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The Design of the Knot Tying Disc outer Teeth

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Abstract: *Knotter is the core component of square baler, and its performance plays a decisive role in baling quality and machine reliability of square baler. In view of the low precision of the reverse model of the knotter gear disk and the high failure probability of the external teeth of the gear disk, it is a reliable guarantee to further improve the knotter quality to study and establish the forward design principle and method of the external teeth of the gear disk. In this paper, through the structural analysis of the knotter driving toothed disk, the phase distribution of external teeth, internal teeth and cams on the toothed disk is determined. Combined with the face gear design method, the parametric design of the external teeth of the gear disk is carried out, and the solid model of the external teeth of the gear disk is established by using MATLAB GUI module and SOLIDWORKS. The research has guiding significance and application value for improving the quality and efficiency of knotter design and knotter performance.*

Keywords: *knotter, tooth disc, face gear, forward design*

I. Introduction

Square straw bale machine has high production efficiency and high density of pressed straw bales, which plays an increasingly important role in reducing forage harvesting losses, lowering storage and transportation costs, and enhancing industrial competitiveness, and has strongly promoted the rapid development of China's grass and pasture industry and the comprehensive utilization of straw [1-2]. The knotter is the core component of the square bale machine, and its performance plays a decisive role in the baling quality and reliability of the square bale machine [3]. At present, the knotter mainly relies on imports, resulting in high purchase and maintenance costs [4]. The spatial structure of the knotter is complex, and the knotter is mainly modeled by reverse scanning technology in China, which lacks rigorous and mature design methods and theoretical guidance, resulting in low modeling accuracy and large processing errors, making it difficult to make breakthroughs in the performance of the knotter, and seriously restricting the localization level of the knotter.

There are various types of knotter failures and complex influencing factors, and an accurate grasp of the failure mechanism helps to deeply understand the structural characteristics of knotter, which is an important step to improve the reliability and life of knotter [5]. Baling practice shows that the failure of the tooth disc accounts for about 39% of the total failure of the knotter, and the location of the tooth disc failure is mainly concentrated in the outer teeth of the tooth disc, accounting for about 77% of the total failure of the tooth disc [6-7]. The inaccurate design of the outer teeth of the tooth disc and the wear of the outer teeth at work can lead to late entry of the teeth in the meshing process, changing the normal working timing of the knotter to the point that it cannot be properly meshed, which eventually leads to the failure of the knotter (Fig 1). Therefore, establishing the principle and method of tooth disc external tooth design and improving the accuracy of tooth disc external tooth model are of guiding significance and application value to improve the quality and efficiency of knotter design and knotter performance.

Fig.1 A diagram of damage to the outer teeth

II. Design parameters of the outer teeth of the tooth disc

2.1 Analysis of tooth disc structure

The tooth disc is the driving part of each moving part of the knotter. Through the analysis of RS3770 knotter tooth disc (Fig 2) produced by RASSPE (Germany), we know that the tooth disc mainly consists of two sections of incomplete teeth and cam track. Among them, the outer teeth are straight teeth, distributed in the outer edge of the tooth disc, the angle on the tooth disc is 45 °, control the knotting mouth movement, complete the winding buckle, clamp bite action, by the failure form analysis can be seen, the knotter tooth disc damage is mainly concentrated in the outer teeth damage; inner teeth are oblique teeth, distributed in the inner ring of the tooth disc, the angle on the tooth disc is 67 °, control the worm gear worm, so as to drive the movement of the clamping rope disc, complete the clamping rope action; cam track is distributed in the innermost circle of the knotter tooth disc, controlling the movement of the rope release lever to complete the rope cutting and release action. The phase difference between the inner and outer teeth is 58°, which determines the timing of the rope clamping action and the buckle winding action; the phase difference between the outer teeth and the far stop of the cam is 111°. The difference value determines the timing of the rope winding action and the rope cutting action.

