Inverse design and accurate optimization of layered structured seeding mechanism for sugarcane planters

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Abstract: Existing sugarcane planters are difficult to have ideal seeding trajectory and motion attitude at the same time, and the speed is difficult to meet the requirements at the critical stage, resulting in poor stability, which ultimately makes it impossible to ensure that the sugarcane seeding is carried out in accordance with the agronomic requirements to ensure that the cane buds are oriented toward the wall of the seeding trench. Aiming at the second-order non-circular planetary gear system pendulum seeding mechanism of the planter, the paper innovatively adopts the combination of inverse design and multi-objective layered accurate optimization to solve the problems of attitude, speed and trajectory that do not meet the requirements of fixed-attitude seeding that still exists in the process of sugarcane seeding. The second-order non-circular planetary gear system is simplified into a three-rod two-degree-of-freedom mechanism, and the radius of the pitch curve of each non-circular gear is solved inversely by actively preplanning the static trajectory of the cane seed motion and analyzing the law of motion of the rod assembly. Determining the range of cane seed attitude angles in different motion phases as the first layer optimization objective, and fine-tuning the position of static trajectory key type value points to achieve the first layer optimization. Based on the non-circular gear pitch curve obtained from optimization, the interpolation points are marked on each non-circular gear pitch curve of the second-order non-circular planetary gear system, and based on the parameter optimization method of human-computer interaction, the radius values corresponding to the interpolation points of the non-circular gear pitch curve are fine-tuned to optimize the pitch curves, so as to satisfy the speed requirements of the cane species in each stage, and at the same time to make the convexity of non-circular gears in line with the principle of gear mesh, so as to complete the second layer of accurate optimization. The results of simulation verification show that the motion trajectory attitude of the virtual prototype is basically consistent with the theoretical model, which verifies the feasibility of the mechanism design. This study provides a new optimized design method for the cane seeding mechanism of sugarcane planters to achieve directional seeding.

Keywords: sugarcane planting, seeding mechanism, inverse design, layered structure optimization
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1 Introduction

Sugarcane sowing agronomy requires that the axis of the cane seed is parallel to the axis of the seed trench, and that the cane buds are oriented towards the trench walls on both sides of the seed trench for directional sowing. At present, the directional seeding of sugarcane is done manually by manually adjusting the direction of cane buds for directional seeding, which is labor-intensive and unable to meet the operation of sugarcane field. Sugarcane directional seeding machine requires the seeding mechanism to be able to accurately pick up the seed, stably transport the seed, and directionally drop the seed to ensure that the attitude of the seed after dropping the seed conforms to the agronomic requirements of sowing[1]. Currently, the domestic sugarcane seeding machine fails to satisfy the above stringent picking-up-transporting-dropping action requirements, and is unable to realize directional seeding.

Planar non-circular gears can realize the requirement of unequal speed transmission in the plane, meet the requirements of different motion trajectories and motion attitude, and are widely used in planting machinery[2-4]. In the late 1980s, Japanese researchers were the first to invent a rotary transplanting mechanism with a planetary gear train, which effectively improved the efficiency and success rate of potting seedling transplanting[5]. Wang et al.[6] designed a planetary bevel gear system transplanting mechanism based on spatial trajectory to realize the mechanized planting of rice potting in wide and narrow rows. Sun et al.[7,8] developed a non-circular gear and connecting rod combination potting transplanting mechanism to improve the success rate of transplanting. Xu et al.[9] utilized the labeled topological structure map in order to screen out the reasonable transplantation structure and solved the structural dimensions inversely on the basis of it, which effectively avoided the blindness of the design and improved the design accuracy. Yu et al.[10] structurally optimized the rice transplanting mechanism with incomplete eccentric circular gears and non-circular planetary gear train to obtain a reasonable

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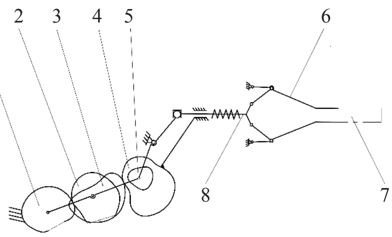
transplanting motion trajectory while realizing the intermittent motion characteristics of the transplanting mechanism. Chen et al.[11] optimized the trajectory attitude during transplanting and reduced the seedling injury rate by establishing a trajectory-targeted inverse model of transplanting mechanism parameters. Ye et al.[12] realized the beaked trajectory of potting seedling transplantation by improving the transmission characteristics of combined non-circular gears, but it is difficult to meet the demand of large seed spacing sowing of sugarcane. Mao et al.[13] developed a pneumatic pendulum transplanting mechanism to improve the efficiency and success rate of transplanting. Zhao et al.[14-16] first proposed the static trajectory transplantation mechanism reverse design method, which can obtain the ideal working trajectory and motion attitude, but there is a non-circular gear pitch curve curvature mutation, and its unequal transmission characteristics are difficult to meet the transplantation speed requirements of the transplantation arm. Liu et al.[17] established a cylindrical cam intermittent planetary gear system five-rod planting mechanism to form a compliant transplanting trajectory and realize zero-speed planting, but the cam-controlled planetary gear system mesh drive cannot make the speed of the end point of the transplanting arm to maintain a smooth continuity, so the whole machine is prone to produce rigid vibration. Zhou et al.[18] used fitted Bessel curves to realize the optimal design of non-circular gear pitch curves to improve the meshing transmission performance of second-order non-circular planetary gear trains, but it is difficult to generate flexible transplantation trajectories. Wu et al.[19] proposed a forward and reverse design method for planetary gear system rice potting plant transplanting mechanism, which combines forward design and reverse fine-tuning to obtain a relatively ideal trajectory. However, this design method does not have good forward optimization characteristics, and the reverse fine-tuning destroys the pitch curve of the forward design, which reduces the gear meshing transmission performance and affects the stability and success of the transplanting process. Yu et al.[3] optimized the seedling attitude and global speed at the key position of seedling transplantation based on the seedling picking mechanism of combined non-circular gear transmission, which effectively improved the success rate of seedling picking and transplanting. However, this optimization method leads to a large offset of the trajectory and attitude of the transplanting arm relative to the ideal state in the process of transporting seedlings, so it is not applicable to the agronomic requirements of stable seed transport in the process of sugarcane seeding.

