Journal of Agricultural Engineering 2022; volume LIII:1412

Optimal design of transplanting mechanism with differential internal engagement non-circular gear trains

Maile Zhou,[1,2] Jiajia Yang,[1] Tingbo Xu,[1] Jianjun Yin,[1,2] Xinzhong Wang[1]

[1]School of Agricultural Engineering; and [2]Key Laboratory of Modern Agricultural Equipment and Technology, Jiangsu University, Zhenjiang, China

Abstract

This study aimed at the problems of unequal speed transmission ratio mutual restriction and side gap accumulation of the transplanting mechanism with single-degree-of-freedom K-H-V non-circular planetary gear train, which leads to poor trajectory and attitude, and poor precision of movement. This study has proposed a simple structure of transplanting mechanism with differential internal engagement non-circular planetary gear trains, which reconstructs the complex transplanting trajectory and attitude of the planting arm through single-stage unequal speed transmission. The working principle of the transplanting mechanism was analysed, and the kinematic theoretical model of the transplanting mechanism was established. The optimal design software for the transplanting mechanism was developed based on the visual platform. The dimensions optimisation on the transplanting mechanism was completed considering the requirements with strong coupling, and multi-objective and a set of superior mechanism parameters were obtained. The design theory and method of the internal engagement non-circular gear pair were proposed based on the generating principle. The correctness and accuracy of the trajectory and attitude of the transplanting mechanism were verified through virtual simulation experiments. The experiments show that the designed transplanting mechanism with internal engagement non-circular planetary gear trains was compact in structure, the trajectory meets the requirements of multi-objective transplanting, and the trajectory and attitude can be accurately reproduced, which provides a new feasible solution for the innovative design of the transplanting mechanism.

Correspondence: Maile Zhou, School of Agricultural Engineering and Key Laboratory of Modern Agricultural Equipment and Technology, Jiangsu University, Zhenjiang 212013, China.
E-mail: zhoumaile@126.com

Key words: Transplanting mechanism; trajectory and attitude; non-circular gear; internal gear; dimensions optimisation.

Acknowledgements: this research was financially supported by the National Natural Science Foundation of China (Grant No. 52005221), Natural Science Foundation of Jiangsu Province (Grant No. BK20200897), Jiangsu Agricultural Science and Technology Independent Innovation Fund Project (Grant No. CX(22)3089), China Postdoctoral Science Foundation (Grant No. 2021M691315), Key Laboratory of Modern Agricultural Equipment and Technology (Jiangsu University), and High-Tech Key Laboratory of Agricultural Equipment and Intelligence of Jiangsu Province.

See online Appendix for additional figures.

Received for publication: 20 March 2022.
Revision received: 15 May 2022.
Accepted for publication: 15 May 2022.

Journal of Agricultural Engineering 2022; LIII:1412
doi:10.4081/jae.2022.1412

Introduction

Rice transplanting is the process of transplanting rice seedlings into paddy fields. Transplanter was widely used to complete the mechanised transplanting. The transplanting mechanism was the core working component of the transplanter, and it planted rice seedlings into the paddy field by tearing off a piece of seedlings from the seedling box. The transplanting mechanism was generally composed of the drive system and planting arms. The drive system constrains the movement law of the planting arms so that the planting arms meet the requirements of the trajectory and attitude of the seedlings picking, conveying, planting, and returning stages. According to the different drive system, the transplanting mechanism can be divided into three types: crank-slider, bar, and gear train (Yu *et al*., 2014). In the crank-slider mechanism, the trajectory and attitude of the planting arm were controlled by the higher pair mechanism; there were problems such as vibration and wear, which have tended to be eliminated. The bar mechanism often used a planar four-bar linkage to drive the planting arm to complete the transplanting process; there were vibration, dynamic imbalance, and other problems generally used for the low-speed operation of the walk transplanter. The gear train mechanism widely used single-degree-of-freedom non-circular planetary gear train as a drive system, which has the characteristics of stable transmission and easy to achieve dynamic balance and was generally used on the riding transplanter (Yu *et al*., 2019; Zhao *et al*., 2011). Thomas used a planar four-bar linkage to design a four-row self-propelled transplanter; by varying the dimensions of various links, he obtained the trajectory for picking, conveying, and planting (Thomas *et al*., 2002). Felezi *et al*. synthesized an optimal reconfigurable four-bar linkage for a rice seedling transplanting machine. They presented some practical curves for the optimum design configuration of the mechanism, meeting the different specifications of both planting depth and forward velocity (Felezi *et al*., 2015). Imran *et al*. designed a system of rice intensification transplanter using a linkage mechanism. The system of rice intensification transplanter has a low fabrication cost (Imran *et al*., 2017). Zhang *et al*. designed a narrow row spacing transplanting mechanism based on the parameters of the existing transplanter. The 'three rates' met the requirements of rice transplanting but did not consider the influence of the vibration of the mechanism (Zhang *et al*., 2015). Fu *et al*. proposed a time-varying uncertainty analysis method for the trajectory of the transplanting mechanism based on the bar mechanism and established a calculation model for the reliability of the transplanting mechanism trajectory (Fu *et al*., 2019). Liu *et al*. established a dynamic model of the bar mechanism with a rotating pair gap, analysed the influence of the rotating pair gap on the trajectory, and provided a reference for the optimal structural design and precision distribution of the transplanting mechanism (Liu *et al*., 2016). Zhu *et al*. optimised the parameters of the crank rocker mechanism for the error squared sum of the trajectory of the seedling needle tip as the optimisation objectives but did not consider the dynamic characteristics of the transplanting mechanism (Zhu *et al*., 2016). Zhao *et al*. designed an eccentric sprocket wheel transplanting mechanism, which transfers power by eccentric sprocket wheel, and the eccentric tensioner eliminates the chain change of the degree of tightness, which was suitable for mechanised transplanting of double-cropping rice. However, the chain drive has problems such as speed fluctuations and can only be used in low-speed working environments (Zhao *et al*., 2000). Guo *et al*. developed the so-called parametric equation for the eccentric planetary gear train and designed a rice transplanting mechanism. However, the effect of the backlash of gear accumulation on the transplanting mechanism was not analysed (Guo *et al*., 2001). Chen *et al*. used K-H-V non-circular planetary gear train to achieve the transplanting trajectory and designed a device for synchronous open-hole transplanting on mulching film. The kinematic analysis was carried out, and there was a problem of unequal speed transmission ratio mutual restriction (Chen *et al*., 2016). Zhang *et al*. took the width of the hole, the push seedling angle, and the trajectory height as the optimisation objectives and adopted the experimental design method of virtual centre response surface combination to design a three-arm transplanting mechanism. The two-stage ellipse gear drive realises the transplanting trajectory, and there are problems such as side gap accumulation (Zhang *et al*., 2016). Li *et al*. proposed a design method to directly solve the non-circular gear transmission ratio of the transplanting mechanism using the static trajectory, reversely designed the non-circular gear pitch curve and its tooth profile and set up a device to reduce the side gap accumulation in the transplanting mechanism (Li *et al*., 2016). Ye *et al*. designed a kind of rice potted seedling transplanting mechanism with a planetary gear train based on the drive with incomplete eccentric circular gear and non-circular gears, then they established the kinematic model. Analysing the influence of design variables on the optimisation objectives of the transplanting mechanism, a set of parameters was obtained through human-computer interaction (Ye *et al*., 2017). Zhao *et al*. analysed the non-circular planetary gear train transplanting mechanism as a combination of the bar-group and gear train system. According to three points with precise position and attitude in the trajectory, they obtained the solution domain of the parameters of the bar group and designed the desired shape of the transplanting trajectory and the transmission ratio curve (Zhao *et al*., 2018). Sun *et al*. proposed an irregular, non-circular bevel gear transmission mechanism based on the sphere pitch curve, and designed a wide-narrow row spacing transplanting mechanism. Based on the influence law of irregular shape non-circular bevel gear on the trajectory and attitude, the structural parameters of the transplanting mechanism were optimised, but no coping strategies were proposed for the side gap accumulation (Sun *et al*., 2015). Yin *et al*. proposed a differential non-circular gear train transplanting mechanism. They designed a cam mechanism as a swing follower that compensates for the side gap accumulation, which can always maintain the working contact of the gear teeth, ensuring the stable position of the planting arm, the accurate position, and the stability of the operation of the transplanting mechanism, but the structure was complex (Yin *et al*., 2012; Sun *et al*., 2015).

