Finite element modal analysis and experiment of rice transplanter chassis

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Abstract: The vibration problem during the operation of rice transplanters is the most common phenomenon. In order that the static and dynamic characteristics of the rice transplanter chassis can meet the requirements of more stable operation, the research took the 2ZG-6DK rice transplanter as the research object to carry out a vibration reduction optimization study. In the research, the Pro/Engineer 5.0 software was first used to model the chassis of the rice transplanter. The constructed finite element model was revised by using the structural parameter revision method and the mixed penalty function method. The model was imported into ANSYS Workbench to solve the modal frequency and vibration shape of the rice transplanter chassis. Based on the MAC (modal assurance criterion) criterion, modal tests were carried out to verify the accuracy of the finite element theoretical analysis. Through the analysis of the characteristics of the external excitation frequency, the chassis is structurally optimized to avoid resonance caused by the natural frequency of the chassis falling within the road excitation frequency range. The final optimization results showed that the first four orders of modal frequencies of the chassis were adjusted to 32.083 Hz, 33.751 Hz, 42.517 Hz, and 50.362 Hz, respectively, in the case that the chassis mass was increased by 6.714 kg (8.8%). They all avoid the range of road excitation frequency (10-30 Hz) so that the rice transplanter can effectively avoid the resonance phenomenon during operation. This study can provide a reference for the design and optimization of the chassis structure of transplanter.

Keywords: transplanter chassis, vibration, finite element analysis, modal experiment, MAC criterion, optimization design
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1 Introduction

The chassis is the main component of transplanters. Its stress condition and stability during operation determine the stability of the whole machine, and directly affect the working quality and efficiency of the transplanter. When the high-speed transplanter is transplanting rice seedlings, it often bumps and vibrates due to the uneven and bumpy working conditions in the fields. In addition, it will also be affected by the vibratory excitation of the internal combustion engine and other vibratory shock loads inside the system. Moreover, resonance will occur when the vibration frequency of the external excitation of the machine draws close to the inherent frequency of an order of the transplanter[1,2]. The dynamic stress generated by resonance not only affects driving comfort, but also may cause fatigue crack of components, accelerate the damage of machine parts, reduce the working efficiency and working quality of the machine, and shorten the service life of the machine[3,4]. Therefore, it is of great practical significance to carry out research on the vibration characteristics of the transplanter chassis for the purpose of vibration reduction. To be specific, such research helps avoid structural damage to the machine in the practical process, improves efficiency, and provides a certain theoretical basis for the improvement and optimization of the transplanter chassis in the future.

To avoid structural resonance, the fundamental frequency or frequency doubling of the structure itself shall not be equal to the external excitation frequency. Increasing the system stiffness without changing the mass, increasing the mass without changing the stiffness, increasing the damping to reduce the response, reducing the damping to improve the response, and changing the excitation function of the system are common methods to avoid resonance[5,6]. Li et al.[7] analyzed the vibration radiation noise of the underground platform of rail transit and found that the natural frequency of the platform cavity is the main resonance frequency of the platform system. The natural frequency is changed by changing the platform height, so as to improve the resonance phenomenon of the platform system. Yao et al.[8] conducted vibration modal test on the frame of maize harvester. By increasing the wall thickness and stiffness of the frame, the modal natural frequency of the frame increases accordingly, so as to avoid the resonance between the frame and the engine.

So far, some analyses and conclusions have been proposed on the problem of violent vibration during normal operation in previous studies of chassis vibration of agricultural machinery equipment in the world. Some researchers have also been conducted on aspects of vibration characteristics of chassis, vibration at driving position, and comfort level during driving. In addition, further research has also been conducted in respect of modal analysis and optimization of sensor test point selection. Jin et al.[9] determined the sensitive parameters of cassava cropper

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frame and established the multi-objective optimization model. In terms of analytical method, Jin et al.[10] obtained the damping ratio and inherent frequency of three structural parts of transplanters, and analyzed the modal modes of the structural parts through the finite element analysis in the ANSYS software. By comparing with the modal parameters of other different structural parts, they provide an important theoretical basis for the structural update of transplanters. Liao et al.[11] obtained the amplitude statistics and power spectrum at some observation points in the frame by analyzing the vibration characteristics of the cropper frame, and analyzed the influence of power and amplitude on the frame vibration characteristics. Through further research, they also obtained the relationships between the wall thickness and rigidity of the frame and the inherent frequency of the frame, and optimized the first-order torsion frequency of the frame.

At present, relevant research results have been achieved in vibration testing and characteristics analysis of agricultural machinery equipment such as transplanters[12,13]. Most of the research results on vibration analysis are concentrated in the fields of aerospace and bridge construction, and there is a large gap in the research on the vibration characteristic analysis of agricultural machinery equipment such as transplanters during operation[14,15].

2 Materials and methods

2.1 Test methods and objects

In modal test, a dynamic test is performed based on external excitation and system response[16]. The excitation force is input and the response data is output through the system. The response data is subjected to parameter identification after signal processing to determine the modal parameters of the system. In the modal test, a vibration exciter or a modal impulse hammer is often used to force the structure of the experimental subject to generate force vibration, and then the frequency, damping, and mode of vibration of the structure are identified by analyzing the mechanical admittance function between the two points, thereby providing a basis for the vibration characteristics analysis of the structural system and the optimization design of the structural dynamic characteristics[17].

The modal test system used in this study consists of a data acquisition module and a modal analysis module[18,19]. The data acquisition module is composed of a three-axis acceleration sensor, impulse hammer, and DH5902 data acquisition instrument; the modal analysis module is composed of DHMA modal analysis software and DH5902 dynamic signal analysis system. Figure 1 is the modal test principle diagram of the rice transplanter chassis.