Forward design[10,20] is to obtain the movement trajectory before seed drop through the structural dimensions of the already determined mechanism, which is easy to solve, but it requires design experience and insufficient optimization, and the work accuracy is low, which makes it difficult to ensure the success rate of seeding. Reverse design[11] is based on the ideal trajectory has been determined, through a reasonable allocation of the transmission ratio, solve the non-circular gear structure size, reverse design is easy to obtain the ideal trajectory, the work accuracy is better, but the curvature of the gear pitch curve obtained by the solution of the degree of curvature change is large, the gear mesh transmission performance deteriorates, resulting in difficult to control the seeding movement speed and stability.

In order to ensure that the sugarcane seeding has ideal motion speed, trajectory and attitude to meet the agronomic requirements of fixed attitude seeding, this paper adopts the second-order non-circular planetary gear train pendulum seeding mechanism, plans the seeding trajectory, and solves the equations of each non-circular gear pitch curve inversely. The non-circular gear pitch curves after the first layer of inverse optimization are obtained by fine-tuning the type value points on the trajectory to ensure a reasonable cane attitude at each stage. A second layer of forward optimization is carried out on the non-circular gear pitch curve after the first layer of inverse optimization, i.e., the interpolation points on the curve are micro-adjusted to achieve the requirement of cane seeding speed of the seeding mechanism at different stages without changing the general shape of the trajectory. Compared with the existing optimization design methods, the research method based on inverse design and multi-objective layered accurate optimization can obtain ideal seeding trajectory, cane seed attitude and reasonable non-circular gear pitch curve and meet the speed requirements at different stages of cane seeding motion.

2 Seeding trajectory and attitude requirements

In order to realize the requirements of precise seed picking, orderly seed transportation and fixed attitude seeding in the process of directional seeding of cane seeds, the motion trajectory of the seeding mechanism is divided into seed picking section, seed transportation section, seeding section and return section. The use of second-order non-circular planetary gear system pendulum seeding mechanism can realize a variety of complex movement trajectories in the plane[3,21,22], its structure sketch shown in Figure 1. The planetary gear train consists of sun gear 1, non-circular gear 2, non-circular gear 3, non-circular planetary gear 4. The planetary gear system is installed in the planetary carrier, the sun gear 1 is fixed in the frame, the non-circular gear 2 and the sun gear have teeth part of the meshing drive and coaxial transmission with the non-circular gear 3, the non-circular planetary gear 4 and non-circular gear 3 meshing drive and the seeding arm is fixed, so that the planetary non-circular gear 4 and the seeding arm has the consistency of movement. When the planetary carrier rotates clockwise during operation, the non-circular gear 2 rotates clockwise with respect to the planetary carrier, and the planetary non-circular gear 4 rotates counterclockwise with respect to the planetary carrier, so that the seeding clamping claw moves along the axis holding the seed according to a predetermined trajectory, and therefore the seed and the seeding arm have the same consistency of motion. Define ***a*** as the seed picking point, ***e*** as the seeding point, seed picking section ***ab***: the cane seed after the steering gripper precise orientation, in order to avoid being taken out of its end and the steering gripper interference, the planetary carrier every rotation of a circle, set 0.5 seconds interval, at this time, the steering gripper open, the cane seed is then taken out by the seeding gripper along the axis, so the change in the curvature of the arc of track arc in the seed picking section is relatively small. Seed transportation section ***bcd***: In order to realize the stable and orderly transportation of cane seeds, and to avoid interference with the conveyor belt of the seed dispenser, the trajectory should keep small curvature changes and the tangent line of the starting point of the seed transportation stage is close to perpendicular to the horizontal ground. Seeding section ***de***: In order to realize zero-speed seeding of cane seed[14,23], so that the seed will fall into the seed trench smoothly, the angle between the axis of the seed and the axis of the seed trench at the seeding point (i.e., the seeding angle) is as small as possible, and the horizontal component of the speed at this point is equal in magnitude to the speed of the locomotive and the direction is opposite. At this moment, the seeding jaws are open, one end of the cane seed touches the ground, and the other end will fall freely with a very small displacement. Return section ***efca***: The seeding jaws are kept open and the trajectory arc length is kept as small as possible without interference. According to the above movement requirements, establish the seeding arm tip point as shown in Figure 2 "8" type movement trajectory.



1. Sun gear 2,3. Intermediate non-circular gear 4. Planetary non-circular gear 5. Opening and closing connection cam 6. Seeding jaws 7. Cane seed 8. Seeding arm

Figure 1 Sketch of a pendulum seeding mechanism with second-order non-circular planetary gear train

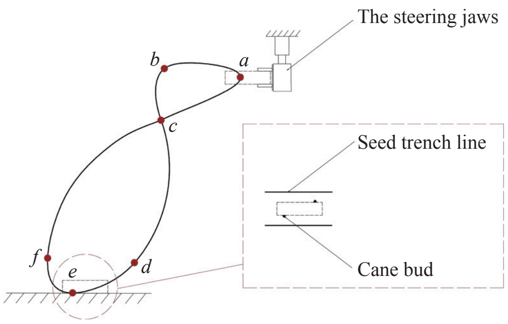


Figure 2 Seeding mechanism operation trajectory

The cane axis at point ***a*** is horizontal, the angle between the cane axis and the ground before seed drop at point ***e*** is small, and the angular displacement of the cane varies less within the trajectory of the seed picking section in order to improve the success rate of seed picking. In the ***bc*** section of the seed transport section, the angular displacement of the cane seeds varies less to ensure the stability of the seed transport and to prevent the attitude of the cane seeds from being difficult to adjust when they enter the seeding section. The ***cd*** section of the seed transport section needs to balance the angular displacement of the cane seed in the ***abc*** section, so that the angle between the axis of the cane seed at ***d*** and the horizontal line is as small as possible, in order to make the ***cd*** trajectory to meet the requirements of the planner, a non-meshing section is inserted between the first stage of the non-circular gears in order to realize that the seeding arm is stationary with respect to the planetary carrier in this motion interval. During the seeding section, the cane seeds maintain a small swing amplitude to increase the success rate of seeding. Since the seeding jaws in the return section are empty stroke, the swing angle attitude of the seeding arm may not be constrained.