In this study, a kind of transplanting mechanism with internal engagement non-circular gear trains was proposed given the problems of unequal speed transmission ratio mutual restriction, side gap accumulation, and small coincidence degree of transplanting mechanism with single-degree-of-freedom K-H-V non-circular planetary gear train, resulting in poor seedling transplanting attitude and poor precision of movement. The mechanism restricts the movement law of the planting arms by differential internal engagement non-circular gear trains so that the planting arms reconstruct the transplanting trajectory and attitude, avoid problems such as side gap accumulation, and have the advantages of compact structure and high motion accuracy. The design theory and method of internal engagement non-circular gear were studied, and the restriction law of differential internal engagement non-circular gear train on the planting arms was determined. Differential internal engagement non-circular gear train and its design method can also be applied to the rice potted seedling transplanting mechanism and vegetable potted seedling picking mechanism, which has important theoretical and practical significance for the development of transplanting machinery industry.

Materials and methods

Composition and working principle

The transplanting mechanism with a single-degree-of-freedom planetary gear train was generally composed of a K-H-V non-circular planetary gear train widely used in the riding transplanter. The transplanting mechanism widely adopts the structure of two planting arms, which can complete two transplanting actions in a working cycle. The K-H-V non-circular planetary gear train driven planting arms composed of sun gear, middle gears, and planetary gears to complete the transplanting process, as shown in Figure 1. The transplanting mechanism with differential internal engagement non-circular gear trains does not need to set the middle gears avoids the side gap accumulation and has the characteristics of a compact structure, as shown in Figure 2. In order to improve the transplanting efficiency, the transplanting mechanism with differential internal engagement non-circular gear trains adopts the structure of three planting arms. The transplanting mechanism with differential internal engagement non-circular gear trains consists of an internal engagement non-circular gear train and three planting arms. The internal engagement non-circular gear train includes non-circular internal gear, planet carrier, and three non-circular planetary gears. The non-circular internal gear and the planetary carrier were coaxially articulated on the frame, the planetary carrier was arranged into a 120° uniform distribution structure, the three non-circular planetary gears were hinged on the corresponding planetary carrier, and the three planting arms were fixed on the three corresponding non-circular planetary gears.

During work, the non-circular internal gear and the planet carrier rotate clockwise at differential speeds. On the one hand, the three non-circular planetary gears rotate circumferentially with the planet carrier; on the other hand, they mesh with the non-circular internal gear and rotate at an unequal speed relative to the planet carrier. In a working cycle, the motion laws of the three planting arms are the same, and their trajectories and attitudes are completely consistent, as shown in Appendix Figure 1. The planting arms were connected with the non-circular planetary gears to make a complex plane movement and realise the transplanting attitude in the picking, conveying, planting, and returning stages and its end-point forms the seedlings transplanting trajectory.