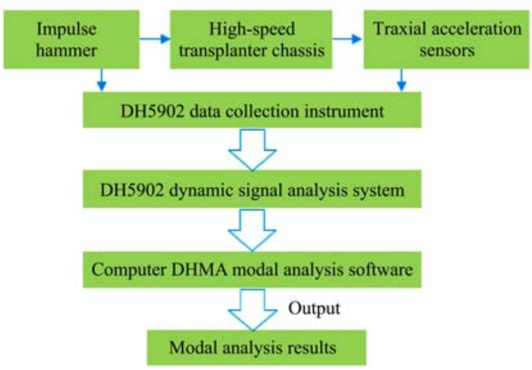
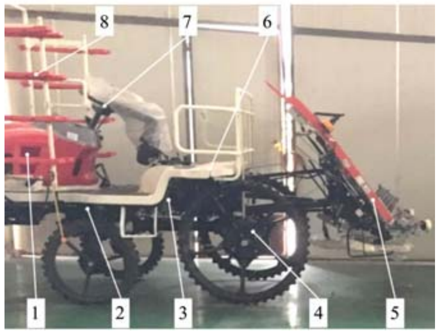


Figure 1 Modal test principle diagram of transplanter chassis

The structure of 2ZG-6DK high-speed rice transplanter is shown as follows, mainly composed of driving parts, transmission mechanism, walking mechanism, chassis frame, rice transplanting system, etc. The structure of the whole machine is shown in Figure 2. The chassis frame is a spatial thin-walled beam structure, which is composed of 2 longitudinal beams with a length of 1500 mm, 8 transverse beams with a length of 1200 mm, 2 support beams, and two connecting steel pipes. The two main longitudinal beams have a hollow rectangular cross section; the transverse beams carrying the planting mechanism have an annular cross section, and other load-bearing transverse beams have an annular cross section; the connecting longitudinal beams are round steel pipes. The diagram is shown in Figure 3. The chassis is the main load-bearing component that supports the entire transplanter. Its stress condition and working stability determine the stability of the whole machine, and even directly affect the quality and efficiency of the transplanter.



1. Drive components 2. Transmission mechanism 3. Chassis 4. Walking mechanism 5. Transplanting system 6. Seat 7. Steering device 8. Loading platform

Figure 2 Structure diagram of high-speed rice transplanter

1. Crossbeam 1 2. Crossbeam 2 3. Connecting beam1 4. Crossbeam 3 5. Crossbeam 4 6. Stringer 1 7. Stringer 2 8. Crossbeam 5 9. Crossbeam 6 10. Connecting beam 2 11. Crossbeam 7 12. Crossbeam 8 13. Support beam 1 14. Support beam 2

Figure 3 Structure diagram of transplanter chassis

2.2 Experimental equipment and system composition

In the modal test, the DH5902 dynamic signal test and analysis system is used to test the vibration characteristics of the 2ZG-6DK transplanter chassis[20]. The test equipment and parameters are shown in Table 1 below.

The modal test has sufficient amplitude level and excitation component for excitation signals, and sets a certain requirement for anti-jamming capability when there is a slight nonlinear factor with the structure. Commonly used excitation signals include sinusoidal excitation signal, pulse signal random excitation signal, random excitation signal, etc.[21] In particular, the force signal of the pulse excitation has a wide frequency spectrum, and multi-order modals can be excited simultaneously by one time of excitation. The spectrum range mainly depends on the stiffness of the contact surface and the mass of the pulse hammer head. Considering that the first four frequencies of the frame model are within 60 Hz, the sampling frequency range is 20-150 Hz. It will not generate side effects such as additional mass or additional rigidity on the test object, and the test equipment features simple operation and great flexibility, making it extremely suitable for field test. Therefore, this study uses the pulse excitation method for the modal test to infer the inherent characteristics of the system by measuring the input and output signals at the same time. The excitation points should not be nodes of vibration mode of any order of the system. In addition, in order to ensure that the signals collected at the test points have a high signal-to-noise ratio while reducing and even avoiding modal omissions, the measuring point should be selected at a location with high rigidity and easy transmission of the excitation signal. The test points should be located at positions with good rigidity and that are convenient for the transmission of excitation signals. The test points should also reflect the structural contour of the test object to a greater extent, avoid the nodes of vibration mode of any order, and clearly show the characteristics of modal vibration mode[22].

Table 1 Equipment for modal testing

Equipment	Instrument No.	Performance index	Parameters	Manufacturers
Three-axis acceleration sensor	356A16	Range/g	50	PCB
		Frequency response/kHz	0.3-6.0	
		Sensitivity/mV·g^{-1}	100	
		Transverse sensitivity/%	5	
Modal impulse hammer	086C03	Output deviation Sensitivity/mV·N kHz	10.3	PCB
		Sensitivity/mV·g^{-1}	2.28	
Data acquisition instrument	DH5902	Number of channels	12	Donghua Test
		Maximum sampling frequency/kHz	100	
		Degree of distortion	0.5	
		Signal input mode	IEPE	
Dynamic Signal Analysis System	DHDAS5902	Maximum analysis frequency/kHz	39.06	Donghua Test
Modal Analysis Software	DH2.6.2	--	--	Donghua Test

Theoretically, the acceleration admittance between the ground and the test object should be zero in the test, and its vibration response should only contain frequency components above tens of thousands of hertz. However, in the actual test, it is difficult to ensure that the admittance at the connection point between the structure of the test object and the ground is much smaller than that at other points. Therefore, a spring that is much less than the rigidity of the test object is used to suspend the chassis to ensure that the chassis is suspended horizontally in the air. When the highest rigid body modal frequency of the support system is controlled at 1/5 to 1/10 of the lowest elastic modal frequency of the test object, the errors caused by the support system can be ignored. The three-axis acceleration sensor is placed on the chassis surface as required and then is connected to the test system in turn with wires. A modal impulse hammer is used to strike the excitation point, and the collected data is identified in the frequency domain. The span of discrimination is selected according to the test contents, the damping and frequency are calculated by the peak-picking technique, and the residue is calculated by the peak-to-peak method. After the modal frequency is calculated, the calculated data is transmitted to the data acquisition system by the modal impulse hammer and acceleration sensor, thus obtaining the system response function. The vibration characteristics of the transplanter chassis can be obtained after the response function of the overall system is analyzed by the modal analysis system.

2.3 Sensor layout at modal test points

In order to improve the recognition precision of modal parameters, the locations of the excitation points should be reasonably arranged to minimize the loss of modal parameters. The function curves selected in this study are clear and smooth. In the purposeful frequency range, points with function values above 0.9 (including 0.9) were selected as the excitation points. The frequency response functions of several excitation points were measured and compared for analysis. Single-point excitation should ensure that the excitation signals can be transmitted to three directions of the chassis. The dynamic properties of the chassis should be solved as precisely as possible. All points of action of external forces, connection points of structures and components, mass concentration points, and important response points were used as test points. The three directions of some necessary test points should be measured. The test points with weak rigidity and easily produced structural vibration and radiated noise should be arranged densely as appropriate[23]. The layout of test points must clearly show the structural modal vibration modes within the frequency range and ensure that the key points of the research are included in the scope of test points.

