless, the long-term corrosion resistance requires further evaluation through accelerated corrosion tests (e.g., salt spray testing), which will be included in our future work.

6. Conclusions

- (1) To meet soil cleaning requirements under clay soil conditions, a composite “up-and-down flapping–lateral rotation” soil cleaning device was proposed, and key components were designed.
- (2) Through means such as finite element static analysis, explicit dynamic simulation and modal analyses, the rationality and reliability of the structural design of the two components of the soil cleaning device, namely the up-and-down flapping-type and the lateral rotation-type devices, were verified. The maximum equivalent static and dynamic stresses of the soil-striking grid plate were 3.2 MPa and 48 MPa, respectively, far lower than the yield strength of 45 steel (355 MPa). Even under extreme impact loads from hard impurities such as stones or roots during field operations, this plate can still maintain elastic deformation and avoid permanent failure due to plastic deformation. The maximum stress at the connection points of the brackets was 81.24 MPa, which met the requirement imposed by the yield strength of 304 stainless steel (360 MPa). The peak stress of the entire lateral rotary-type soil cleaning device was 4.43 MPa; among important values were the maximum stress of the key, axle sleeve and squirrel-cage rotating wheel, at 0.14 MPa, 0.36 MPa and 4.43 MPa, respectively. These values were far lower than their individual yield limits of 785 MPa, 355 MPa and 455 MPa, respectively. This indicates that the dynamic characteristics meet the requirements imposed by extreme working conditions in the field. Modal and harmonic response analysis showed that the device’s natural frequency lies outside the vibration range present under working conditions, and its anti-vibration performance was excellent.
- (3) Through large-scale field tests and experiments, it was found that the hinge points, fixed points, welding points, and possible stress concentration points of the composite soil cleaning device, as well as the component profiles, did not exhibit any signs of strength failure such as deformation or fracture. The soil cleaning rate of the device was over 90%, and its total loss rate was less than 5%. The composite soil cleaning device, configured for use in semi-feed peanut combine harvesters, meets the structural strength and harvest quality requirements for large-scale operations in clay soil.

- (4) The composite soil cleaning device benefits the sustainability of peanut production in hilly and mountainous areas with clay soil in terms of energy consumption, soil conservation, and economic viability for smallholder farms. Overall, the device provides a reliable, low-cost, and energy-efficient technical solution that supports the transition toward sustainable peanut production in hilly and mountainous regions.

Author Contributions: Conceptualization, F.G., and F.W.; methodology, Y.J., Z.Y. and F.W.; software, H.X.; validation, Z.H.; formal analysis, F.D.; investigation, H.X. and Z.H.; resources, F.W.; data curation, F.D.; writing—original draft preparation, Y.J., Z.Y. and F.D.; writing—review and editing, Y.J., Z.Y., F.G. and Z.L.; visualization, H.X.; supervision and project administration, F.G., and Z.L.; funding acquisition, Z.H. All authors have read and agreed to the published version of the manuscript.

Funding: The study was supported by the pilot project for the integrated development, manufacturing, promotion and application of agricultural machinery equipment in Shandong Province (SDNJYTH-202505), the pilot project for the integrated development, manufacturing, promotion and application of agricultural machinery equipment in Jiangxi Province (YCTY202502), and the Agricultural science and technology innovation program of the Chinese Academy of Agricultural Sciences.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: The data presented in this study are available on request from the corresponding author due to privacy.

Conflicts of Interest: Author Fujie Ding is employed by Linshu Dongtai Machinery Co., Ltd. The author declares that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