Kinematic analysis

The structural parameters of the three planting arms in the transplanting mechanism with differential internal engagement non-circular gear trains were the same, and they differ only by 120° with the change of the phase angle of the planetary carrier. That is, the position of planting arm II was when planting arm I rotated 120° with the planetary carrier, and the position of planting arm III was when planting arm II rotated 120° with the planetary carrier. The following was to establish a kinematic model of the transplanting mechanism using planting arm I as an example. In order to facilitate mechanism analysis and parameter optimisation, the initial meshing point of the non-circular gear was taken as the initial position, and the initial position of the corresponding planetary carrier is consistent with that of the non-circular gear. The initial installation angle of the planetary carrier was φ_{H0}, the non-circular internal gear rotates clockwise at the angular velocity ω_1, the planetary carrier rotates clockwise at the angular velocity ω_2, and $\omega_1 \neq \omega_2$. The non-circular planetary gears mesh with the non-circular internal gear while rotating with the planetary carrier. A schematic diagram of the movement of the transplanting mechanism is shown in Figure 3; the solid line indicates the initial installation position of the transplanting mechanism, and the dotted line represents the position after the movement time t by the initial position.

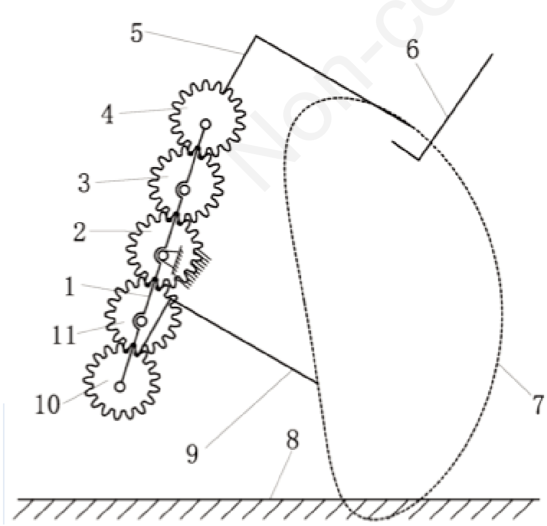

Figure 1. Transplanting mechanism with single-degree-of-freedom planetary gear trains. 1. Planetary carrier; 2. Sun gear; 3. Middle gear I; 4. Planetary gear I; 5. Planting arm I; 6. Seedling box; 7. Relative motion trajectory; 8. Ground; 9. Planting arm II; 10. Planetary gear II; 11. Middle gear II.

After time t, the angle at which the non-circular internal gear was turned clockwise from the initial position:

$$\theta_1 = \omega_1 t \tag{1}$$

The angle at which the planetary carrier rotates clockwise from its initial position:

$$\theta_2 = \omega_2 t \tag{2}$$

Angular displacement of the planetary carrier:

$$\varphi_H = \varphi_{H0} - \theta_2 \tag{3}$$

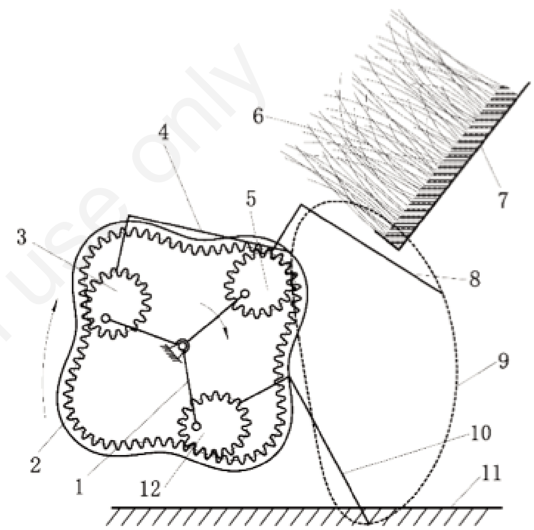

Figure 2. Transplanting mechanism with differential internal engagement non-circular gear trains. 1. Planetary carrier; 2. Non-circular internal gear; 3. Non-circular planetary gear I; 4. Planting arm I; 5. Non-circular planetary gear II; 6. Seedlings; 7. Seedling box; 8. Planting arm II; 9. Relative motion trajectory; 10. Planting arm III; 11. Ground; 12. Non-circular planetary gear III.

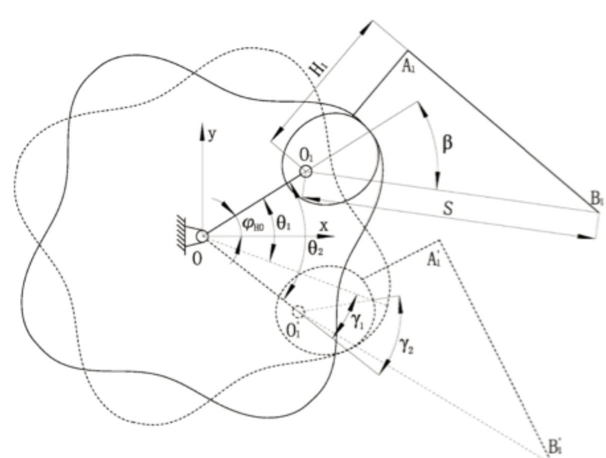

Figure 3. Schematic diagram of kinematic analysis.

The angle difference between the planetary carrier and the non-circular internal gear, that is, the planetary carrier rotates clockwise relative to the non-circular internal gear (the non-circular internal gear rotates counter clockwise relative to the planetary carrier):

$$\gamma_1 = \theta_2 - \theta_1 \tag{4}$$

The non-circular planetary gear meshes with the non-circular internal gear, and the non-circular planetary gear rotates counter clockwise relative to the planetary carrier:

$$\gamma_2 = \int_0^{\gamma_1} r_1 / r_2 \, d\theta \tag{5}$$

Among them, r_1 and r_2 were the radii of the mesh point corresponding to the non-circular internal gear and the non-circular planetary gear.