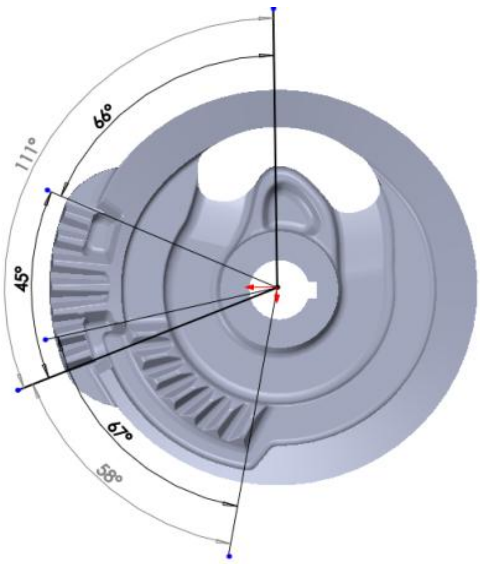

Fig.2 Phase diagram of the tooth structure outside the tooth plate

2.2 Determination of external tooth parameters

By reviewing the literature and the analysis of the knotter motion timing [8], it is known that in the actual operation, the speed of the tooth disc is 90r/min (540°/s) and the speed of the knotting nozzle bevel gear n2 is 600r/min, so the transmission ratio i between the tooth disc spindle and the knotting nozzle bevel gear is:

$$i = \frac{n_2}{n_1} = \frac{600}{90} = 6.67 \quad (1)$$

The knotted mouth bevel gear is an incomplete gear, through actual measurement, to take the ideal number of teeth of knotted mouth bevel gear z1 = 8, the big end of the tooth top circle diameter $d_{\varepsilon 1}$ is 38.2mm, so combined with the empirical formula can be obtained from the ideal number of teeth of the outer teeth of the tooth disc z2 and knotted mouth bevel gear modulus m1 respectively:

$$z_2 = i \times z_1 = 6.67 \times 8 \approx 54 \ (2)$$

$$m_1 = \frac{d_{\varepsilon 1}}{z_1+2} = \frac{38.2}{8+2} = 3.82 \ (3)$$

The outer teeth of the tooth disc are incomplete bevel gears without back cone, so the diameter of the large end of the outer teeth of the tooth disc is used as its indexing circle diameter to obtain the modulus of the large end of the outer teeth of the tooth disc. After measurement, the index circle diameter d_{a2} of the outer tooth of the tooth disc is 218mm, so the modulus m2 of the outer tooth of the tooth disc is:

$$m_2 = \frac{d_{a2}}{z_2} = \frac{218}{54} \approx 4.04 \ (4)$$

As the number of teeth of the knotted bevel gear is less than 13, the pressure angle should be larger to avoid the root cutting phenomenon, and the recommended pressure angle is 25° according to the Gleason standard. The basic parameters of the knotted spout bevel gear and the outer teeth of the tooth plate are detailed in Table 1.

Table.1 Tooth plate outer tooth parameters

Projects	Knotted Nozzle Bevel Gear	Outer teeth of gear plate
Theoretical number of teeth z	8	54
Modulus m	4	4
Pressure angle a	25	25

III. Design method of external teeth of gear disc

3.1 External tooth mathematical model

The analysis shows that the tooth shape of the outer teeth of the tooth disc is straight, so the current design on the outer teeth of the drive tooth disc mainly favors the conventional straight tooth design, but it fails to meet the working requirements and often fails prematurely. In this section, by referring to the literature related to the design of straight face gears and combining with the gear meshing principle, the tooth surface equations, transition surface equations, face gear root cut constraint conditions and face gear tooth top variable cusp constraint conditions of the outer teeth of the tooth disc are derived. The method of constructing the mathematical model of tooth disc external teeth is shown in Fig 3.