Sensor layout directly determines the validity of data collected by the test system. The traditional modal tests use uniformly distributed points and many sensors, which increases the experimental costs and the difficulty of data processing. The chassis tested in this study had an irregular structure, and the test noise in the actual experiment will cause problems such as non-orthogonal modal vectors, affecting the accuracy of test data. Thus, it is necessary to optimize the sensor layout, place a fixed number of sensors in the optimal position, and introduce the modal assurance criterion (MAC). The matrix can be expressed as:

$$\mathbf{MAC}_{ij} = \frac{(\boldsymbol{\Phi}_i^{\mathrm{T}}\boldsymbol{\Phi}_j)^2}{(\boldsymbol{\Phi}_i^{\mathrm{T}}\boldsymbol{\Phi}_j)(\boldsymbol{\Phi}_j^{\mathrm{T}}\boldsymbol{\Phi}_j)} \tag{1}$$

where, $\boldsymbol{\Phi}_i$, $\boldsymbol{\Phi}_j$ are the modal vectors of the i^{th} and j^{th} order, respectively. The value range of the non-diagonal elements of the $\mathbf{MAC}_{ij}$ matrix is [0,1]. In the process of optimizing the sensor layout, the $\mathbf{MAC}_{ij}$ matrix should be as small as possible, proving that the more the modal vector tends to be orthogonal, the more reasonable the layout of sensor test points.

The arrangement and selection of measuring points mainly consider that the engine, seedling platform, and planting mechanism are the main vibration sources of the chassis vibration of the transplanter. In addition, it is necessary to ensure that the excitation point and measurement point are at a certain distance from the modal node. The dynamic equation of the structure after finite element analysis can be expressed as:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = Bu(t), y(t) = D\ddot{x}(t) \tag{2}$$

where, M is the mass matrix; D is the damping matrix; K is the stiffness matrix; C is the excitation point matrix; $u(t)$ is the excitation force vector; $y(t)$ is the output vector.

Through $x(t)=\Phi\eta(t)$, $\Phi\equiv[\varphi_1,\varphi_2,\ldots,\varphi_N]$, Equation (2) can be transformed into modal coordinates:

$$\ddot{\eta}(t)+(\alpha I+\dot{\beta}\wedge)\dot{\eta}(t)+\wedge\eta(t)=\bar{B}u(t), y(t)=\bar{D}\ddot{\eta}(t) \tag{3}$$

$$\wedge\equiv\mathrm{diag}(\lambda_1,\lambda_2,\ldots,\lambda_N) \tag{4}$$

$$\bar{B}\equiv\Phi^T B=[\bar{B}_1,\bar{B}_2,\ldots,\bar{B}_N] \tag{5}$$

$$\bar{D}\equiv D\Phi=[\bar{D}_1,\bar{D}_2,\ldots,\dot{\bar{D}}_N] \tag{6}$$

$$\eta(t)\equiv\{\eta_1(t),\eta_2(t),\ldots,\eta_N(t)\}^T \tag{7}$$

It is difficult to determine from Equation (2) whether excitation force $\bar{B}_0u(t)$ can excite all the modal information. Therefore, the singular value of $\bar{B}_0\in R^{r\times P}$ is decomposed to obtain:

$$\bar{B}_0=U_0S_0V_0^T$$

where, $U_0\in R^{r\times r}$, $V_0\in R^{P\times P}$, $S_0=\begin{bmatrix}\sum_0 & 0\\ 0 & 0\end{bmatrix}$, $\sum_0=diag(\sigma_{01},\sigma_{02},\ldots,\sigma_{0m})$, $\sigma_1\geq\sigma_2\geq\ldots\geq\sigma_m\geq0$, $m\leq Min(r,P)$, $U_0^TU_0=I_r$, $V_0^TV_0=I_P$. Meanwhile, a set of repetitive mode coordinates $\rho_0(t)$ is introduced. Taking $x(t)=[\Phi_0U_0\quad\Phi_d]\begin{Bmatrix}\rho_0(t)\\ \cdots\\ \eta_d(t)\end{Bmatrix}$ into Equation (1), then

$$\begin{Bmatrix}\ddot{\rho}_0(t)\\ \cdots\\ \ddot{\eta}_d(t)\end{Bmatrix}+\begin{bmatrix}(\alpha+\beta\lambda_0)I_r & 0\\ 0 & \alpha I_d+\beta\wedge_d\end{bmatrix}\begin{Bmatrix}\dot{\rho}_0(t)\\ \cdots\\ \dot{\eta}_d(t)\end{Bmatrix}+\begin{bmatrix}\lambda_0I_r & 0\\ 0 & \wedge_d\end{bmatrix}\begin{Bmatrix}\rho_0(t)\\ \cdots\\ \eta_d(t)\end{Bmatrix}=\begin{bmatrix}U_0^T\Phi_0^TB\\ \cdots\\ \Phi_0^TB\end{bmatrix} \tag{8}$$

Remove the coordinate equation of repeated frequency mode from Equation (8) and make $U_0^T\Phi_0^TB=S_0V_0^T$, get:

$$\ddot{\rho}_0(t)+(\alpha+\beta\lambda_0)\dot{\rho}_0(t)+\lambda_0\rho_0(t)=S_0V_0^Tu(t) \tag{9}$$

Thus, the necessary and sufficient condition for all repetition frequency modal coordinates to be excited is: $M=R$. And the necessary condition of $M=R$ is $p\geq R$. Where m is the number of nonsingular values, P is the number of excitation points, and R is the modal multiplicity.

A given number of DOF numbers is input through the Matlab program, and the node position and direction of the corresponding model can achieve the purpose of optimizing the test points. The optimized test points are shown in Figure 4, and there was a total of 64 test points. The model structure can better define the outline shape of the rice transplanter chassis. The model has 64 modal test sites, which can completely define the modal parameter model and maintain good test efficiency.