Angular displacement of non-circular internal gear:

$$\varphi_1 = \varphi_{H0} - \theta_1 \tag{6}$$

Angular displacement of non-circular planetary gear:

$$\varphi_2 = \varphi_{H0} - \theta_2 + \gamma_2 \tag{7}$$

Coordinates of rotation centre of the non-circular planetary gear after time t from the initial position:

$$\begin{cases} O'_{1x} = |OO_1| \cos \varphi_H \\ O'_{1y} = |OO_1| \sin \varphi_H \end{cases} \tag{8}$$

Displacement of point A_1 on the planting arm after time t from the initial position:

$$\begin{cases} A'_{1x} = O'_{1x} + H_1 \cos\left(\varphi_2 + \beta + \arccos H_1 / S\right) \\ A'_{1y} = O'_{1y} + H_1 \sin\left(\varphi_2 + \beta + \arccos H_1 / S\right) \end{cases} \tag{9}$$

Among them, the β is the initial installation angle of the planting arm relative to the non-circular planetary gear, H_1 is the distance from the rotation centre of the non-circular planetary gear to the point A_1 on the planting arm, and S is the distance from the rotation centre of the non-circular planetary gear to the point B_1 on the planting arm.

Displacement of point B_1 on the planting arm after time t from the initial position:

$$\begin{cases} B'_{1x} = O'_{1x} + S \cos(\varphi_2 + \beta) \\ B'_{1y} = O'_{1y} + S \sin(\varphi_2 + \beta) \end{cases} \tag{10}$$

Development of optimal design software and parameter optimisation

Optimal design software

Transplanting mechanism with differential internal engagement non-circular gear trains in the working process needs to go through the stages of picking, conveying, planting and return, *etc*. Its working trajectory and attitude were complex, from the driving link to the planting arms needed to pass a complex unequal speed transmission. It was challenging to obtain the trajectory and attitude under different parameters through the drawing method. Based on the kinematic theoretical model of the transplanting mechanism, the known variables were the angle of the non-circular internal gear and the angle of the planetary carrier. An optimal design software for the transplanting mechanism was developed based on the visual programming environment, as shown in Appendix Figure 2. The optimal design software can calculate the trajectory and attitude of the transplanting mechanism in real-time according to the different parameters of the mechanism and intuitively display the mechanism motion simulation under the current parameters. Designers can use the software to analyse the trajectory and attitude of the transplanting mechanism, the angle, and speed of the planting arm, the unequal transmission ratio of the non-circular gear train, *etc*., which can significantly shorten the design cycle of the transplanting mechanism.

Parameter optimisation

Rice mechanised transplanting is a complex operation process, the motion trajectory and attitude requirements of the transplanting mechanism are high, and parameter optimisation is a complex, strong coupling, multi-objective, multi-parameter, non-linear optimisation problem. Therefore, it is necessary to analyse the law of influence of the mechanism parameters on the trajectory and attitude and then complete the parameters optimisation of the transplanting mechanism. Based on the requirements of the key stages of picking, conveying, and planting in the transplanting operation and the motion characteristics of the transplanting mechanism, this study analyses the law of influence of each parameter on the trajectory, as shown in Appendix Figure 3. The differential ratio of the two prime movers and the unequal speed transmission ratio of non-circular gear affect the trajectory and attitude of the transplanting arm.

Based on the work requirements of the key stages of picking, conveying and planting in the process of transplanting operation and the kinematic characteristics of the transplanting mechanism, the functional relationship between the objectives and parameters was established based on the kinematic model. A series of digital optimisation objectives were proposed: i) non-interference among the three planting arms; ii) non-interference between the planting arms and the seedling box; iii) the seedling picking angle is greater than 320° and less than 350°; iv) the seedling pushing angle is greater than 270° and less than 310°; v) the angle difference is greater than 45° and less than 65°; vi) the rectangular seedling is removed; vii) the planted seedlings are not bridged; viii) the planting arms do not push down the seedlings; ix) the width of the hole is less than 30 mm; x) the distance between gearbox and ground is greater than 20 mm; xi) the modulus of the non-circular gear is greater than 2.5 mm; xii) the seedling needle does not press the seedlings when picking. Based on the optimal design software of the transplanting mechanism, a set of mechanism parameters meeting a series of optimisation objectives were obtained: n=4, r_0=100 mm, r_1=98.5 mm, θ_1=30°, r_2=101 mm, θ_2=60°, r_3=81 mm, θ_3=90°, r_4=82 mm, θ_4=120°, r_5=66 mm, θ_5=150°, r_6=64 mm, θ_6=180°, r_7=79.5 mm, θ_7=210°, r_8=83 mm, θ_8=240°, r_9=95 mm, θ_9=270°, r_{10}=98 mm, θ_{10}=300°, θ_{11}=330°, Φ_{H0}=40°, S=150 mm, β= –40°, H_1=80 mm, Yx=165 mm, Yy=70 mm, Ya=52°. Among them, n is the average transmission ratio between the non-circular planetary gear and the non-circular internal gear, r_i and θ_i are the parameters of the pitch curve of non-circular gear, Φ_{H0} is the initial installation angle of the planetary carrier, S is the distance from the rotation centre of the non-circular planetary gear to the tip of the planting arm, the β is the installation angle of the planting arm relative to the non-circular planetary gear, H_1 is the distance from the rotation centre of the non-circular planetary gear to the break point of the planting arm, Yx and Yy are the coordinates of the seedling box, and Ya is the inclination angle of the seedling box. Under this set of parameters, the working trajectory of the transplanting mechanism is shown in Figure 4. The figure's angle data are the planetary carrier's angle corresponding to the trajectory point, which respectively represents the rotation angle of the planetary carrier relative to the initial position when the three planting arms move to the corresponding position of the trajectory.