According to the above movement trajectory planning requirements combined with the overall structural dimensions of the mechanism, it is necessary to constrain the key structural dimensions such as the small ring buckle composed of ***abc*** in the trajectory, the angle of the circle of the corresponding arc section of the seed transport section ***cd***, the large ring buckle composed of ***cdef***, the difference in height between the center of rotation of the seeding mechanism and the bottom of the seed trench, and the height of the seeding section ***de***. The trajectory planning parameters are listed in Table 1.

Table 1 Parameters of trajectory planning

Specific parameters	Value/mm
Height of small ring buckle	>130
Width of small ring buckle	(180,220)
Height of large ring buckle	>400
Width of large ring buckle	>250
cd corresponds to the angle of the center of the circle	>40
Difference in height of sowing section	(60,100)
Difference in height between the center of rotation of the mechanism and the bottom of the seed trench	>260

3 Inverse structural parameterization based on motion trajectories

3.1 Motion static trajectory curve expression

Obtaining the coordinates of any point on the planned trajectory is the key to calculating the structural parameters of the seeding mechanism. The local curvature of the curve can be adjusted arbitrarily by fitting a given point with a 3rd order non-uniform B-spline curve, which also ensures the second order continuity of the curve points[24]. The 27 points q_i ($i = 0, 1, \ldots 26$) are selected sequentially as the type-valued points on the curve, and the mathematical equations of the segmented curves are solved, which can be fitted to the ideal static trajectory curves (e.g., the preplanning trajectory ***t*** in Figure 3), so the coordinates of any point on the trajectory (x_t, y_t) are known quantities.

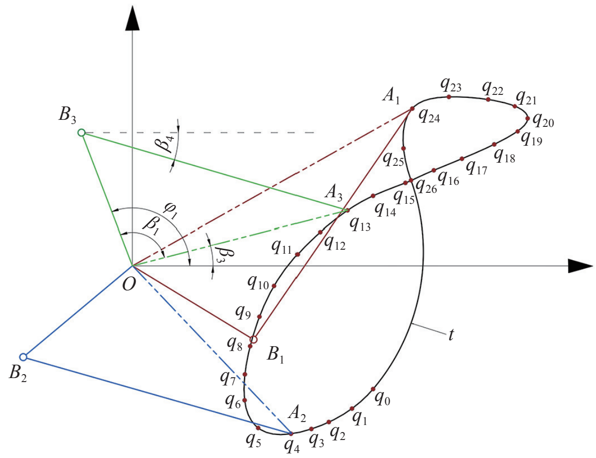


Figure 3 Simplified model of open chain triple rod mechanism

3.2 Seeding mechanism model simplification and motion analysis

Neglecting the mutual meshing relationship of the transmission gears, a reference coordinate system is established with the center of the sun gear as the coordinate origin, and the seeding mechanism is simplified as an open-chain three-bar two-degree-of-freedom model[15,25] (Figure 3). OB is the crank (planetary carrier), BA is the pendulum (seeding arm), and the point A is the tip of the seeding arm (A_1, A_2, and A_3 are the marking points where the point A is in different stages of motion), which is moving in accordance with the pre-planned trajectory. Thus the absolute motion of the planetary gear axis B in line with point A is a reciprocating pendulum, i.e., the absolute angular displacement of the planetary gear is a periodic function of an angle[26]. The length of OB is L_1 and the angle with the X-axis is φ_1 with an initial value of φ_{10}. The length of BA is L_2 and the angle with the horizontal plane is β_4 with an initial value of β_{40}. The angle between OB and OA is β_1 with an initial value of β_{10}. The angle between OA and the X-axis is β_3 with an initial value of β_{30}.

Crank OB length L_1 and pendulum BA length L_2 respectively:

$$\begin{cases} L_1 = \dfrac{\max\sqrt{(x_t^2+y_t^2)}-\min\sqrt{(x_t^2+y_t^2)}}{2} \\ L_2 = \dfrac{\max\sqrt{(x_t^2+y_t^2)}+\min\sqrt{(x_t^2+y_t^2)}}{2} \end{cases} \tag{1}$$

When the seeding arm tip A moves on a predetermined "8" trajectory, there is:

$$\beta_3 = \beta_{30} + \Delta\beta_3 = \arctan\frac{y_t}{x_t} \tag{2}$$

Obtained from the cosine theorem for triangles:

$$\beta_1 = \beta_{10} + \Delta\beta_{10} = \arccos\left[\frac{L_1^2+x_t^2+y_t^2-L_2^2}{2L_1\sqrt{x_t^2+y_t^2}}\right] \tag{3}$$

Since $\beta1\in(0,\pi)$, according to the law of motion of plane rods, $\varphi_1=\beta_1+\beta_3-2\pi$ when the seeding arm tip point is in the q_9-q_{20} return section; $\varphi_1=\beta_1+\beta_3$ when it is in the other stages. therefore, the coordinates of the articulation point B between the two rods are:

$$\begin{cases} x_B = L_1\cos\varphi_1 \\ y_B = L_1\sin\varphi_1 \end{cases}$$

Then the angle of deflection of the pendulum with respect to the horizontal plane is

$$\beta_4 = \beta_{40} + \Delta\beta_4 = \left|\arctan\frac{y_t-y_B}{x_t-x_B}\right| = \left|\arctan\frac{y_t-L_1\sin\varphi_1}{x_t-L_1\cos\varphi_1}\right| \tag{4}$$

3.3 Inverse second-order non-circular planetary gear train

Second-order non-circular planetary gear system can realize the crank uniform circumferential motion at the same time, the pendulum relative to the crank for unequal speed intermittent motion,[7] seeding mechanism motion diagram shown in Figure 4.

