Vibration Analysis of Corn Header Silage Module Rack Based on Technological System Evolution

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Abstract. The silage module of the corn header is transformed from the existing silage corn header. The tractor drives the silage module to harvest silage corn, and the performance of the silage module directly affects the quality of the harvest. In view of the strong vibration and unstable performance of the silage module during operation, it is necessary to perform vibration analysis on the frame of the silage module. The 3D model of the silage module was established by Solidworks. The modal analysis of the frame structure of the silage module was carried out by ANSYS. The first 6 modes of the frame and its natural frequency are obtained. By analyzing the intrinsic relationship between the mode and vibration of the silage module frame, the first-order mode of the silage module frame is obtained to resonate with the road surface excitation frequency. Using the technical system evolution rule to analyze the patents related to avoiding resonance methods, it is concluded that the current patent evolution is at a mature stage. It shows that there are enough ways to avoid resonance. Combined with the scientific effects of the How-to model and related patent cases, resonance is avoided by changing the structure of the frame to avoid the excitation frequency of the road surface.

1. Introduction
The quantitative and qualitative analysis of the relevant patents on corn headers was carried out using the technical evolution rule of TRIZ theory. It is concluded that the current corn harvesting platform is in a period of decline in the technological system. During this period, the super system evolution rule was applied to the innovative design of the corn header. A corn header that harvests grain corn and harvests silage corn is obtained. The corn header is a backpack-type header, which is mainly composed of a silage module (Figure 1) and a picking module (Figure 2). The tractor drives the silage module to harvest the silage corn. This working state is called state one, as shown in Figure 1. After the picking module is installed on the silage module, the formed corn cutting head can carry out the operation of the stem and ear of the grain corn under the tractor. This working state is called state two, as shown in Figure 3. In order to achieve the matching and disassembly of the two modules, the silage module applied in the state one is transformed from the existing silage corn header.
Figure 1. Silage module.

Figure 2. Picking module.

Figure 3. Corn header.

The state-silage module is motivated in many ways during field work, which may cause undesirable vibration of the silage module. Poor vibration affects the reliability of silage module operation. This not only affects harvesting efficiency, but also causes mechanical fatigue failure. Therefore, the identification of the natural frequency and mode shape determined by the mass and stiffness distribution in the silage module structure is of great significance for guiding the transformation of the silage module frame [1]. The modal analysis method mainly studies the vibrational shape and resonance properties of the structure at the natural frequency [2]. And it can acquire the modal parameters of the structure, namely the natural frequency, mode shape, and damping ratio [3]. Modal analysis provides a direct and efficient method for studying the vibration characteristics of structures. Many domestic scholars use modal analysis methods to find the weak links of the structure in the process of studying mechanical vibration, and optimize it [4-6]. Based on a corn header, this study calculates the natural frequency and mode shape of the silage module frame by ANSYS, and analyzes the influence of external excitation frequency on the header vibration. The header frame is further optimized by combining the patent cases and the knowledge base to avoid resonance.
2. Research route

Through the Patent Search website of the State Intellectual Property Office, relevant patents for “avoiding resonance methods” were found. After screening the patents obtained, a total of 32 related patents from 2007 to 2009 were obtained. According to the difference of the original understanding level, 32 related patents were qualitatively analyzed, and the patent time and grade statistics were obtained, as shown in Table 1. According to the change in the number of patents in Table 1 over time, the trend of the number of patents related to the “avoid resonance method” from 2007 to 2019 is shown in Fig. 4. According to the change of the patent grade in Table 1, the trend of the patent-grade trend of the “avoid resonance method” from 2007 to 2019 is shown in Fig. 5. In Figure 4, the curve shows a slow upward trend in 2019, and the curve in Figure 5 shows a downward trend in 2019. From this it can be concluded that the “avoid resonance method” is at the end of maturity. The patent library and scientific effects can be applied to solve the resonance problem according to the actual situation.

3. Analysis of patents

Through the Patent Search website of the State Intellectual Property Office, relevant patents for “avoiding resonance methods” were found. After screening the patents obtained, a total of 32 related patents from 2007 to 2009 were obtained. According to the difference of the original understanding level, 32 related patents were qualitatively analyzed, and the patent time and grade statistics were obtained, as shown in Table 1. According to the change in the number of patents in Table 1 over time, the trend of the number of patents related to the “avoid resonance method” from 2007 to 2019 is shown in Fig. 4. According to the change of the patent grade in Table 1, the trend of the patent-grade trend of the “avoid resonance method” from 2007 to 2019 is shown in Fig. 5. In Figure 4, the curve shows a slow upward trend in 2019, and the curve in Figure 5 shows a downward trend in 2019. From this it can be concluded that the “avoid resonance method” is at the end of maturity. The patent library and scientific effects can be applied to solve the resonance problem according to the actual situation.