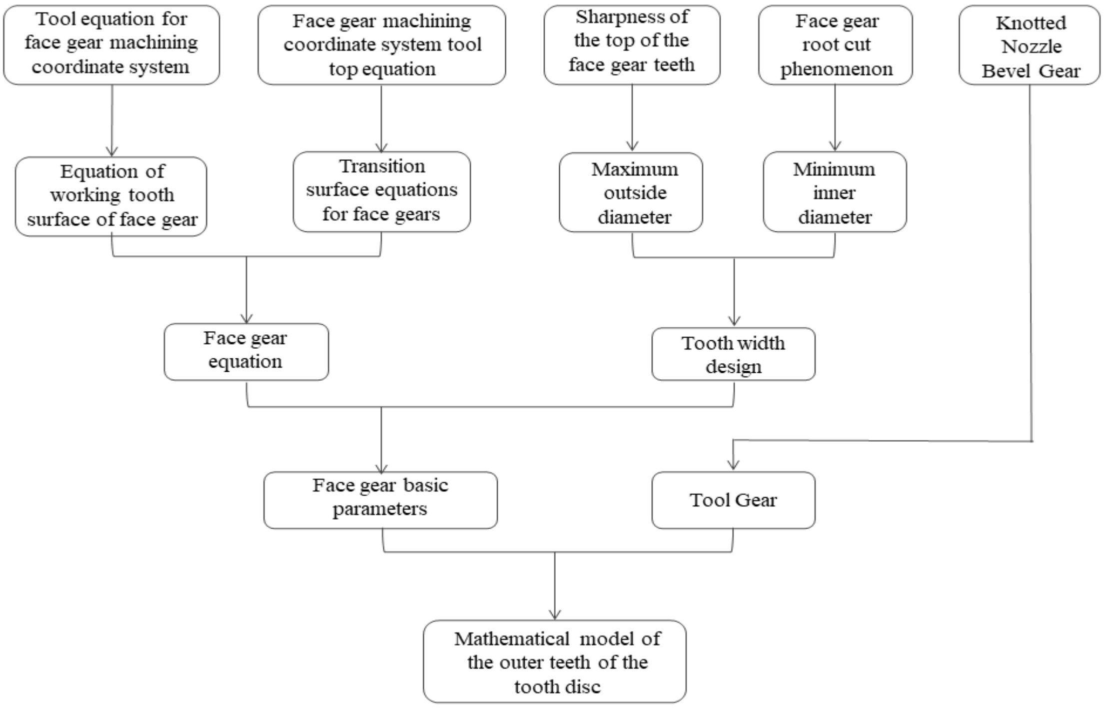

Fig.3 The mathematical model construction method of the outer teeth of the tooth disc

3.2 Non-orthogonal face gear design and processing principle

Face gears are difficult to process due to their complex tooth shape, so in practice they are mainly processed and manufactured by the method of inserting gears. The principle is to simulate the meshing process between the face gear and the tool gear, so that the tool gear completes the cutting motion, the circumferential feed motion of the tool gear, the splitting motion, the radial feed motion of the tool gear and the tool letting motion on the tooth blank to realize the processing and manufacturing; through the analysis of the face gear tooth top tip and tooth root root cutting phenomenon, the minimum inner diameter without root cutting and the maximum outer diameter without top tip occur; due to the processing and The number of teeth of the selected tool gear should be 1-3 teeth more than the number of teeth of the bevel gear meshing with the face gear due to the movement of the contact marks caused by machining and mounting errors [9]. For non-orthogonal face gears, the tool gear rotation axis should have an axis intersection angle with the face gear tooth blank rotation axis, and the schematic diagram of non-orthogonal face gear machining is shown in Fig4.