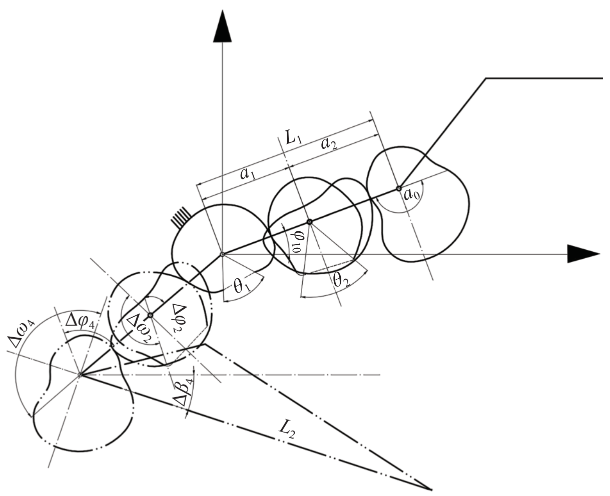


Figure 4 Sketch of the motion of the seeding mechanism

In order to obtain the planetary gear train pitch curve, a reasonable ratio allocation is made for the seeding mechanism[27,28]. Let the circumcentric angle corresponding to the toothless part of the sun gear be θ_1, then the circumcentric angle of the non-meshing region of the non-circular gear 2 is:

$$\theta_2 = 2\arctan\left(\frac{R_2\sin\dfrac{\theta_1}{2}}{a_1-R_2\cos\dfrac{\theta_1}{2}}\right) \tag{5}$$

where, R_2 is the edge radius of the toothless part of the sun gear.

When the first-order non-circular gears mesh and move to the non-toothed portion, the total planetary gear train ratio i_0=1, and the seeding arm is stationary with respect to the planetary carrier. When the first-order non-circular gears mesh to the toothed part, the seeding arm moves with unequal speeds relative to the planetary carrier. When the planetary carrier turns at an angle of $\Delta\varphi_1$, the non-circular planetary gear 4 turns at an angle relative to the planetary carrier:

$$\Delta\varphi_4 = |\Delta\beta_4 - \Delta\varphi_1| = \left|\arctan\frac{y_t-y_B}{x_t-x_B} - \beta_{40} - \Delta\varphi_1\right| \tag{6}$$

The ratio of the non-circular planetary gear to the planetary carrier $i_{41} = \dfrac{d\Delta\varphi_1}{d\Delta\beta_4}$ and the total ratio is:

$$i_0 = i_1 i_2 = \frac{d\Delta\varphi_1}{d\Delta\varphi_4} = \frac{d\Delta\varphi_1}{d(\Delta\varphi_1-\Delta\beta_4)} = \frac{i_{41}}{i_{41}-1} \tag{7}$$

where, i_1 is the primary transmission ratio formed by the sun gear 1 and the non-circular gear 2; i_2 is the secondary transmission ratio formed by the non-circular gear 3 and the planetary non-circular gear.

Set the intermediate non-circular gear 2 at this time relative to the planetary carrier angle of $\Delta\varphi_2$, in order to ensure that the pitch curve of the gear 2 is closed, according to the principle of non-circular gear mesh transmission, the sun gear rotates relative to the planetary carrier for one revolution, and the gear 2 rotates relative to the planetary carrier for one revolution[19], then there is:

$$\Delta\varphi_2 = \int_0^{2\pi}\frac{1}{i_1}\mathrm{d}\Delta\varphi_1 = 2\pi \tag{8}$$

In order to make the ratio curve of second-order non-circular gears approximate to the pitch curve, take the initial value of the ratio of first-order non-circular gears as: $i_{10} = \sqrt{\dfrac{1}{i_0}} = \sqrt{1-\dfrac{1}{i_{41}}}$, substitute it into Equation (8) for iterative calculations, and control the accuracy of 0.01, when meet the requirements of the calculation accuracy, the output of the exact value of i_1 and i_2, the first stage and the second stage of the non-circular gear center distance $a_1 = a_2 = \dfrac{L_1}{2}$. Then the pitch curve radius function of each gear in the second-order planetary gear system is:

$$\begin{cases} r_4 = \dfrac{a_2 i_2}{1+i_2},\ r_3 = a_2 - r_4 = a_2 - \dfrac{a_2 i_2}{1+i_2} \\ r_2 = \dfrac{a_1 i_1}{1+i_1},\ r_1 = a_1 - r_2 = a_1 - \dfrac{a_1 i_1}{1+i_1} \end{cases} \tag{9}$$

3.4 Reverse design results of the seeding mechanism

Considering the curvature control of the trajectory in section 2, combined with the height requirements of the trajectory and the degree of concavity and convexity of the non-circular gears of the inverse, the specific position of each type of value point is determined step by step in the process of adjustment, and a preliminary fit to meet the requirements of the size of the sowing mechanism of the movement of the trajectory, and set up the trajectory on the type of the coordinates of the value point as listed in Table 2 (the origin of the coordinates of the mechanism of the center of rotation). The 27 set type-valued points are fitted to form a 3rd-order non-uniform B-spline curve with trajectory circular segments cd corresponding to a 45° centroid angle. Extract the coordinates of the farthest and nearest points on the curve, and substitute them into Equation (1) to get the length of the planetary carrier L_1=254 and L_2=498, so the center distance of all levels in the planetary gear system is a_1=a_2=127.

In order to facilitate the calculation and check the motion law of each component, the initial coordinates of the cane seeding motion are set as the seeding point $q_{19}(x_{q19}, y_{q19})$ and the offset angle δ of the planetary carrier is constant at 180°, at this time, the planetary carrier is co-located with the seeding arm, β_{10} = 0°, β_{30} = β_{40} = 20.5°, the initial mounting angle of the planetary carrier φ_{10} = 20.5°, and the initial coordinates of the point B are (264, 89). The working process of the seeding mechanism and the relationship between the planetary carrier angle is shown in Figure 5 (the red solid line is the circle of the planetary carrier rotation cycle, and at the same time, it is regarded as the baseline of the seeding angular displacement of 0°, and the black solid line is the angular displacement of the seeding in the same cycle). According to the inverse calculation, the maximum angular displacement of the cane seed in the seed picking section ***ab*** with respect to the initial position is 37°; the angular displacement of the cane seed in the seed transporting section ***bc*** is –7.5°; at the end of the seed transporting section, the angular displacement of the cane seed reaches –15.5°; and the peak angular displacement of the cane seed in the sowing section is –20°, and the sowing angle of the cane seed is –35.5° .