Table 1. 2007-2019 “Avoiding Resonance Method” related patent time and rank statistics.

| Year | Number | Rank |
|------|--------|------|
| 2007 | 2      | 1.33 |
| 2008 | 1      | 3    |
| 2009 | 0      | 0    |
| 2010 | 1      | 2    |
| 2011 | 2      | 4    |
| 2012 | 3      | 3    |
| 2013 | 0      | 0    |
| 2014 | 1      | 2    |
| 2015 | 6      | 3.33 |
| 2016 | 4      | 2.67 |
| 2017 | 1      | 2    |
| 2018 | 7      | 1.86 |
| 2019 | 4      | 1    |
Three ways to avoid resonance are obtained by qualitative analysis of patents: (1) For an object with a constant natural frequency, the excitation frequency is avoided by the natural frequency of the object to avoid resonance. (2) For an object whose natural frequency changes, the excitation frequency is monitored in real time, and the excitation frequency is suppressed, so that it is difficult for the excitation frequency to reach the natural frequency to avoid resonance. (3) The excitation frequency is monitored in real time for objects whose natural frequencies have changed. When the probability frequency is about to reach the natural frequency of the object, the excitation frequency is rapidly increased (or lowered) to skip the natural frequency to avoid resonance.

4. Definition problem

4.1. Theoretical basis of modal analysis

Modal analysis focuses on the vibration characteristics of structures and machine components. Modal analysis requires knowledge of the boundary conditions, geometry, and material properties of the structure. The mass distribution, stiffness distribution and damping distribution of the structure are expressed by mass matrix, stiffness matrix and damping matrix respectively. These data are used to determine the modal parameters of the system and to represent the mechanical characteristics of the system [8]. Object Multidimensional Differential Equation:

\[ [M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F(t)\} \tag{1} \]

In the formula: \([M]\) is the mass matrix; \([C]\) is the damping matrix; \([K]\) is the stiffness matrix; \([F(t)]\) is the force matrix; \([u]\) is the displacement matrix; \([\dot{u}\}\) is the velocity matrix; \([\ddot{u}\}\) is the acceleration matrix.

When calculating the natural frequency of the header frame, an undamped free vibration differential equation is used, \([M]\{\ddot{u}\} + [K]\{\dot{u}\} = 0\). Solving this differential equation results in \(([K]-\omega_i^2[M])\{\Phi_i\} = 0\). In the formula, \(\omega_i\) is the n-order natural frequency of the structure, and \([\Phi_i]\) is the mode vector of the structure corresponding to \(\omega_i\).
4.2. Finite element modal analysis
In the finite element modal analysis of silage modules, it is mainly for the rack. A boundary condition is applied to displacements of three directions of x, y, and z. Considering the constraints of the two lifting rods and one telescopic rod of the fixed frame, three fixed constraints are added here. Using a tetrahedral mesh, the smallest unit of the mesh is a 30mm refined mesh. Combined with the actual structure and load conditions, the sixth-order mode is selected to meet the requirements of modal analysis [10].

4.3. Finite element modeling
The finite element model is the basis of modal analysis. Reasonable simplification is pointed out in many structural modal analyses [10]. Small threaded corner fillets are not considered. The ribs and reinforcing triangles on the entire silage module frame are retained. The type of contact between the members of the panel is fixedly connected. The material type of the model is Q235A. The modulus of elasticity is 211GPa. The Poisson's ratio is 0.35. The density is 7.85g/cm³. The finite element model of the silage module frame is shown in Figure 6.

The boundary condition of the silage module frame is to fix three hinge axes. The hinged shaft is hinged to the tractor hinge, as shown in Figure 7.

4.4. Meshing of silage module racks
Meshing is the most important step in modal analysis. Meshing takes a tetrahedral element. In order to control the quality of the meshing and improve the accuracy of the meshing, the smallest unit size is
30 mm. In order to pass the grid quality check standard [11], the coefficient of application ANSYS control unit quality is close to 1. The scale factor of Jacques is less than 5. The adjustment of the two main parameters is conducive to improving the quality of the grid. The number of units in the final finite element model is 46,829, and the number of nodes is 127,236.

4.5. Rack modal analysis results
As can be seen from Table 2, the first-order vibration is the vertical bending vibration of the front end of the frame. The second-order vibration is the left and right bending vibration of the frame. The third and sixth-order vibrations are bending torsional vibrations on both sides of the frame. The fourth-order vibration is the tension vibration of the front end of the frame. The fifth-order vibration is the left right bending vibration of the speed reducer fixing plate and the tension vibration of the front end. The specific mode shape is shown in Figure 8.

| Order   | Frequency (Hz) | Largest variable (mm) | Maximum deformation                       |
|---------|----------------|------------------------|-------------------------------------------|
| First order | 37.82          | 9.46                   | Intermediate divider front end            |
| Second order | 69.03          | 9.04                   | Reducer fixing plate front end            |
| Third order  | 75.15          | 8.39                   | Right side divider fixed plate front end  |
| Fourth order | 78.74          | 8.85                   | Intermediate divider front end            |
| Fifth order  | 81.39          | 32.59                  | Reducer fixing plate front end            |
| Sixth order  | 111.13         | 7.86                   | Right upper end of the upper beam         |