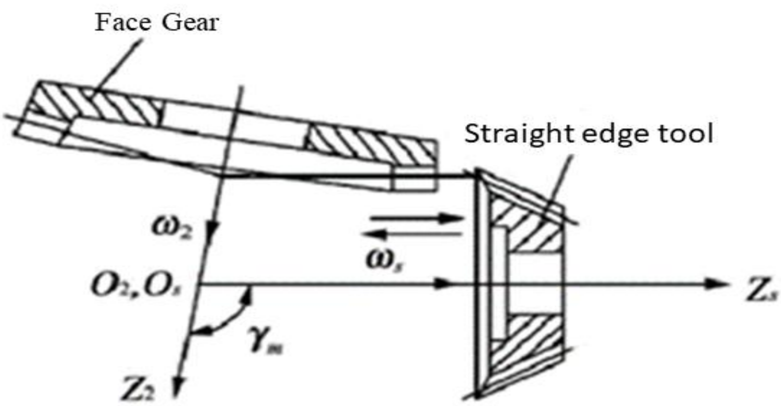

Fig.4 Non-orthogonal gear machining coordinate system

3.3 Mathematical model of non-orthogonal face gears

From the principle of face gear machining, it is known that the non-orthogonal face gear is derived from the tool gear involute equations and the machining coordinate system of the tool gear, and the non-orthogonal face gear tooth surface equations and transition surface equations are derived through coordinate conversion combined with the gear meshing principle. The equations of the non-orthogonal surface gear tooth surface are [10]:

$$\begin{cases} x_2 = r_{bs}\begin{pmatrix} A_\gamma \cos\varphi_2 - B_\gamma \cos\gamma \sin\varphi_2 \\ -C_\gamma \sin\varphi_2 \end{pmatrix} \\ y_2 = -r_{bs}\cos\gamma \begin{bmatrix} A_\gamma \sin\varphi_2 \\ +B_\gamma \cos\gamma \\ \times(\cos\varphi_2 + \tan^2\gamma) \\ +C_\gamma(\cos\varphi_2 - 1) \end{bmatrix} + r_{sm} \\ z_2 = r_{bs}\sin\gamma \begin{bmatrix} A_\gamma \sin\varphi_2 \\ +B_\gamma \cos\gamma(\cos\varphi_2 - 1) \\ +C_\gamma(\cos\varphi_2 + \cot^2\gamma) \\ +r_{sm}\cot\gamma \end{bmatrix} \end{cases} \quad (5)$$

In the equation:

$$\varphi_2 = q_{2s}\varphi_s \quad (6)$$

$$\begin{cases} A_\gamma = \sin\varphi_\theta \mp \theta_s \cos\varphi_s \\ B_\gamma = \cos\varphi_\theta \pm \theta_s \sin\varphi_s \\ C_\gamma = \frac{1-q_{2s}\cos\gamma}{q_{2s}\cos\varphi_\theta} \end{cases} \quad (7)$$

$$q_{2s} = \frac{N_S}{N_2} \quad (8)$$

$$\varphi_\theta = \varphi_s \pm (\theta_{s0} + \theta_s) \quad (9)$$

$$\theta_{s0} = \frac{\pi}{2N_s} - inv\alpha_0 \quad (10)$$

In the above equation：

φ_2-- Face gear corner；

φ_s-- Tool corner；

γ-- Axis corner；

θ_s-- Angle parameter at a point of tool involute；

r_{bs}-- Tool involute base circle radius；

q_{2s}-- Face gear to tool tooth ratio；

N_2-- Number of teeth of face gear

N_s-- Number of tool teeth；

θ_s-- The angle of a point on the involute of the tooth surface of the tool；

θ_{s0}-- The angle from the line of symmetry of the tool flutes to the start of the involute；

α_0-- Tool indexing circle pressure angle；

Inv-- Involute function；

r_{sm}-- Distance between face gear tooth top and tool.