Table 2 Setting coordinates of trajectory-type value points

Type value point	x	y	Type value point	x	y	Type value point	x	y
q_0	429.49	–220	q_9	227.24	–90.77	q_{18}	646.11	216.18
q_1	391.76	–254.9	q_{10}	253.27	–35.77	q_{19}	688.72	240.05
q_2	350.48	–279.09	q_{11}	294.84	20.57	q_{20}	705.00	264.00
q_3	319.07	–219.36	q_{12}	335.00	60.07	q_{21}	682.29	284.67
q_4	283.64	–300	q_{13}	383.69	98.25	q_{22}	635.41	296.36
q_5	224.99	–289.63	q_{14}	429.49	124.21	q_{23}	565.07	301.02
q_6	200.71	–240	q_{15}	487.32	147.32	q_{24}	500.00	280.81
q_7	201.22	–193.8	q_{16}	538.08	169.83	q_{25}	483.89	209.05
q_8	210.61	–143.46	q_{17}	588.37	190.51	q_{26}	497.28	152.00

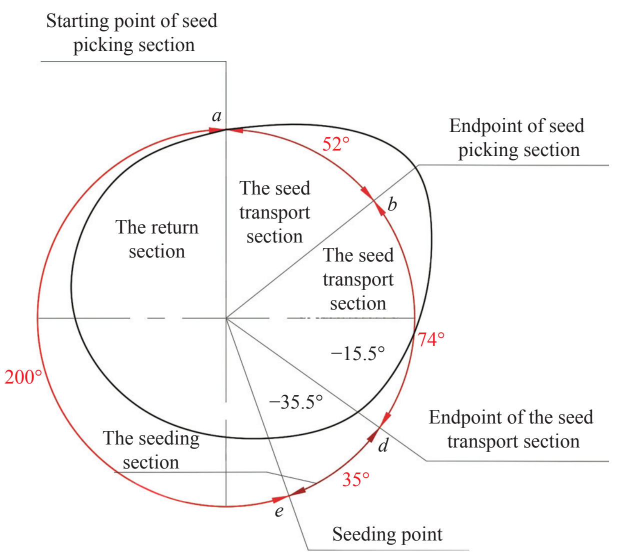


Figure 5 Working process of the seeding mechanism in relation to the angle of rotation of the planetary carrier

4 Multi-objective layered precision optimization of seeding mechanisms

4.1 Optimization Objectives and Optimization Approach

In order to meet the agronomic requirements of cane seed directional sowing, under the premise that the convexity of the pitch curve of the non-circular gear meets the requirements of meshing transmission, and constrain the attitude and speed of the cane seed in each stage[3,28], the optimization objectives are determined as follows:

1) The seed picking section ***ab*** cane angular displacement is less than 30° to avoid the end of the cane interfering with the steering jaws.

2) The angular displacement of the cane seed is less than 20° in the seeding section ***de*** to ensure seeding stability and success.

3) The angle (sowing angle) between the cane seed axis and the seed furrow axis at the sowing point ***e*** is less than 30° to ensure that the cane seed does not produce a large circumferential offset angle when it is dropped, and to realize precise fixed-attitude sowing.

4) In the sowing section ***de***, the speed of the sowing arm tip point A is lower than 1.5 m/s to improve the stability during sowing.

5) The absolute velocity of the seeding arm in the return section at the type value points q_4-q_8 corresponding to the dynamic trajectory (corresponding to the static trajectory ***ef*** section) is greater than 1.1 m/s, in order to avoid the large circumferential deflection displacement of the cane seed at the moment of detaching from the clamping jaws due to the continuous movement and attitude change of the clamping jaws when the seed is falling from the seeding jaws with the seed clamps open.

Optimization of a pendulum seeding mechanism with a second-order non-circular planetary gear train is a multi-parameter, multi-objective, strongly coupled and complex optimization[6]. This paper adopts the multi-objective hierarchical optimization method to meet the change requirements of cane seeding attitude in the optimization objective as the first layer of inverse optimization, and takes the predetermined motion trajectory and the relevant parameters of the seeding mechanism as the prerequisite to fine-tune the coordinates of the key value points on the trajectory one by one and obtain the non-circular gear pitch curves with the quadratic optimization characteristics after the optimization. Taking the requirement of satisfying the speed change in each stage of the optimization objective as the second layer of forward optimization, by fine-tuning the pitch curve radius of the non-circular gears after the first layer of inverse optimization, it can ensure that the motion trajectory and attitude of the cane seed can be maintained within a reasonable range at the same time, so that the motion speed of the cane seed can also satisfy the optimization objective in each stage.

4.2 Inverse optimization of the first layer of the seeding mechanism

To satisfy the first level of the inverse optimization objective, the positions of the type value points on the key trajectories are adjusted and corrected to progressively regulate the different attitudes of the cane species using the inverse optimization method in Section 4.1. The change in trajectory shape before and after inverse optimization (the optimized trajectory is denoted as a dashed line and the position of the seeding point is noted as ***g***) is shown in Figure 6. During the movement of the mechanism, the change in the swing angle of the seeding arm, the change in the angle of the planetary non-circular gear relative to the intermediate non-circular gear 3 $\Delta\beta_4$ and the change in the angular displacement of the cane are equal, and the change curve before and after the optimization is shown in Figure 7. After the optimization, the change curve of the angular displacement of the cane is more gentle, and the swing angle of the cane in the seed taking section is less than 30°, and the change in the angular displacement of the cane in the seed sowing section is smaller than that before the optimization and is less than 20°, and the seeding angle is –27.5°, which meets the optimization objective. The ratio curve of the non-circular planetary gear train after optimization is shown in Figure 8, after optimization, the total ratio of the planetary gear train changes are large, and the ratio changes are more obvious in the seed transport section and the seeding section, so the resulting non-circular gear pitch curve shown in Figure 9 has a quadratic forward optimization characteristics. The changes in attitude swing angle under different motion phases of cane species before and after reverse optimization are listed in Table 3.