**Figure 8. Modal modes of each order.**
4.6. Modal analysis of silage module rack

The dynamic design of the silage module frame requires the inherent frequency of the structure to avoid external excitation. The spindle speed of the pendulum ring mechanism is 500-600r/min, and the excitation frequency of the corresponding cutter mechanism to the header frame is 8.3-10Hz. The reel has a rotational speed of 40-60r/min, and the corresponding excitation frequency is low. The effect of the engine on the header is mainly based on the burst excitation frequency \[12\]. The calculation formula is \( f_1 = \frac{2n1z}{60\tau} \). (2)

Formula: \( f_1 \) is the excitation frequency, Hz; \( z \) is the number of engine cylinders, (constant) 4. This study used an in-line four-cylinder four-stroke diesel engine. The rated speed is 2400r/min. The excitation frequency at the time of normal engine operation was calculated to be 80Hz. So the low-order idle speed will cause the header to vibrate.

The load of the road surface is calculated by the harvester speed of 3.5~40km/h. The road surface excitation frequency is \( f = \frac{v}{(3.6n)} \). (3)

In the formula: \( f \) is the excitation frequency of the road surface, Hz; \( v \) is the forward speed of the machine, km/h; \( n \) is the wavelength of the road roughness. This study took 0.32m. Therefore, the excitation frequency for calculating the road surface roughness is 3.04 to 34.72Hz.

The relationship between the critical speed and the frequency is \( n_3 = \frac{60f_3}{\tau} \). (4)

In the mechanical design, in order to ensure its safe operation and normal operation, the working speed \( n \) of each rotating shaft should be separated from a certain range of its critical speed.

The general requirement is that the axis of the working speed \( n \) is lower than its first-order critical speed, \( n < 0.75n_{c1} \); For an axis whose working speed is higher than its first-order critical speed, \( 1.4n_{ci} < n < 0.75n_{ci+1} \) is required. It can be seen from Table 2 that the working speed corresponding to the road surface excitation frequency of 34.72Hz at 40 km/h is \( n = 2083.2Hz \). It is greater than 0.75n_{ci}. Therefore, it will cause a strong resonance of the header.

### Table 3. Natural frequency and corresponding critical speed of the first 6 modes of the silage module frame.

| Modal order | Natural frequency (Hz) | Corresponding critical speed (r/min) |
|-------------|------------------------|-------------------------------------|
| First order | 37.82                  | 2269.2                              |
| Second order| 69.03                  | 4141.8                              |
| Third order | 75.15                  | 4509.0                              |
| Fourth order| 78.74                  | 4724.4                              |
| Fifth order | 81.39                  | 4883.4                              |
| Sixth order | 111.13                 | 6667.8                              |

This corn harvester header has the following four speeds: The main drive shaft speed is 600r/min. The disc cutter speed is 52r/min. The cutting knife speed was 600r/min. The feed knife speed is 150r/min. It can be seen that the rotational speed existing in the header and the critical speed of the first-order mode of the header are greatly different. Therefore the system does not resonate.

5. Solution

Avoid resonance as a general functional analysis problem. So such problems are solved by using the How to model.

The established How to model: How to stabilize the working status of the corn header?

Implemented features: F24 Form the required structure, Stabilize the structure of the object.

Scientific effects and phenomena recommended by TRIZ: Resonance.

The frame structure is ribbed to increase the vibration frequency.

The patented frame composite seismic structure (application number: 201420458366.9) is referred to. The corn header frame is treated as follows. In order to improve the first-order vibration frequency
of the upper and lower bending vibrations of the front end of the rack in the first-order mode, the triangular block is added and the front end quality of the rack is reduced. Modeling is shown in Figure 9.

Figure 9. Finite element model of the optimized silage module frame.

A finite element analysis of the modified silage module rack yields: The first-order vibration is the up-and-down bending vibration of the front end of the frame. The frequency is $49.372\text{Hz}$. The corresponding critical speed is $2962.3\text{r/min}$. Therefore, the operating speed corresponding to the road surface excitation frequency of $34.72\text{Hz}$ at $40\text{ km/h}$ is $n=2083.2\text{Hz}$. It is less than $0.75n_{c1}$. Therefore, it does not cause strong resonance of the silage module.

6. Conclusion

Three methods commonly used to avoid resonance problems are summarized by analyzing related patents. This test is based on the ANSYS finite element modal analysis of the natural frequency and mode shape of the silage module frame. The research foundation was laid to optimize the vibration of the entire cutting platform. Comparing the natural frequency and the excitation frequency of the header, it can be seen that the rotational speed existing in the header has a large difference from the natural frequency of the sixth-order mode of the header and the corresponding critical speed. By calculating the excitation of the ground irregularity, it is known that the excitation of the road surface has a large vibration to the header. The first method is applied based on the evolution rule of the technical system. By changing the structure of the rack, the first-order frequency of the rack is increased. Thereby the excitation of the road surface is reduced. Strong resonances are avoided.

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