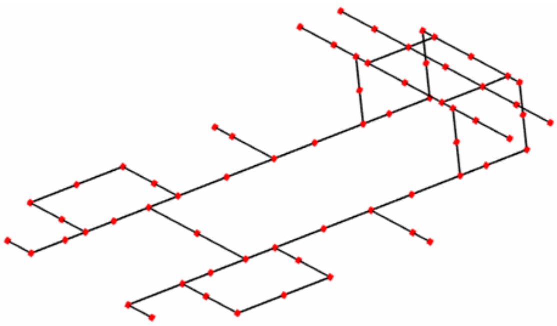


Figure 4 Schematic diagram of test point arrangement

Figure 5 Physical layout of test points

2.4 Building of finite element model

In the dynamic analysis of any structure, it is necessary to establish its physical model and mathematical model. Through the calculation of the model, the motion parameters such as displacement, velocity, and acceleration of the structure are solved, and then the dynamic behavior of the structure is accurately described. In order to analyze the vibration behavior of the transplanter chassis frame, it is necessary to establish the finite element model of the transplanter chassis frame for modal analysis. Modal analysis is an analytical method of vibration analysis. It transforms the physical coordinates of each mass motion of the system into modal coordinates and converts the physical equations describing the motion of the system into equations described by modal coordinates and modal parameters. Through modal analysis, the problem of multi-degree of freedom system can be decomposed into multiple single degree of freedom systems in modal space, and then each modal component can be superimposed and transformed into the actual physical space.

Mode refers to the inherent vibration characteristics of mechanical structures. Each mode has a specific inherent frequency, damping ratio, and modal vibration mode. Each order corresponds to a mode, and each order has a specific frequency, damping ratio, and vibration mode. Modal analysis is a method to study the dynamic characteristics of structures. The inherent frequency and vibration mode can be determined through the theoretical modal analysis.

The modal analysis is planned to be carried out on the transplanter chassis following the idea of "breaking up the whole into parts, and integrating parts into the whole". In the process of dynamic analysis, this not only makes the computational process clearer but also ensures better computational accuracy. After dividing the complex chassis structure into several sub-structures, the dynamic characteristics of each sub-structure were obtained through model tests, and then the sub-structures were combined again to simplify the calculation of complex structures.

The structural material characteristics of the transplanter chassis determine its rigidity characteristics. In the case of ignoring the impact of the damping of the transplanter chassis, the vibration of the structure can be written as:

$$M\ddot{x}+Kx=f \tag{10}$$

where, M is the mass matrix of the system; K is the rigidity matrix of the system; x and f are the displacement vector and force vector of the structure.

The finite element models generally use elements to connect nodes and build relationships of different degrees of freedom. If the internal degrees of freedom of substructures are represented by 1 and 3, respectively, and the interface degrees of freedom of the interface of connected substructures are represented by 2, the rigidity matrix and mass matrix of the structure are defined as follows:

$$M=\begin{bmatrix} M_{11} & M_{12} & 0 \\ M_{21} & M_{22} & M_{23} \\ 0 & M_{32} & M_{33} \end{bmatrix} \tag{11}$$

$$K=\begin{bmatrix} K_{11} & K_{12} & 0 \\ K_{21} & K_{22} & K_{23} \\ 0 & K_{32} & K_{33} \end{bmatrix} \tag{12}$$

According to the theory of modal analysis, the superposition of a series of simple harmonic vibrations can constitute the free vibration of any elastomer. The modal vibration mode of the overall structure is formed by the superposition of vibrations of substructures and connecting interface structure of substructures. The simple harmonic motion of the structure can be expressed as:

$$x = X\sin(\omega_n + \varphi) \tag{13}$$

where, X is the node amplitude of the chassis, mm; ω_n is the inherent frequency of the system, Hz; φ is the phase angle, (°).

The vibration modes of substructures can be obtained by the equation as follows:

$$K_{ii}\phi_i = \Lambda_i M_{ii}\phi_i (i=1,3) \tag{14}$$

where, Λ_i, ϕ_i are the eigenvalue and eigenvector matrix of the ith substructure, respectively.

The vibration modes of the first n orders of the substructure include:

$$\phi_i = [\phi_{i1}, \phi_{i2}, ..., \phi_{in}](i=1,3) \tag{15}$$

$$\begin{bmatrix} K_{11} & K_{12} & 0 \\ K_{21} & K_{22} & K_{23} \\ 0 & K_{32} & K_{33} \end{bmatrix}\begin{bmatrix} \phi_1 \\ \phi_2 \\ \phi_3 \end{bmatrix} = \begin{bmatrix} \Lambda_1 & 0 & 0 \\ 0 & \Lambda_2 & 0 \\ 0 & 0 & \Lambda_3 \end{bmatrix} \times \begin{bmatrix} M_{11} & M_{12} & 0 \\ M_{21} & M_{22} & M_{23} \\ 0 & M_{32} & M_{33} \end{bmatrix}\begin{bmatrix} \phi_1 \\ \phi_2 \\ \phi_3 \end{bmatrix}$$

The vibration displacement of the first n orders of the substructure interface AB is:

$$\begin{bmatrix} \phi_1^{AB} \\ \phi_2^{AB} \\ \phi_3^{AB} \end{bmatrix} = \begin{bmatrix} \phi_{11}^{AB} & \phi_{12}^{AB} & \cdots & \phi_{1n}^{AB} \\ \phi_{21}^{AB} & \phi_{22}^{AB} & \cdots & \phi_{2n}^{AB} \\ \phi_{31}^{AB} & \phi_{32}^{AB} & \cdots & \phi_{3n}^{AB} \end{bmatrix} \tag{16}$$

Based on the vibration equation and simple harmonic motion equation of the chassis structure, the following can be obtained:

$$(K-\omega_n^2 M)X = 0 \tag{17}$$

Under the condition free of excitation, if the free vibration is triggered to the chassis by initial conditions, there must be a displacement node, and $(K-\omega_n^2 M)X = 0$ must have a non-zero solution, i.e.:

$$|K-\omega_n^2 M| = 0 \tag{18}$$

where, M and K are determined by inherent attributes such as mass and geometry of the system. The solutions of the equation $|K-\omega_n^2 M| = 0$ can be obtained by setting the response frequency. If the system has n orders of degree of freedom, the equation $|K-\omega_n^2 M| = 0$ is the nth-power algebraic equation about ω_n^2, thus n eigenvalues and their corresponding eigenvectors can be obtained, i.e. the inherent frequency and vibration mode of the chassis system.

2.5 Finite element calculation and model updating method

The chassis, as an important part of the high-speed transplanter, is a support of the whole machine. In order to make the performance of static and dynamic characteristics of the transplanter chassis satisfy the requirements for slight vibration and stable operation, the structure of the transplanter chassis needs to be optimized to improve the dynamic performance of the transplanter. The finite element method is used to establish a model of the high-speed transplanter chassis, and the modal analysis is conducted for the model. The model needs to be imported into ANSYS software to determine its frequency, damping ratio, and vibration mode. However, due to the deviation between the actual value and the finite element model value, the finite element model of the chassis needs to be revised. The finite element method is used to replace the original non-individual body with a collection of finite number of elements, which can satisfy the requirements of optimum structural design.