The non-orthogonal face gear tooth root transition surface is based on the intersection of the tool gear tooth apex circle and the face gear tooth blank during meshing motion. The transition surface equation can be derived by making θ_s a constant in the tool tooth surface equation. The transition surface equation is as follows [10].

$$\begin{cases} r_{bs}\begin{pmatrix} A_\gamma \cos\varphi_2 - B_\gamma \cos\gamma \sin\varphi_2 \\ -C_\gamma \sin\gamma \sin\varphi_2 \end{pmatrix} \\ -r_{bs}\begin{pmatrix} A_\gamma \sin\varphi_2 + B_\gamma \cos\varphi_2 \\ +C_\gamma \cos\varphi_2 \end{pmatrix} \\ -r_{bs}\left(B_\gamma \sin\gamma - C_\gamma \cot\gamma\right) \end{cases} \quad (11)$$

When the internal diameter of the face gear is too small, a singularity with a radius of curvature of 0 will appear on the tooth surface, and thus the phenomenon of root cutting occurs; to avoid root cutting, the internal diameter of the face gear should be larger than this critical value. The minimum internal diameter constraint for the face gear not to undergo root cutting can be found by combining the meshing equations as [11].

$$R_1 = \sqrt{(x_2^*)^2 + \begin{pmatrix} -y_2^* \sin\gamma + z_2^* \cos\gamma \\ +r_{sm} \cot\gamma \end{pmatrix}^2} \quad (12)$$

The top tip of the tooth is characterized by the intersection of both faces of the face gear at the top of the tooth, i.e., the tooth thickness at this location is 0, i.e., the value of the transverse coordinate x at this location is 0. The maximum outer diameter constraint for the top invariant tip of the face gear tooth can be found by combining the meshing equations as [11].

$$R_2 = -\frac{y_2^*}{\sin\gamma} \quad (13)$$

IV. Parametric design steps for the external teeth of the gear disc

4.1 GUI interface and function design method

Enter the GUIDE command in the command line window in MATLAB platform to enter the GUI editing interface and open the wizard editor. Next, drag and drop four buttons in the appropriate positions in the controls provided, which are used for limit radius calculation, simulation calculation, exporting calculation results and clearing the screen, etc.; 10 static text boxes are filled with the names of parameters; 9 dynamic text boxes are filled with the values of parameters; and one coordinate axis is used to display the calculation design results. The result of the GUI interface design is shown in Fig 5.