References

1. Liao, B.S. A Review on Progress and Prospects of Peanut Industry in China. *Chin. J. Oil Crop Sci.* **2020**, *42*, 161–166.
2. Wang, Y.; Yang, J.; Peng, K.; Hua, B. Current situation and prospects of peanut industry in China. *Agric. Outlook* **2024**, *20*, 86–91.
3. China Statistical Yearbook 2024. National Bureau of Statistics of China. 2025. Available online: <https://www.stats.gov.cn/sj/ndsj/2024/indexch.htm> (accessed on 26 March 2025).
4. Chen, Z.Y.; Gao, L.X.; Chen, C.; Butts, C.L. Analysis on technology status and development of peanut harvest mechanization of China and the United States. *Trans. Chin. Soc. Agric. Mach.* **2017**, *48*, 1–21.
5. Gao, L.X.; Chen, Z.Y.; Chen, C.; Butts, C.L. Development course of peanut harvest mechanization technology of the United States and enlightenment to China. *Trans. Chin. Soc. Agric. Eng.* **2017**, *33*, 1–9.
6. Fang, B. Design of Small Peanut Harvester and Simulation of Key Components in Low Hill and Gentle Slope Area. Master's Thesis, Anhui Agricultural University, Hefei, China, 2024.
7. Chen, Y.Q.; Xiong, S.; Chen, M.L.; Luo, Q.J. Research status and prospects of peanut harvesting mechanization. *Mod. Agric. Equip.* **2023**, *44*, 24–30.
8. Zhan, X.M.; Cao, Z.H.; Zhou, Y.H.; Liu, R.S. Study on the integration of agricultural machinery and agronomy in hilly and mountainous areas. *J. Chin. Agric. Mech.* **2018**, *39*, 112–114.
9. Tian, F.Q.; Xiong, Y.; Hu, H.; Wang, L.J.; Zhuang, Q.T.; Yuan, R.H. Development status and suggestions of agricultural mechanization in Sichuan Provinces. *J. Chin. Agric. Mech.* **2024**, *45*, 321–327.
10. Sun, J.B.; Liu, Z.J.; Yang, F.Z.; Sun, Q.; Liu, Q.; Luo, P.X. Advancing agricultural mechanization in China's hilly and mountainous areas: Key strategies and technological focus. *Trans. Chin. Soc. Agric. Mach.* **2025**, *41*, 39–50.
11. Song, J.W.; Liu, H.X. Speed up breaking bottlenecks of the hilly agricultural mechanization development. *Trans. Chin. Soc. Agric. Mach.* **2010**, *31*, 16–19.
12. Zeng, Y.; Fang, Z.C. Status and development strategy of full mechanization of peanut in the hilly area of south China. *Agric. Eng. Equip.* **2021**, *48*, 4–9.