Design theory and method of internal engagement non-circular gear

Numerical envelope model of the gear tooth profile of non-circular gear

The tooth profile of the non-circular gear is complex. The tooth profile of each gear tooth of the same gear is different, and the tooth profile on both sides of the same gear tooth is also quite different. Therefore, the tooth profile design is difficult for the non-circular design. Non-circular internal gear has the characteristics of a high contact ratio and small sliding coefficient, but the tooth profile design method of non-circular internal gear was rarely reported. This study proposed a theory and method of the internal engagement non-circular gear pair. Based on the generating principle, this study deduces the numerical envelope model of the gear cutter. The gear cutter tooth profile envelops the tooth profile of non-circular gear and theoretically can form the non-circular gear tooth profile of the arbitrary unequal speed transmission law. The tooth envelope process of the non-circular internal gear pair was shown in Figure 5, and the pitch curve of the gear cutter was purely rolled along the non-circular gear pitch curve; that is, the gear cutter pitch circle was tangent with the pitch curve at the corresponding engagement point of the pitch curve. In order to ensure that the

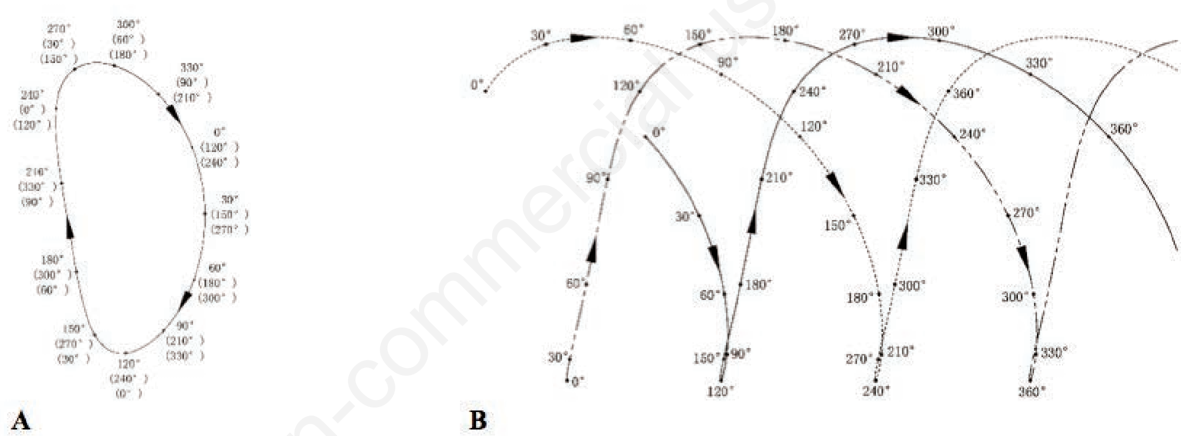

Figure 4. Trajectories of the transplanting mechanism. A) Relative motion trajectory; B) Absolute motion trajectory.

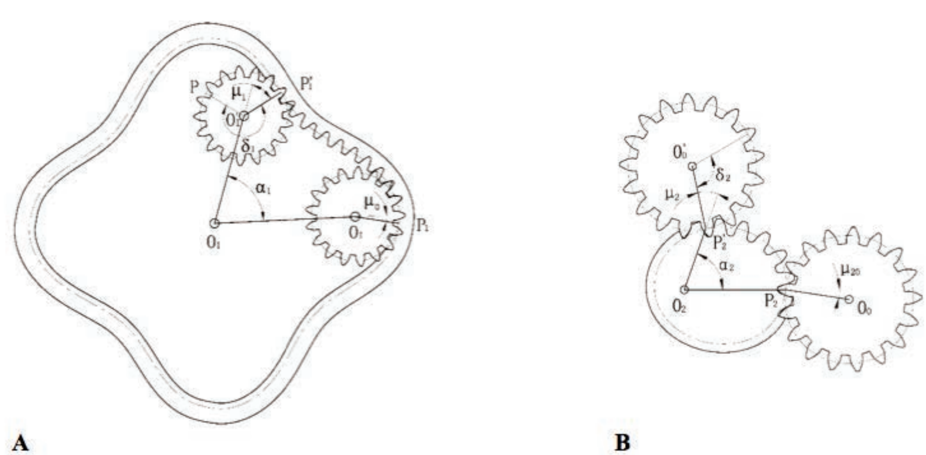

Figure 5. Diagram of the envelope of internal engagement non-circular gear pair. A) Non-circular internal gear; B) Non-circular external gear.

non-circular internal gear pair could be correctly engaged and rotated when enveloping the tooth profile of the driving gear and driven gear, the gear cutter was rolled counter clockwise and enveloped along the non-circular gear pitch curve by the initial engagement position.

The tooth profile envelope process of the non-circular internal gear is shown in Figure 5A. The pitch curve's engagement point corresponding to the initial envelope position is P1, while the engagement point corresponding to a specific envelope position in the tooth profile forming is P_1'. Taking a series of point sets (x_i, y_i) on the pitch curve as the known data, the numerical envelope model of the non-circular internal gear (driving gear) was derived:

At the initial engagement point position P_1, the centre coordinate of the gear cutter pitch circle is:

$$\begin{cases} O_{1x} = P_{1x} + \dfrac{d}{2}\cos\left(\arctan\dfrac{y_1 - y_0}{x_1 - x_0} + \dfrac{\pi}{2}\right) \\ O_{1y} = P_{1y} + \dfrac{d}{2}\sin\left(\arctan\dfrac{y_1 - y_0}{x_1 - x_0} + \dfrac{\pi}{2}\right) \end{cases} \tag{11}$$

where d is the pitch circle diameter of the gear cutter.