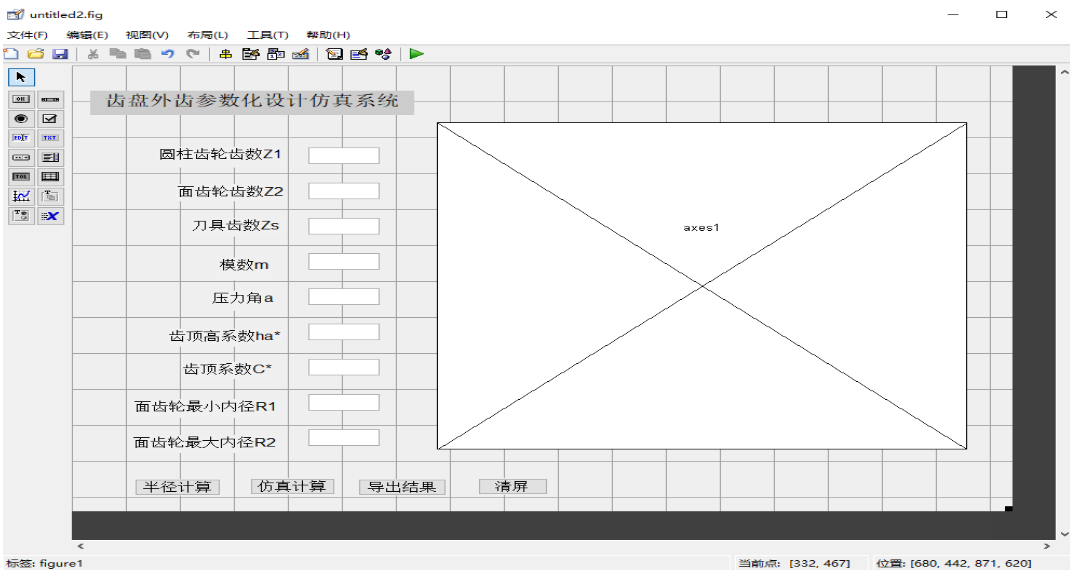

Fig.5 GUI interface

Write the callback function based on the working tooth equation, transition curve equation, maximum outer diameter and maximum inner diameter constraints of the non-orthogonal face gear. Firstly, the tool gear parameters are determined according to the knotted mouth bevel gear, and the limit radius of the outer tooth of the tooth disc is calculated by combining the parameters of the outer tooth of the tooth disc, and the values of the inner and outer diameters are determined according to the position of the tangent line of the knotted mouth bevel gear and the outer tooth of the tooth disc to complete the tooth width design; then the Z-directional section release modeling method is used, write the callback function based on the working tooth equation, transition curve equation, maximum outer diameter and maximum inner diameter constraints of the non-orthogonal face gear. Firstly, the tool gear parameters are determined according to the knotted mouth bevel gear, and the limit radius of the outer tooth of the tooth disc is calculated by combining the parameters of the outer tooth of the tooth disc, and the values of the inner and outer diameters are determined according to the position of the tangent line of the knotted mouth bevel gear and the outer tooth of the tooth disc to complete the tooth width design; then the Z-directional section release modeling method is used, that is, let $z_2 = c$, c is a constant, and the functional relationship between θ_s and φ_s is obtained $\varphi_s = f(\theta_s)$. Take the distance from the root to the top of the tooth as the value interval, take any Z_i value, find the intersection point of the tangent line between the working tooth surface and the transition surface and the corresponding radius Re; take the tooth width distance as the value interval, solve φ_s, θ_s cyclically in certain steps, when Rx≥ Re, substitute into the equation of the working tooth surface, otherwise substitute into the equation of the transition surface. Through continuous cyclic calculation, n sets of discrete points can be obtained, and the points are fitted to lines and lines are fitted to surfaces to obtain the tooth surface model of the outer tooth of the tooth disc [12]. The method of writing the callback function for the external tooth surface model of the tooth disc is shown in Fig 6.