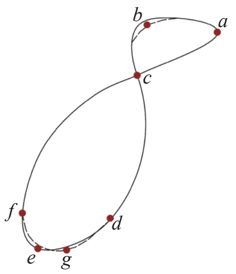


Figure 6 Trajectory shape change before and after reverse optimization

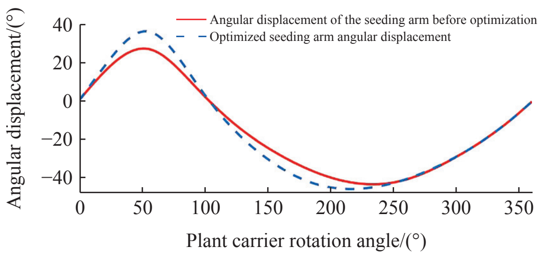


Figure 7 Variation in angular displacement of cane seeds

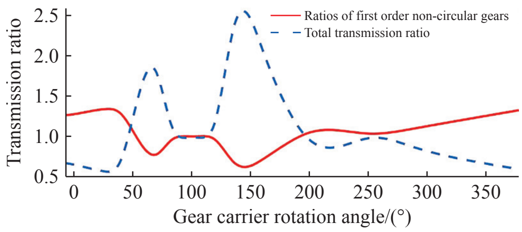


Figure 8 Variation of ratios of non-circular gear planetary gear train after reverse optimization

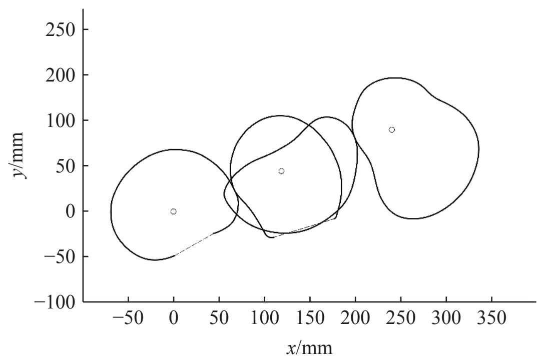


Figure 9 First layer of inverse optimized gear pitch curves

Table 3 Comparison of cane attitude changes before and after reverse optimization

parameters	Peak angular displacement of seed picking section/(°)	Peak angular displacement of seeding section/(°)	Sowing angle/(°)
before inverse optimization	37	20	–35.5
after inverse optimization	28	16	–27.5

4.3 Forward optimized motion analysis

Based on the equations of the pitch curves of each non-circular gear in the second-order non-circular planetary gear train after inverse optimization, interpolation points are set on the pitch curves of each non-circular gear in turn, and the optimization target is achieved by fine-tuning the interpolation points on the pitch curves of each gear. In order to improve the non-circular gear pitch curve adjustment range, so that the seeding mechanism has a stronger optimization performance, the use of Hermite interpolation method to fit the molding of the selected interpolation point, so that the optimization of the molding of the non-circular gear pitch curve transition is smoother, to improve the stability of the mechanism movement[29-31]. In order to make the fitted non-circular gear pitch curve closed and meet the requirements of meshing transmission, and to avoid the calculation of the order is too high and produce the curve of the sudden change of the “Runge” phenomenon, set n+1 = 8 interpolation points and each interpolation point radius is expressed in turn as $r_{1,i}$, $r_{2,i}$, $r_{3,i}$, $r_{4},_i$ $(i=0,1,2\ldots8)$.

Setting the initial position of the optimization mechanism as the seed picking point, when the planetary carrier rotates clockwise by an angle $\Delta\varphi_1$, then the non-circular gear 1 and the non-circular gear 2 rotate by an angle $\Delta\omega_2$ with respect to the sun gear:

$$\Delta\omega_2=\left|\frac{r_{2i}\Delta\varphi_1}{a_1-r_{2i}}-\Delta\varphi_1\right| \tag{10}$$

Then the angle $\Delta\varphi_4$ of the planetary uncircular gear relative to the planetary carrier is:

$$\Delta\varphi_4=\left|\Delta\omega_2\frac{r_{3i}}{a_2-r_{3i}}+\Delta\varphi_1\right| \tag{11}$$

At this point the equation for the rotation center displacement of the non-circular gear 1 is:

$$\begin{cases}x_{\mathrm{II}}=a_1\cos(\varphi_{10}-\Delta\varphi_1)\\ y_{\mathrm{II}}=a_1\sin(\varphi_{10}-\Delta\varphi_1)\end{cases} \tag{12}$$

Therefore, the displacement equation of the rotation center of the non-circular planetary gear is:

$$\begin{cases}x_{\mathrm{IV}}=x_{\mathrm{II}}+a_2\cos(\varphi_{10}-\Delta\varphi_1)\\ y_{\mathrm{IV}}=y_{\mathrm{II}}+a_2\sin(\varphi_{10}-\Delta\varphi_1)\end{cases} \tag{13}$$

Then the displacement equation for the seeding arm tip point A is:

$$\begin{cases}x_A=x_{\mathrm{IV}}+L_2\cos(\varphi_{10}-\Delta\varphi_1+\varphi_{40}+\Delta\varphi_4+\alpha_0)\\ y_A=y_{\mathrm{IV}}+L_2\sin(\varphi_{10}-\Delta\varphi_1+\varphi_{40}+\Delta\varphi_4+\alpha_0)\end{cases} \tag{14}$$

Where α_0 is the initial angle between the line from the center of the planetary gear axis to the tip of the seeding arm A and the line from the center of the planetary carrier.