13. Li, S.; Yang, Y.; Zhang, X.H.; Zhu, X.L.; Li, J.; Xu, Y.J.; Hou, R.; Zhang, X.J. Current situation and development suggestions of mechanized production of peanuts in Sichuan province. *Bull. Agric. Sci. Technol.* **2024**, *4*, 145–147.
14. Wang, C.C.; He, L.; Chen, Z.W.; Deng, W.P.; Liu, W. Research status and development trends of peanut combined harvesters in Guangdong province. *China South. Agric. Mach.* **2025**, *56*, 20–23.
15. Li, N.; Jiang, W.; Zhou, J.; Zhang, H.; Cui, Z.K.; Di, Z.F. Present situation and countermeasures of peanut production mechanization in Shandong Province. *J. Chin. Agric. Mech.* **2019**, *40*, 42–50, 71.
16. Tang, Z.X.; Chen, S.Y.; Lin, Q.; Xu, R.R.; Chen, X.Y. Problems and countermeasures of peanut mechanical production in Fujian Province. *J. Chin. Agric. Mech.* **2016**, *37*, 235–238, 273.
17. Chen, C.Q.; Li, K.P.; Luan, X.Y. Experimental research for model selection of peanut combine. *J. Chin. Agric. Mech.* **2013**, *34*, 55–59.
18. Yang, R.B. Study on Key Study on Design Principle and Tests for Main Parts of 4HQL-2 Peanut Combine. Ph.D. Thesis, Shenyang Agricultural University, Shenyang, China, 2009.
19. Wang, D.W.; Shang, S.Q.; Zhao, D.J.; Zhu, H.J. Type-4HBL-4 two-ridges and four-lines semi-feeding self-propelled peanut combine harvester. *Trans. Chin. Soc. Agric. Mach.* **2013**, *44*, 86–92.
20. Shi, Z.W.; Chen, X.G.; Liu, H.S.; Sang, S.Q.; Wang, D.W.; He, X.N.; Zhang, Y.D.; Kong, X.M. EDEM-based simulation test of peanut seedling cleaning device sieving performance. *J. Agric. Mech. Res.* **2023**, *45*, 81–90.
21. Zhang, S.H.; Li, H.G.; Li, Z.P.; Yu, X.H.; Yang, F.J.; Gu, S.W. Design and test of intelligent semi-feed peanut combine harvester. *J. Agric. Mech. Res.* **2020**, *42*, 93–98.
22. Yang, R.B.; Wang, Z.Z.; Shang, S.Q.; Chen, M.D.; Wang, J.; Wang, Z.C. Design and test of straw coating device for peanut combine harvester. *Trans. Chin. Soc. Agric. Mach.* **2020**, *51*, 109–117.
23. He, X.N.; Li, M.Z.; Chang, X.L.; Sang, S.Q.; Wang, D.W.; Wang, J.S.; Yang, Z.A.; Wang, R.Z. Sign and experiment of the LS-DYNA-based disk-type peanut root-cutting device. *Trans. Chin. Soc. Agric. Eng.* **2025**, *41*, 32–42.
24. Wang, D.W. Study on Key Mechanism of Peanut Combine Harvester. Ph.D. Thesis, Shenyang Agricultural University, Shenyang, China, 2013.
25. Zhang, N.; Jiang, C.L.; Mao, P.H.; Wang, G.Y.; Wang, D.W.; Yuan, L.M.; Zhang, G.J. Design and Experiment of Shovel Chain Peanut Harvester Based on DEM - MBD. *Trans. Chin. Soc. Agric. Mach.* **2026**, *57*, 282–291.
26. Chai, K.J.; Hu, Z.C.; You, Z.Y.; Zhang, Y.H.; Wang, J.N. Modal analysis of the frame of a semi-inclined peanut harvesting machine based on ANSYS. *Jiangsu Agric. Sci.* **2016**, *44*, 407–409.
27. Ling, G.L.; Li, Z.F. *ANSYS 14.0 From Beginner to Expert*; Tsinghua University Academic Press: Beijing, China, 2013.
28. Fu, Z.F.; Hua, H.X. *Modal Analysis Theory and Application*; Shanghai Jiao Tong University Press: Shanghai, China, 2000.
29. Wang, D.W.; Shang, S.Q.; Han, K. Design and test of picking mechanism in 4HJL-2 peanut combines. *Trans. Chin. Soc. Agric. Eng.* **2013**, *29*, 15–25.
30. Ansys, Inc. *Ansys Learning Forum: Prestressed Harmonic Analysis*; Ansys, Inc.: Canonsburg, PA, USA, 2020. Available online: <https://forum.ansys.com/forums/topic/prestressed-harmonic-analysis-2/> (accessed on 6 March 2025).
31. Ansys, Inc. Fundamentals of Damping—Damping Effects (Lesson 2). Ansys Innovation Space, 2020. Available online: <https://innovationspace.ansys.com/courses/wp-content/uploads/2020/12/2.6.2-Fundamentals-of-Damping-New-Template.pdf> (accessed on 9 March 2025)).
32. Dhotare, S.A.; Mohan, V.M.; Chikurdekar, P.R. Effect of damping in layered beam using structural joints. In *Proceedings of the 2017 International Conference on Nascent Technologies in Engineering (ICNET-2017)*, Vashi, India, 27–28 January 2017; IEEE: Piscataway, NJ, USA, 2017.
33. Liu, P.; Jin, C.Q.; Ning, X.J. Field performance test and analysis of the cleaning sieve of soybean harvesters. *Trans. Chin. Soc. Agric. Eng.* **2020**, *36*, 36–43.
34. Wang, Z.L.; Kong, F.T.; Xie, O.; Zhang, Y.Y.; Sun, Y.F.; Wu, T.; Chen, C.L. Design and Testing of a Crawler Chassis for Brush-Roller Cotton Harvesters. *Agriculture* **2024**, *14*, 1832.
35. NAFEMS. The Importance of Mesh Convergence. NAFEMS Knowledge Base, 2020. Available online: <https://www.nafems.org/publications/knowledge-base/the-importance-of-mesh-convergence-part-1/> (accessed on 17 March 2025)).

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.