In the modeling process, the plate element is selected as the basic discrete element. Double-click mesh under component systems to use an independent meshing module. After importing the model, right-click A2 - edit geometry in design modeler, edit the geometric model in DM, and then right-click import generate to generate the model. After generating the model, right-click mesh generate mesh to automatically generate the mesh. After generating the mesh, Click mesh> Quality> mesh quality: element quality to check the mesh quality. The abscissa is the distribution of mesh quality from 0 to 1, and the standard is the side length ratio, which is closer to "1" It shows that the closer the length of each side of the grid is, the better the quality of the grid is; The ordinate is the number of grids. In Figure 8, the grid quality is concentrated below 0.6 and has a maximum value of 0.1, so the grid quality is not ideal. Then set the default physical reference of meshing to further improve the mesh quality. The default mesh is used for CFD analysis, and the system will automatically refine the mesh to improve the mesh quality. In Figure 8, the grid quality is obviously concentrated above 0.5, and the maximum value is 0.86.

After the finite element model is built, idealized simplification and assumptions of the structure will often be made. Therefore, there will always be a deviation between the response value analyzed and predicted by the finite element analysis and the results obtained from experiments. The actually measured response information is used to update the finite element model of the structure, making the calculated response value of the updated finite element model consistent with the experimental value. This process is exactly the updating of the finite element model of the chassis structure. The finite element model updating methods mainly include the system matrix updating method and the structural parameter updating method. The system matrix updating method is mainly used to update the structural rigidity and mass matrix. These are generally updated by solving matrix equations or optimization, and it is assumed that the changes in rigidity and mass do not affect each other. The structural parameter updating method mainly updates the structure through the sensitivity of the structural parameters and often uses observations such as vibration, mode shape, and frequency response function.

The matrix model updating method needs to be carried out with the help of the mass matrix and the rigidity matrix, but it is difficult to be used in actual applications because its updating results have no clear physical meaning. The direct updating of design parameters such as structural materials, cross-sectional shapes, and geometric dimensions has a clear physical meaning, making it the most suitable model updating method for engineering applications at present. Based on the first-order optimization design of ASNSYS software, this study adopts an iterative optimization method to update the model of the transplanter chassis.

In the ANSYS optimization, the objective function, design variables (DVs), and state variables (SVs) are called together as optimization variables. In the ANSYS optimization of the transplanter chassis, the objective function is the relative errors of the inherent frequency of the model of the transplanter chassis. Design variables are uncertain model cross-section parameters in the modeling process, and state variables are the inherent frequencies of the first four orders of the model. In order to select design variables with better accuracy and sensitivity, we used ASNSYS software to obtain the sensitivity of the objective function and state variables to the design variables, and then compare the sensitivities to sort out the most suitable quantity, and finally take appropriate optimization methods for iterative optimization on each quantity to obtain a finite element model with higher accuracy.

Design variables can be written as $x=[x_1,x_2,...,x_n]$, $x_j' \le x_j \le x_j''$ $(j=1,2,...,n)$, where, the lower limit and upper limit of the design variable x_j can be expressed as x_j', x_j''.

The reference state of the objective function is $f_t(x)=f(x^{(t)})$, so the objective function can be expressed as:

$$f_t=\left[\frac{\partial f_t}{\partial x_1},\frac{\partial f_t}{\partial x_2},...,\frac{\partial f_t}{\partial x_n}\right] \tag{19}$$

The ANSYS program provides two optimization methods: the zero-order approximation method and the first-order method. The zero-order approximation method can solve most engineering optimization problems. The first-order method optimizes engineering problems based on the sensitivity between the objective function and the design variables. In order to improve the optimization of the chassis model, we choose the first-order method.

Minimum value:

$$f=f(x) \tag{20}$$

Constraint conditions:

$$\begin{cases} x_j' \le x_j \le x_j'' \ (j=1,2,...,n) \\ g_i \le g_i'' \ (i=1,2,...,m_1) \\ h_i' \le h_i(x) \ (i=1,2,...,m_2) \\ \omega_i' \le \omega_i(x) \le \omega_i'' \ (i=1,2,...,m_3) \end{cases} \tag{21}$$

The mixed penalty function method is used to convert it into an unconstrained single-object optimization problem with dimension as one. The penalty function is expressed as:

$$Q(x,q)=\frac{f}{f_0}+\sum_{j=1}^{n}p_x(x_j)+q\left[\sum_{i=1}^{m_1}p_g(g_i)+\sum_{i=1}^{m_2}p_h(h_i)+\sum_{i=1}^{m_3}p_\omega(\omega_i)\right] \tag{22}$$

where, P_x, P_g, P_h, and P_W are penalty factors of state variables and design variables. Thus, the gradient method can be used to optimize problems. The iterative equation can be written as:

$$x^{(j+1)}=x^{(j)}+s_j d^{(j)} \tag{23}$$

where, s_j is the optimal step factor.

The convergence conditions of interaction are:

$$\begin{cases} \left|f^{(j)}-f^{(j-1)}\right| \le \tau \\ \left|f^{(j)}-f^{(b)}\right| \le \tau \end{cases} \tag{24}$$

where, m is the accuracy requirement of the objective function. The flowchart of optimizing the chassis model with the mixed penalty function method is shown in Figure 6.

During the finite element analysis, a finite element mechanical model is built for the chassis structure using the beam element and the bar element, and the updating of the finite element dynamic model is finished. The errors (structure error, order error, and parameter error) in the finite element model are caused by the different model materials, and the differences in boundary conditions and geometric parameters make the finite element model inaccurate. By analyzing the finite element model of the chassis, it is found that there are obvious errors in the following parts:

1) The modulus of elasticity of the chassis support E;

2) The chassis height H;

3) The parameters such as the front and rear beam frame span L and the beam cross sections; the section parameters of beam cross were difficult to be determined.