At the position P_1' of the engagement point, the centre coordinate of the gear cutter pitch circle is:

$$\begin{cases} O'_{1x} = P'_{1x} + \dfrac{d}{2}\cos\left(\arctan\dfrac{y_{i+1} - y_{i-1}}{x_{i+1} - x_{i-1}} + \dfrac{\pi}{2}\right) \\ O'_{1y} = P'_{1y} + \dfrac{d}{2}\sin\left(\arctan\dfrac{y_{i+1} - y_{i-1}}{x_{i+1} - x_{i-1}} + \dfrac{\pi}{2}\right) \end{cases} \tag{12}$$

When the gear cutter rolls from the initial engagement point P_1 to the engagement point P_1', the distance at which the pitch curve was purely rolled is:

$$L = \sum_{j=0}^{i-1}\sqrt{\left(x_{j+1} - x_j\right)^2 + \left(y_{j+1} - y_j\right)^2} \tag{13}$$

When the gear cutter rolls from the initial engagement point P_1 to the engagement point P_1', the angular displacement of gear cutter due to pure rolling is:

$$\delta_1 = L \Big/ \left(d/2\right) \tag{14}$$

At the initial engagement point P_1 position, the included angle between the rotation centre line and the normal pitch curve is:

$$\mu_0 = \arctan\frac{O_{1y}}{O_{1x}} \pm \arctan\frac{P_{1y} - O_{1y}}{P_{1x} - O_{1x}} \tag{15}$$

At the engagement point P_1' position, the included angle between the rotation centre line and the normal pitch curve is:

$$\mu_1 = \arctan\frac{O'_{1y}}{O'_{1x}} \pm \arctan\frac{P'_{1y} - O'_{1y}}{P'_{1x} - O'_{1x}} \tag{16}$$

During the enveloping process, the angular displacement of gear cutter due to the change in the centre of the gear cutter pitch circle:

$$\alpha_1 = \arctan\frac{O'_{1y}\text{-}O_{1y}}{O'_{1x}\text{-}O_{1x}} \tag{17}$$

When the gear cutter rolls from the initial engagement point P_1 to the engagement point P_1', the absolute angular displacement of the gear cutter is:

$$\theta_1 = \alpha_1 + \left(\mu_1 - \mu_0\right) + \delta_1 \tag{18}$$

The gear tooth envelope process of the non-circular external gear (driven gear) is shown in Figure 5B, and the engagement point of the initial envelope position is P_2, and the engagement point of one envelope position is P_2'. The enveloping process of the gear cutter rolling along the driven gear pitch curve is similar to that of the non-circular internal gear (driving gear), but the gear cutter is enveloping on the outside of the driven gear pitch curve.

The tooth profile of the non-circular gear is complex and difficult to form. Wu *et al.* used a rack cutter to envelop the tooth profile of the elliptical gear and obtained the characteristic points of the elliptical gear tooth profile by solving the intersection point of the rack cutter tooth profile. However, the rack cutter can only form non-circular gear profiles with fully convex pitch curves and cannot be used to form non-circular internal gears (Wu *et al.*, 2008). Based on the numerical envelope model of the non-circular gear profile, this study develops the 3D forming software for non-circular internal gear pair. The software can automatically establish the model of non-circular gear bases and gear cutters based on the input non-circular gear parameters (pitch curve and tooth profile parameters) and can form non-circular gear profiles based on envelope features. In the process of forming the 3D model of the non-circular internal gear pair, the gear cutter was enveloped along the non-circular gear pitch curve on the one hand, and the Boolean subtraction operation was performed with the non-circular gear base on the other hand, and the 3D model of the non-circular gear can be obtained after the envelope was completed, as shown in Figure 6.

Because the tooth profile of each tooth of non-circular gear is different, it is difficult to be formed by conventional machining. However, after designing the tooth profile of non-circular internal gear, non-circular internal gear can be manufactured by wire cutting or powder metallurgy process, and the manufacturing cost is slightly higher than that of circular gear.

Virtual experiment studies

Trajectory and attitude verification

According to the optimised mechanism parameters, the structural design of the transplanting mechanism was completed, and the virtual prototype model of the transplanting mechanism was established. In addition, the trajectory and attitude of the transplanting mechanism were verified, as shown in Figure 7. The virtual simulation experiments show that the three planting arms of the transplanting mechanism can complete the picking, conveying, planting, and returning process, and the attitude of the planting arms meet the seedling transplanting requirements. The virtual simulation results are highly consistent with the optimal design software results, and the correctness of the virtual experiment, kinematic theory analysis, and optimal design software was verified by each other, as shown in Figures 8 and 9. Appendix Figure 4 shows the change of the planting arms' velocities with the planetary carrier's rotation angle in one cycle when the angular velocity of the planetary carrier is constant.

In one working cycle (the planetary carrier rotates 360°), the three planting arms alternately complete the picking, conveying, planting and return processes, as shown in Appendix Figure 5. In the initial position, planting arm I was in the stage of return, planting arm II was in the stage of conveying, and planting arm III was in the stage of planting. For planting arm I, the stage of picking seedlings was (82°, 94°), the stage of conveying was (94°, 226°),