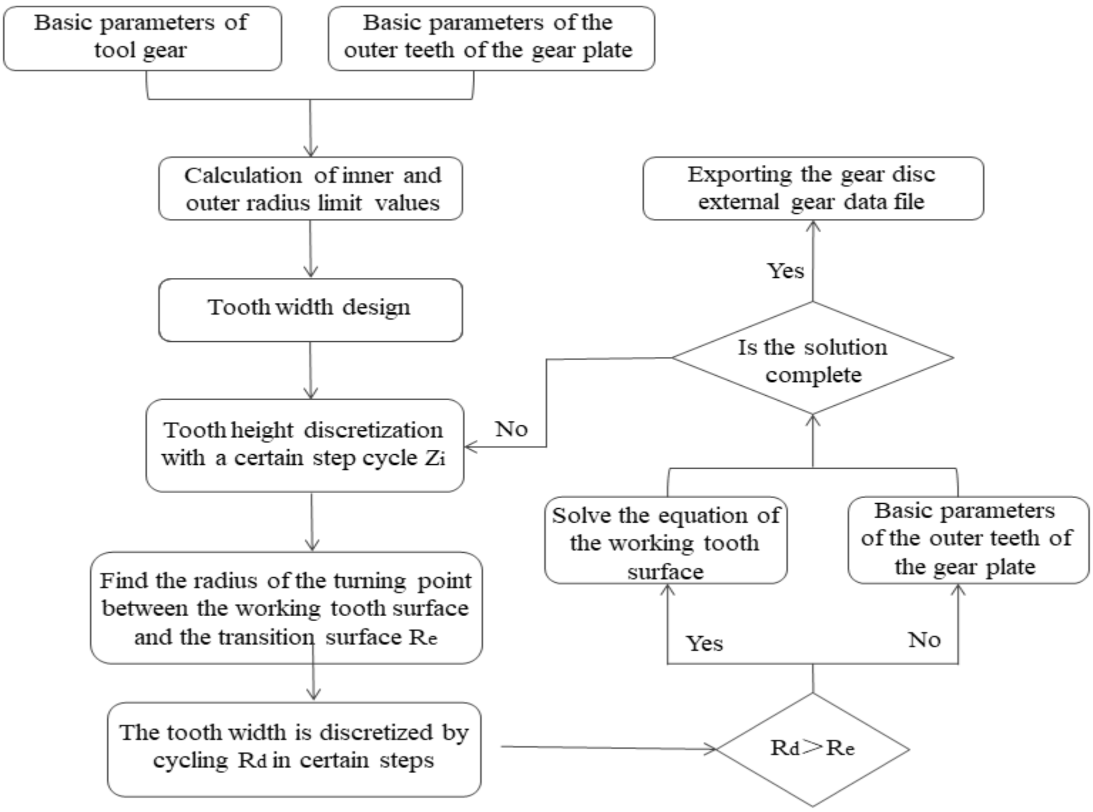

Fig.6 The design method of the flank model callback function

4.2 Simulation results

Enter the parameters calculated above in the GUI interface, click the Calculate Simulation button to run the program, and you can get the tooth surface model as shown in Fig 7.

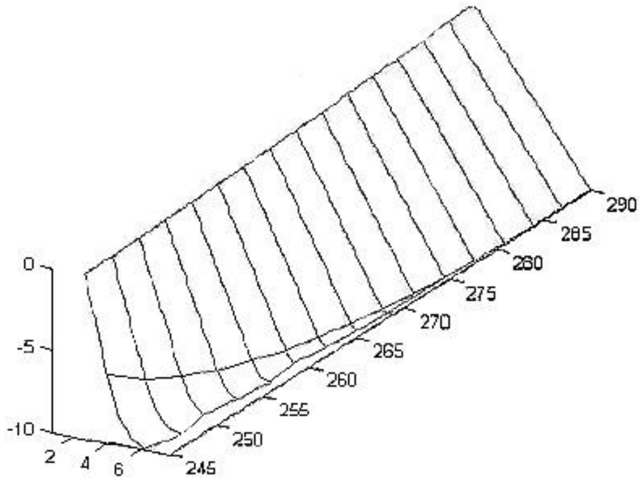

Fig.7 Teeth simulation results outside the tooth plate

4.3 Build 3D model

By calculating the discrete point coordinates of the outer tooth surface of the tooth disc in MATLAB, exporting these coordinate values and saving them as .ibl files, importing them into SOLIDWORKS, connecting the points into curves and forming surfaces from the curves to form a single tooth disc outer tooth model, obtaining seven outer tooth models by circumferential arrays, and combining these outer tooth models together. Combined with the structural parameters of the clamped rope incomplete bevel gear, the four tooth shapes of the outer edge are trimmed to obtain the 3D model shown in Figure 8.

Fig.8 Tooth plate outer tooth model

V. Conclusion

Aiming at the failure characteristics of the knotter, the structural analysis of the knotter drive tooth disc is carried out, and the phase distribution of the outer tooth, inner tooth and cam on the tooth disc is determined; the structural parameters of the knotter nozzle bevel gear are analyzed, and the structural parameters of the outer tooth of the knotter tooth disc are clarified by combining with its motion timing; the parameterized design of the outer tooth of the tooth disc is carried out by the face gear design method, and the three-dimensional model of the outer tooth of the tooth disc is constructed with the help of MATLAB GUI module and SOLIDWORKS is used to construct a 3D model of the outer tooth of the tooth disc. The thesis research provides a specific and reliable design method and procedure for the positive design of the knotter disc, which is conducive to promoting the localization process of the knotter.

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