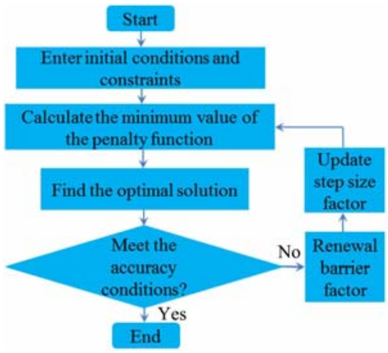


Figure 6 Flow chart of mixed penalty function method

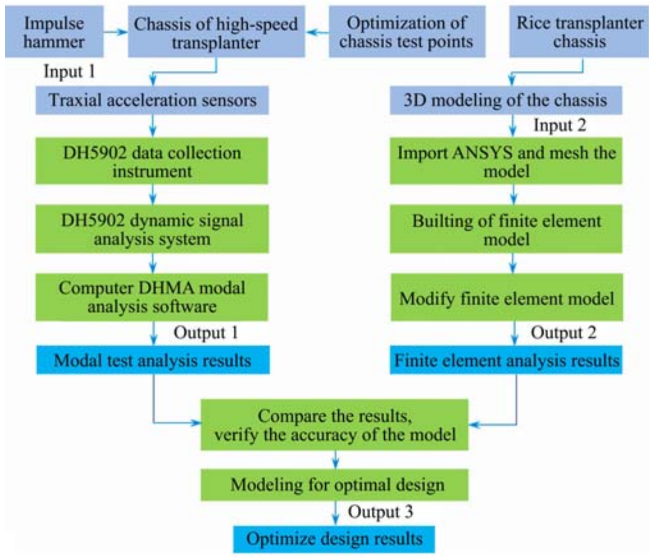


Figure 7 Modal analysis and experimental principle of the rice transplanter chassis

The eigenvalue sensitivity analysis is performed for the above parts, and the non-sensitive parameters are excluded. The design variables E, H, L composed of the parameters to be updated are determined. The mathematical model and constraint conditions of the optimal design of the chassis structure are obtained as follows:

$$|K^2-f^2M|=0 \tag{25}$$

$$\max f_1=f_1(X) \tag{26}$$

$$x=[x_1 \quad x_2 \quad x_3]=[E \quad H \quad L] \tag{27}$$

$$\text{s.t.}\begin{cases} 2\times10^{10} \le E \le 3\times10^{10} \\ 40.0 \le H \le 700 \\ 350.0 \le L \le 1500 \end{cases} \tag{28}$$

where, f_1 is the inherent frequency of the first-order mode of the chassis; E, H, L represent the modulus of elasticity (mm), height (mm), and front and rear beam frame span (mm), respectively.

In the optimization process, the solution procedure of the system's inherent frequency is integrated into the framework of the multi-disciplinary optimization design ISIGHT software, and then the input files and output files are parsed. ISIGHT software uses the sequential quadratic programming method to optimize the design variables and objective responses, carries out iterative calculations according to the preset number of iterations, and integrates the entire process into a system that can be automatically executed[25].

3 Results and discussion

3.1 Calculation results of the chassis model

Nantong FLW 2ZG-6DK high-speed transplanter has a chassis made of spatial thin-walled beams, with a length of about 1655 mm and a width of about 1300 mm. The chassis is composed of a non-loaded spatial boundary beam with variable heights, two main longitudinal beams with hollow rectangular cross-sections, and several load-bearing transverse beams with annular cross sections.

A solid finite element model of the transplanter chassis needs to be built, i.e. a three-dimensional solid model of each key component of the transplanter chassis is built using the Pro/Engineer 5.0 software. The three-dimensional model needs to be simplified in order to increase the computation speed, and the simplified conditions include:

(1) The influence of the support weld on the vibration characteristics of the chassis is not considered;

(2) All auxiliary holes and undersized structures are ignored.

The model was analyzed using ANSYS 18.1 software. We use Q235 structural steel as the model material, and set the material parameters as elastic modulus E=210 GPa, Poisson's ratio 0.3, density 7850 kg/m^2, tetrahedral grid, grid encryption size 0.01, grid average mass 0.5. The number of generated grids is 102 134 and the number of nodes is 51 906. The division of the finite element model grids is shown in Figure 8.

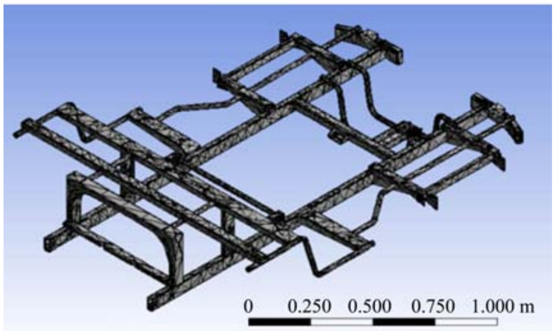
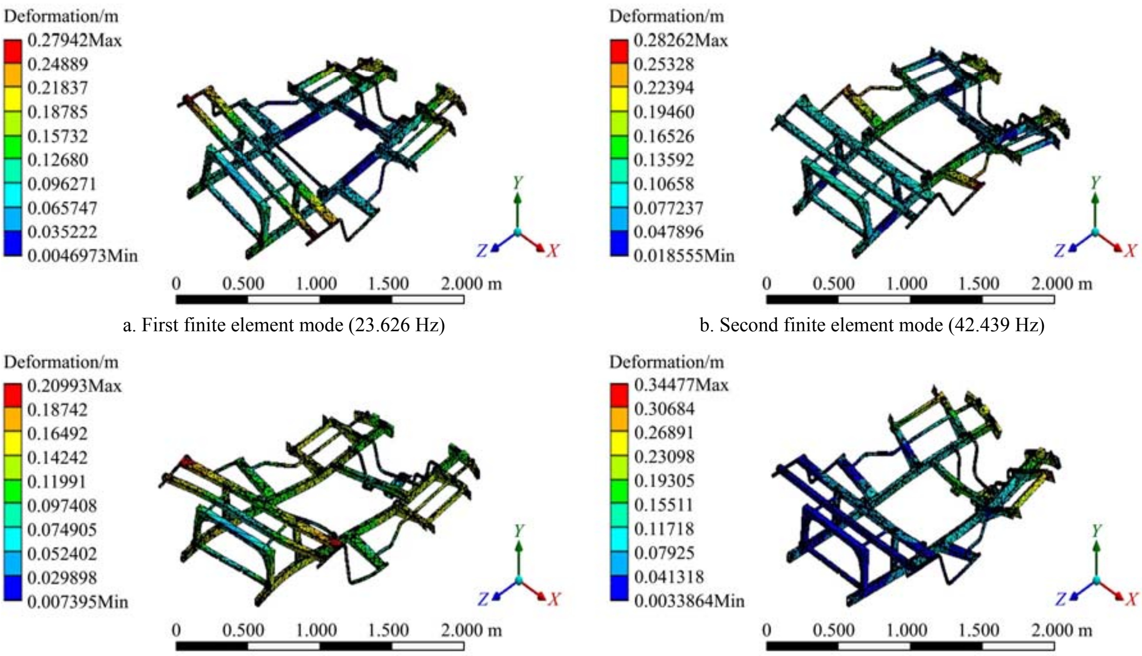