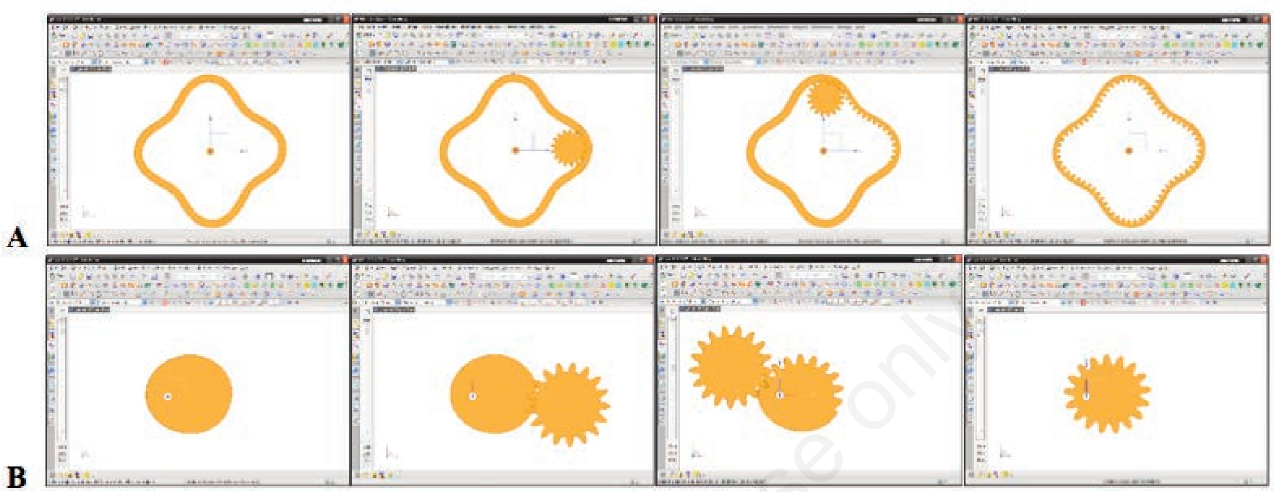

Figure 6. Schematic diagram of non-circular gear 3D forming. A) Driving gear forming process; B) Driven gear forming process.

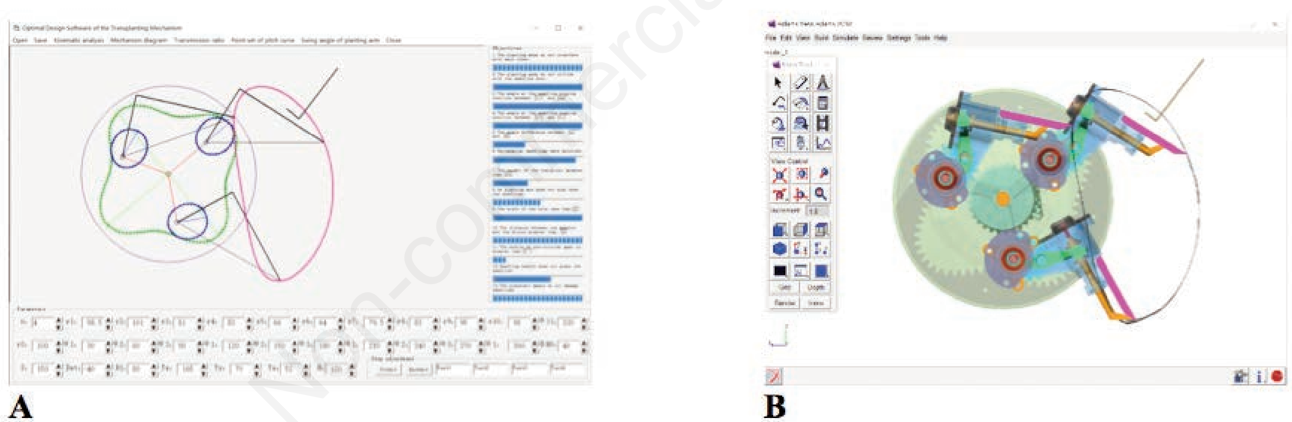

Figure 7. Trajectory comparison chart. A) Optimal design software; B) Virtual simulation.

Figure 8. Results of optimal design software. A) Picking; B) Conveying; C) Planting.

and the stage of planting was (226°, 243°). For planting arm II, the stage of picking was (322°, 334°), the stage of conveying was (334°, 360°) ∪ (0°, 106°), and the stage of planting was (106°, 123°); For planting arm III, the stage of picking was (202°, 214°), the stage of conveying was (214°, 346°), the stage of the planting was (346°, 360°) ∪ (0°, 3°).

Results and discussion

The trajectory and attitude of the transplanting mechanism determine the performance of the mechanised transplanting. According to the performance indexes of seedlings picking, conveying, and planting and the motion reliability of the transplanting mechanism, the achievement degree of the optimisation objectives was analysed through the virtual simulation experiment of the transplanting mechanism:

i) *Non-Interference among the three planting arms:* During work, the three planting arms alternately complete the picking, conveying, and planting stages, requiring that the three planting arms cannot interfere with each other. After virtual simulation analysis, one planting arm and the other two planting arms have two possible interference positions in one work cycle, as shown in Figure 10A and B. The analysis found that non-interference occurred in the two potential interference positions, and the three planting arms avoided each other during the movement, ensuring the movement space of each planting arm.

ii) *Non-Interference between the planting arms and the seedling box:* Considering that the planting arms need to complete the stages of the picking and planting, their internal structure was designed with cams, forks, seedling needles, and other parts, resulting in large size of the tail structure of the planting arms (unable to pass through the seedling door). However, after simulation analysis, during the work of the planting arms, only the front part of the seedling needle enters the seedling door and picking seedlings with a kind of seedling removal attitude, and the tail of the planting arms does not pass through the seedling door, and interference no occurs with the planting arms and the seedling box, as shown in Figure 10C.

iii) *The seedling picking angle is greater than 320° and less than 350°:* Seedling picking is one of the key stages of the transplanting mechanism for rice transplanting, which has an essential impact on the indexes such as the rate of injury to seedlings, the rate of seedlings leakage, and the amount of picking seedlings. Therefore, the transplanting mechanism should complete the seedling picking action at an angle greater than 320° and less than 350°. The average angle of picking the seedling process of the planting arms in the transplanting mechanism was 342° (Figure 10D), which meets the requirements of mechanised seedling picking.

iv) *The seedling pushing angle is greater than 270° and less than 310°:* The seedling pushing stage determines the seedlings' planting quality, which is another key stage for the transplanting mechanism to complete the transplanting. Studies have shown that the attitude (angle) of the planting arms should be greater than 270° and less than 310° when pushing seedlings to meet the planting requirements. After simulation analysis, the average angle of the planting arms of the transplanting mechanism was 296° (Figure 10E) when pushing seedlings, which meets the requirements of mechanised planting.