From the cosine theorem of the triangle in the initial state:

$$\alpha_0=\arccos\frac{L_1^2+L_2^2-x_{q19}^2-y_{q19}^2}{2L_1L_2} \tag{15}$$

Then the velocity equation for the seeding arm tip point A can be obtained from Equation (14):

$$\begin{cases}\dot{x}_A=\dot{x}_{\mathrm{IV}}-L_2\sin(\varphi_{10}-\Delta\varphi_1+\varphi_{40}+\Delta\varphi_4+\alpha_0)(\Delta\dot{\varphi}_1+\Delta\dot{\varphi}_4)\\ \dot{y}_A=\dot{y}_{\mathrm{IV}}+L_2\cos(\varphi_{10}-\Delta\varphi_1+\varphi_{40}+\Delta\varphi_4+\alpha_0)(\Delta\dot{\varphi}_1+\Delta\dot{\varphi}_4)\end{cases} \tag{16}$$

Therefore, the combined velocity of the static trajectory of the seeding arm tip is $v_A=\sqrt{\dot{x}_A^2+\dot{y}_A^2}$, Let the seeding locomotive horizontal uniform velocity and the direction of the speed and the seeding point velocity of the horizontal component of the direction opposite to the size of v_c, then the seeding arm tip point of the combined velocity of the dynamic trajectory is:

$$v_{ae}=\sqrt{(\dot{x}_A-V_c)^2+\dot{y}_A^2} \tag{17}$$

For zero-speed seeding, then:

$$v_c=v_{A(G')}\cos\left[\frac{\pi}{2}-\Delta\beta_{4(G')}\right] \tag{18}$$

where, $v_{A(G')}$ is the combined velocity of the cane seeds at the seeding point and $\Delta\beta_{4(G')}$ is the seeding angle.

4.4 Analysis of parameter forward optimization results

A second-order non-circular planetary gear system pendulum seeding mechanism optimization platform is built[19,32-34] (Figure 10), and the eight interpolation points selected by the non-circular gears are fine-tuned, so that the seeding trajectory and attitude can meet the requirements on the premise of conforming to the optimization target of the speed that has been set at each stage.

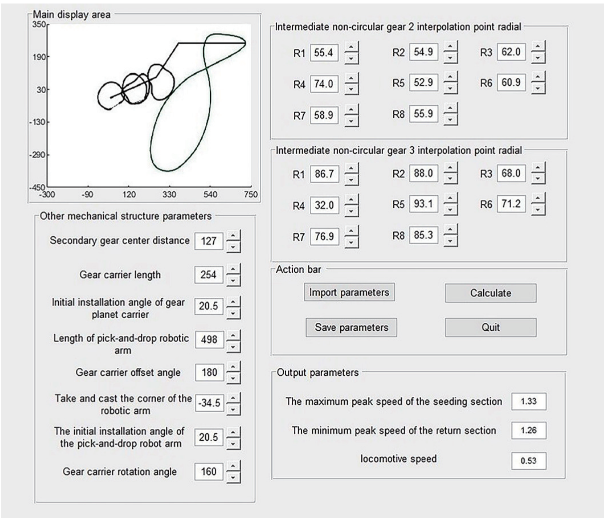


Figure 10 Optimization platform for pendulum seeding mechanism with second order non-circular planetary gear train

The results show that the intermediate non-circular gear's radial value has a greater influence on the combined speed of cane seed in the seeding section and the return section, and the change in the radial value of $r_{2,4}$ and $r_{3,3}$ corresponding to the interpolation point has a more obvious influence on the maximum speed of the seeding section; the influence results are shown in Figure 11. Interpolation point corresponding to $r_{2,5}$ and $r_{3,4}$ to the diameter value change on the ***ef*** section of the minimum peak speed of the influence is more obvious, the influence of the results shown in Figure 12. Fine-tuning the radius values of the two sets of interpolated points had no significant effect on the shape of either trajectory, and therefore had no significant effect on the attitude of the cane seed in the two target phases. The selected interpolation point for non-circular gear 2 has a slightly higher effect on the two-stage speed than non-circular gear 3. In order to avoid the large adjustment distance of the radius value of the selected interpolation point affecting the shape of the trajectory and the convexity of the intermediate non-circular gear, the radius value of the interpolation point $r_{2,4}$ of the intermediate non-circular gear 2 is first adjusted slightly, and the radius value of the interpolation point $r_{3,3}$ of the non-circular gear 3 is fine-tuned when the speed of the sowing section is close to the speed optimization target. Similarly, the radius value of the interpolation point $r_{2,5}$ of the non-circular gear 2 is slightly adjusted in priority, and when the minimum peak speed of the ***ef*** section is close to the optimization target, the radius value of the interpolation point $r_{3,4}$ selected for the non-circular gear 3 is fine-tuned.

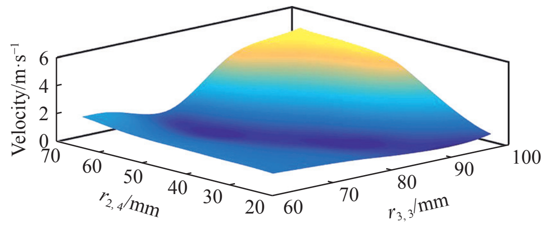


Figure 11 Effect of selected interpolation points of gears on the maximum speed of the seeding section

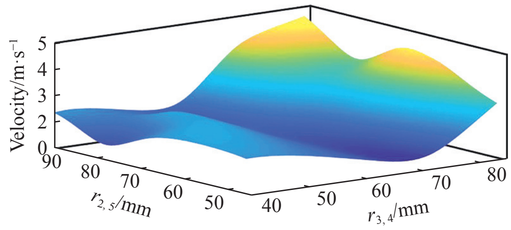


Figure 12 Influence of selected interpolation points of the gears on the minimum speed of the return section

Combining the optimization platform of the seeding mechanism with the non-circular gear convexity, transmission ratio, track structure size, operating attitude and other requirements, the global coordinate area of the trajectory is taken as the resolution area, and the optimal solution with the non-circular gear pitch curve in section 4.2 as the base condition is output[8,35], so that the radius of the pitch curve of the non-circular gear, the final optimized speed curve (Figure 13) and the pitch curve of the non-circular gear (Figure 14), shown in Table 4, are obtained. The results show that the maximum speed of the static trajectory of the sowing section is 1.33 m/s, the minimum speed of the ***ef*** section corresponding to the static trajectory is 1.14 m/s, so the minimum speed corresponding to the dynamic trajectory is 1.26 m/s, all of which satisfy the optimization objective.