Figure 8 Finite element model of chassis

After the system grids are divided, the solid model is imported into the ANSYS software for the modal analysis of the rice transplanter chassis. The modal vibration shape of the rice transplanter chassis is shown in Figure 9.



a. First finite element mode (23.626 Hz)

b. Second finite element mode (42.439 Hz)

c. Third finite element mode (50.73 Hz)

d. Fourth finite element mode (57.856 Hz)

Figure 9 Finite element modal analysis results of transplanter chassis

Table 2 Finite element modal frequency and vibration mode of 1st-4th order of transplanter chassis

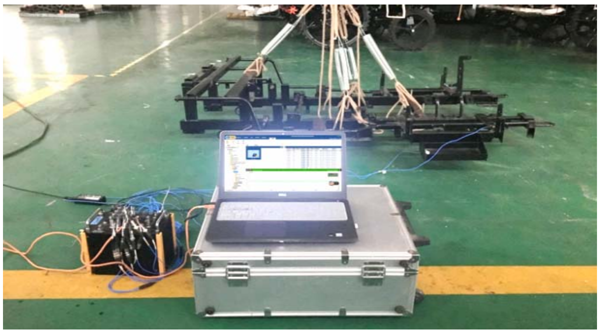
Order	Frequency/Hz	Vibration mode
1st order	23.626	Overall torsion
2nd order	42.439	Bend forward and backward
3rd order	50.73	Overall torsion
4th order	57.856	Overall torsion

3.2 Modal test results

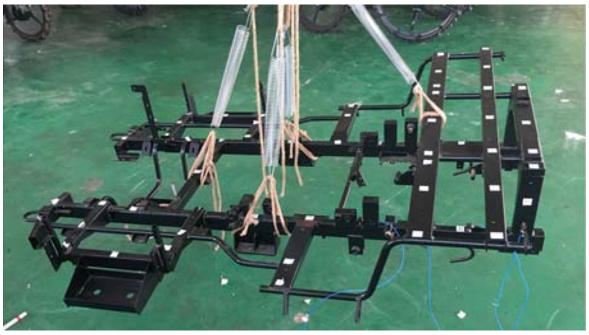
In modal testing or natural frequency testing, analysis and calculation are required based on the frequency response function of the research object. In order to test the frequency response function of the rice transplanter chassis, this experiment was carried out by the method of free excitation. This method uses the free support method to make the test object in a suspended state. In the test, 8 springs are used to hang on the four points of the rice transplanter chassis, so that the chassis is in a horizontally suspended state. In order to avoid structural rigidity changes caused by static stress when the structure is suspended, and to ensure the stability of the suspension system, the suspension point should be selected as close as possible to the node with the greater structural rigidity of the test object. When the excitation signal reaches a certain high amplitude, it can excite all resonance frequencies or resonance modes of the measured object. In order to ensure that the identified frequency response function does not miss a certain mode, the excitation point should not be selected near the node or node line. In the test, two excitation directions, lateral (Y-axis direction) and vertical (Z-axis direction) are selected for testing. The test site is shown in Figure 10.

Import the data obtained by the data acquisition instrument into the DHDAS modal analysis software for analysis. Then match the modal parameter data to the established modal test model. Thereby, modal parameters such as the natural frequency and vibration shape of the rice transplanter chassis are obtained. Because the dynamic characteristics of the structure are mainly determined by the low-order mode shape, the low-order vibration has a greater impact on the dynamic characteristics of the structure. Therefore, the first 4 natural frequencies and vibration modes of the chassis frame of the rice transplanter were extracted for analysis. The Cloud charts of vibration modes were shown in Figure 11.

The finite element mode is compared with the experimental mode as shown in Table 3.

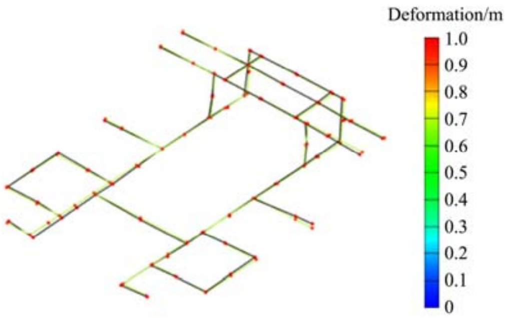


a. Test mode equipment connection diagram

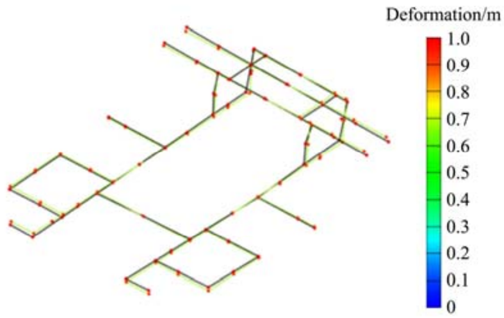


b. Modal test site

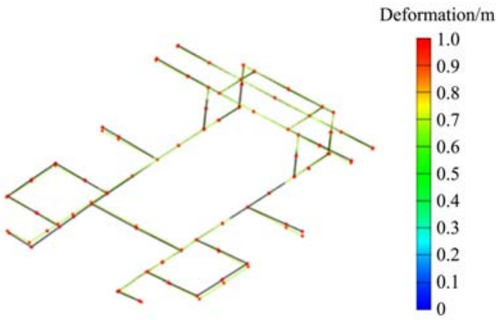
Figure 10 Field diagram of modal test



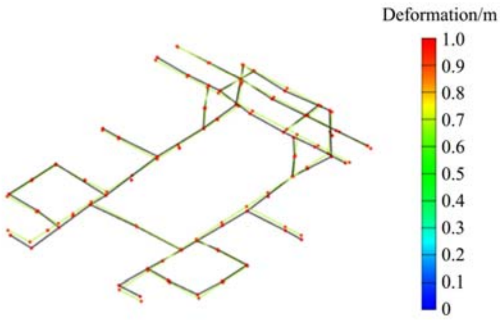
a. First order vibration mode



b. Second order vibration mode



c. Third order vibration mode



d. Fourth order vibration mode

Figure 11 Cloud charts of vibration modes of the first four orders in the chassis modal test