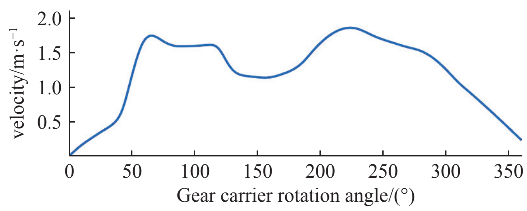


Figure 13 Variation curve of the magnitude of the optimized seeding arm tip closing velocity

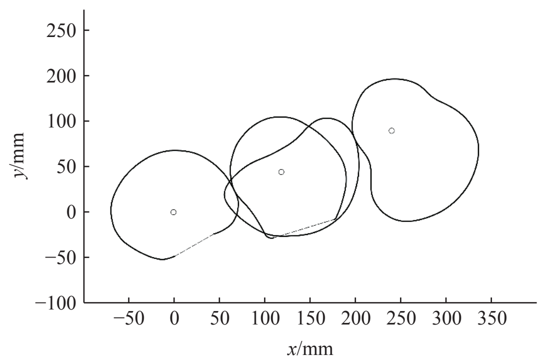


Figure 14 Optimized non-circular gear pitch curve

Table 4 Preferred values of interpolation points for intermediate non-circular gears

Parameters	Better value/mm	Parameters	Better value/mm
$r_{2,1}$	55.4	$r_{3,1}$	86.7
$r_{2,2}$	54.9	$r_{3,2}$	88.0
$r_{2,3}$	62.0	$r_{3,3}$	68.0
$r_{2,4}$	74.0	$r_{3,4}$	32.0
$r_{2,5}$	52.9	$r_{3,5}$	93.1
$r_{2,6}$	60.9	$r_{3,6}$	71.2
$r_{2,7}$	58.9	$r_{3,7}$	76.9
$r_{2,8}$	55.9	$r_{3,8}$	85.3

5 Simulation verification of seeding mechanism

According to the optimized design parameters of the seeding mechanism, add the motion constraints of each component based on Adams, and set the rotation speed of the planetary carrier to 360°/s, and the simulation time to 1 s, the simulation obtains the motion trajectory as shown in Figure 15, and based on the motion analysis of the planar rods in section 3.2, the attitude change of the cane seed is basically the same as that of the theoretical analysis results. The speed curve of simulation and theoretical analysis is shown in Figure 16. Therefore, the results prove that the key structure and motion parameters of the seeding mechanism are consistent with the theoretical analysis of optimization, indicating the correctness and reliability of the combination of reverse design and multi-objective quadratic accurate optimization.

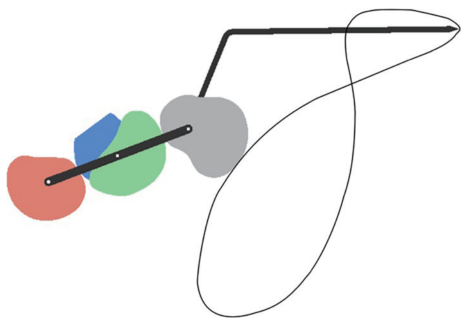


Figure 15 ADAMS virtual simulation trajectory of sugarcane seeding mechanism

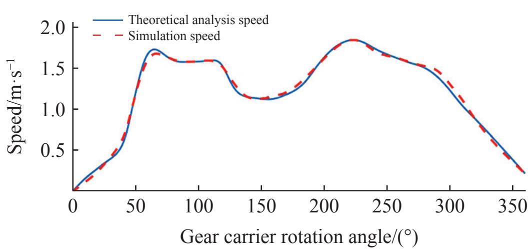


Figure 16 Speed analysis comparison

6 Conclusions

1) Aiming at the current sugarcane seeding process, it is difficult to simultaneously take into account the seeding trajectory, attitude, movement speed and stability to meet the agronomic requirements of the mechanism for orderly seed picking, stable seed transportation and directional seeding, the proposed method combines the inverse design of the sugarcane seeding mechanism with the multi-objective secondary accurate layered optimization, so as to make the advantages of the inverse design, the preliminary optimization and the forward accurate optimization complement each other. This method can avoid the blindness of forward design and the problem that it is difficult to meet the requirements of the convexity of the gear pitch curve in reverse design, and it is suitable for the mechanism that has complex requirements on trajectory attitude and speed. Reverse design and reverse preliminary optimization are used to obtain more ideal trajectory shape and cane seed motion attitude, and determine the non-circular gear pitch curve with forward optimization characteristics. The second forward optimization fine-tunes the interpolation points on the non-circular gear pitch curve so that the cane seed motion meets the speed optimization target, and at the same time avoids the variation of the non-circular gear pitch curve.

2) In order to make the seed movement trajectory, attitude and speed satisfy the optimization requirements at the same time, the inverse design kinematics model of the seed picking mechanism is established, and the non-circular gear pitch curve equations are obtained based on the inverse solution of the movement trajectory and the first layer of inverse optimization is carried out, and the optimization results are used as the basis of the second layer of forward optimization, and the final results are the seed picking section of the seed pendulum angular displacement is less than 28°, the seed sowing section of the seed pendulum angular displacement is less than 16°, and the seed sowing angle is –27.5°, the peak velocity of the seeding section is 1.33 m/s, and the minimum value of the velocity at the tip point of the seeding arm corresponding to the moving trajectory of the return section is 1.26 m/s, which all satisfy the optimization objectives. By solving for the magnitude of the locomotive's travel speed, the cane seed is simultaneously sown at the seeding point to satisfy the zero-speed sowing requirement.

3) Based on Adams, the second-order non-circular planetary gear system pendulum seeding mechanism is simulated, and the simulation results show that its static trajectory, mechanism motion parameters are basically consistent with the optimized theoretical static trajectory and related parameters, which verifies the correctness and reasonableness of the reverse inverse design combined with the quadratic accurate optimization method.

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