Table 3 Comparison of finite element mode and experimental mode results

Order	Mode of vibration		Vibration frequency		Frequency error	Frequency relative error rate/%
	Finite element	Test	Finite element	Test		
1	Overall torsion	Overall torsion	23.626	23.796	−0.170	0.71
2	Bend forward and backward	Bend forward and backward	42.439	42.428	0.011	0.03
3	Overall torsion	Overall torsion	50.730	50.838	−0.108	0.21
4	Overall torsion	Overall torsion	57.856	58.635	−0.779	1.33

Table 3 shows that the finite element mode of the transplanter chassis is basically identical to the experimental mode. The inherent frequencies of the first four orders were 23.626 Hz, 42.439 Hz, 50.730 Hz, and 57.856 Hz, respectively. The inherent frequency of each order increased with the increase of the order, but no regular rules were shown. The vibration modes of the first four orders were dominated by overall torsion, and the maximum relative error of inherent frequency is 1.33%. Therefore, the analysis data of finite element mode were reliable.

3.3 Optimization design of chassis structure

The transplanter is equipped with an air-cooled 4-stroke 2-cylinder OHV gasoline engine, and the crankshaft turns 180°.

The impact of the excitation frequency generated by the field pavement on the dynamic characteristics of the chassis is correlated with the running speed and road roughness. When the running speed of the transplanter ranges from 20 km/h to 30 km/h, the calculated road excitation frequency is 10-30 Hz. However, the first-order inherent frequency of the chassis is 23.796 Hz, which exactly lies between 10-30 Hz. Therefore, when the transplanter completes the high-speed transplanting, it may cause resonance of the frame, thus affecting the quality of the seedling transplanting. If the chassis is designed for vibration reduction, the overall inherent frequency of the chassis needs to be increased, enabling the first-order inherent frequency to get farther away from the excitation frequency of road roughness.

The comparison of finite element mode analysis and experimental mode analysis results of the transplanter chassis showed that the modal results of the first four orders are similar. The modal vibration mode was dominated by the overall torsion and the inherent frequencies were not all the same. The possible reason may be that the obtained results were different from the actual model after the model was simplified when the finite element analysis is conducted. During the modal test, the knocking direction of the impulse hammer is not parallel to the sensor axis, thus generating test errors. The materials were sticky to the hammer while being knocked, which generate excitation signals that are not pulse signals.

Based on the above analysis, in order to keep the inherent frequency of the chassis away from the range of road excitation frequency, a model was established on the premise of slightly changing the overall weight of the chassis for structure optimization. The thicknesses of the longitudinal and transverse beams of the chassis were taken as design variables, and their inherent frequency was optimized. The relevant variables selected for the optimization design of the chassis and the optimization results are shown in Table 4.

Table 4 Values of variables before and after optimization

No.	Variable name	Before optimization	After optimization	Increasing amount	Rate of change%
1	Total volume/m^3	0.09532	0.105	0.00968	9.2
2	Total mass/kg	75.99	82.704	6.714	8.8
3	1st order frequency/Hz	23.68	32.083	8.403	35.5
4	2nd order frequency/Hz	42.64	33.751	−8.889	20.8
5	3rd order frequency/Hz	50.83	42.517	−8.313	16.4
6	4th order frequency/Hz	58.46	50.362	−8.098	13.9
7	Stringer 1/mm	3	3.0	0	0
8	Crossbeam 1/mm	2	2.5	0.5	25.0
9	Crossbeam 2/mm	2	2.5	0.5	25.0
10	Crossbeam 3/mm	3	2.5	−0.5	16.7
11	Crossbeam 4/mm	3	3.0	0	0
12	Crossbeam 5/mm	3	3.0	0	0
13	Round steel (12 pcs)/mm	2	2.5	0.5	25.0

From the optimization results, it can be seen that after the total mass of the chassis increases by 6.714 kg, the modal frequencies of the first four orders are 32.083 Hz, 33.751 Hz, 42.517 Hz, and 50.362 Hz, respectively. In particular, the first-order frequency increases to 32.083, which broadens its interval with the range of road excitation frequency (10-30 Hz). In this way, the possible resonance can be avoided when the transplanter works. The optimization effect for the chassis of the rice transplanter was good, and it can meet the needs of vibration and noise reduction in actual operations. The following figure can more intuitively show the situation before and after optimization of each indicator.

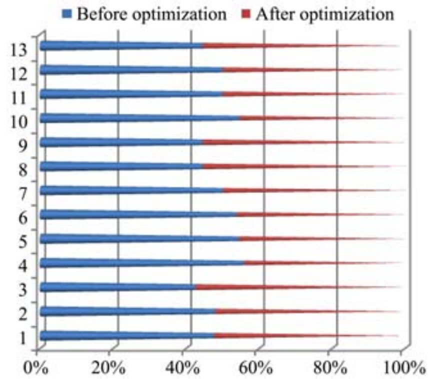


Figure 12 Comparison chart before and after optimization

4 Conclusions

1) Pro/Engineer 5.0 was used to build a model for the transplanter chassis. ANSYS Workbench was used to solve the modal frequencies and vibration modes of the first four orders of the chassis. A modal test was carried out based on MAC, and the test results contrasted with the theoretical calculation, verifying the accuracy of the theoretical analysis of finite elements.

2) The relationship between the inherent frequency of the chassis and the road excitation frequency was analyzed through comparison. The frequencies of the first four orders are 23.796 Hz, 42.428 Hz, 50.838 Hz, and 58.635 Hz, respectively. The first-order frequency exactly lies in the range of the road excitation frequency. The data show that the resonance of chassis is quite obvious when the rice transplanter works.

3) Relevant variables for the optimization design of the chassis and the optimization results are shown in Table 3. It can be seen that after the total mass of the chassis increases by 6.714 kg (8.8%), the modal frequencies of the first four orders are 32.083 Hz, 33.751 Hz, 42.517 Hz, and 50.362 Hz, respectively. In particular, the first-order frequency increases to 32.083, which broadens its interval with the range of road excitation frequency (10-30 Hz). In this way, the possible resonance can be avoided when the rice transplanter works.

Acknowledgements

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