v) *The angle difference is greater than 45° and less than 65°:* The angle difference refers to the angle difference of the planting arm between the picking and planting positions, which is the crucial factor in determining the upright planting. After picking the seedlings, the planting arm conveys the seedlings to the planting position and turns them at a certain angle (about the inclination of the seedling box) to ensure that the seedlings are upright when planting. The inclination angle of the seedling box is generally set to 55° to meet the requirements of a stable seedling supply. However, to ensure the planting performance, the angle difference is generally greater than 45° and less than 65°. In the transplanting mechanism, the angle difference was 46°.

vi) *The rectangular seedling is removed:* Removing the rectangular seedling means that the matrix interface of the removed seedlings was rectangular, which can effectively avoid the main root being cut off in the matrix, which is the primary measure to reduce the injury of seedlings, especially the root. This measure was calculated by the size difference in the upper and lower surface of the removed seedlings matrix. For example, taking the case of matrix thickness of 20 mm and a single sample of 8 mm (Figure 10F), the average length of the upper surface of the removed seedlings was 7.99 mm, and the average length of the lower surface was 8.05 mm, with a difference of 0.06 mm, which can be regarded as the rectangular seedlings.

vii) *The planted seedlings are not bridged:* The bridge phenomenon refers to the phenomenon that when mechanised rice transplanting is carried out, the leaf tip of the previous seedlings is inserted into the root of the latter seedlings, and the planted seedlings become the shape of an arch bridge. The bridge phenomenon occurs because the trajectory height of the

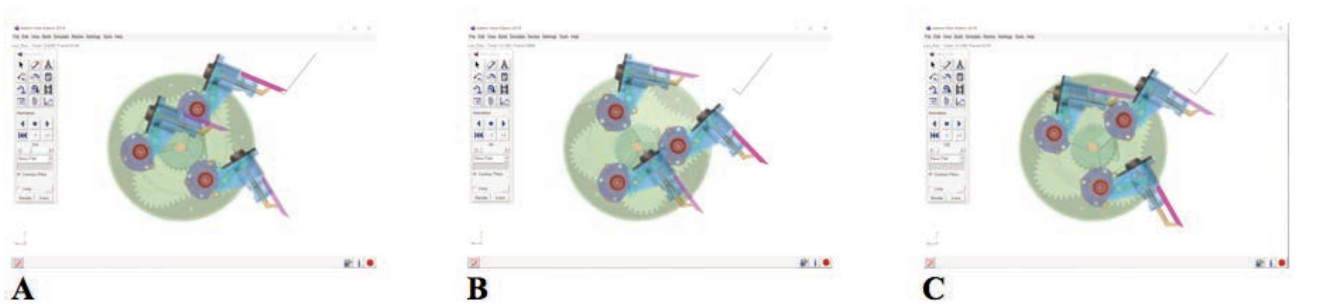

Figure 9. Results of virtual simulation. A) Picking; B) Conveying; C) Planting.

transplanting mechanism is not high enough to cross the seedlings' leaf tip, so that seedlings are planted into the paddy field in the next working cycle. The transplanting mechanism is designed with a trajectory height of 245 mm, which can meet the planting requirements of most seedlings.

viii) The planting arms do not push down the seedlings: After the planting arms complete the planting action, they should move up from the rear of the planted seedlings to the leaf tip of the seedlings, bypass the planted seedlings and then move forward, avoid pushing down the planted seedlings, and ensure the perpendicularity of the seedlings. The rice mechanisation standard stipulates that the planting uprightness should ensure that the angle between the seedlings and the ground is greater than 70°. After the transplanting mechanism was planted, its absolute motion trajectory forward inclination angle was 78° (Figure 10G), meeting the requirements of planting uprightness.

ix) The width of the hole is less than 30 mm: The width of the hole refers to the length of the hole formed after the planting arms enter the soil along the forward direction when the transplanting mechanism works with the transplanter. When the hole's opening is wide, the planted seedlings are easily dumped. Tests have shown that when the hole width is less than 30 mm, the planted seedlings are guaranteed to be stood. The hole width is directly related to the plant spacing, and the virtual simulation shows that the transplanting mechanism meets the plant spacing range of 60-180 mm when the hole width is less than 30 mm.

x) The distance between gearbox and ground is greater than 20 mm: On the one hand, the transplanting mechanism rotates to complete the picking, conveying and planting stages; on the other hand, it moves forward with the transplanter. In order to reduce power losses, the gearbox of the transplanting mechanism should be avoided from stirring in the soil. Considering the unevenness of the paddy field, the distance from the gearbox to the ground is more than 20 mm.

xi) The modulus of the non-circular gear is greater than 2.5 mm: The modulus of non-circular gears has a greater impact on the strength of the gears; with reference to the resistance of the working process of the transplanting mechanism, the modulus of the non-circular gears in the transplanting mechanism should be greater than 2.5 mm. The non-circular gear module designed for this study was 2.78 mm, which meets the design requirements.

xii) The seedling needle does not press the seedlings when picking: When the planting arms enter the seedling door to take the seedlings, they press them until they contact the matrix and start tearing them. In order to prevent crushing the seedling stalks and reducing the rate of injury seedlings when planting arms picking the seedlings, the planting arms should insert the seedlings at a certain angle and remove the seedlings. The transplanting mechanism has an angle of 16° between the planting arm and the growth direction of the seedlings when picking seedlings (as shown in Figure 10H), which can avoid the phenomenon of the planting arm pressing seedlings.

Conclusions

Using the differential non-circular gear train to constrain the motion law of planting arms, a transplanting mechanism with the differential non-circular gear trains was designed. Compared with the single-degree-of-freedom non-circular gear train transplanting mechanism, the transplanting mechanism with differential non-circular gear trains avoids the problems of mutual restriction of unequal speed transmission ratio and side gap accumulation. In addition, it has the advantages of simple structure and high motion accuracy. Differential internal engagement non-circular gear train mechanism can also be used for the design of rice potted seedlings transplanting mechanism and the picking mechanism for potted vegetable seedlings, which is of great practical significance for the

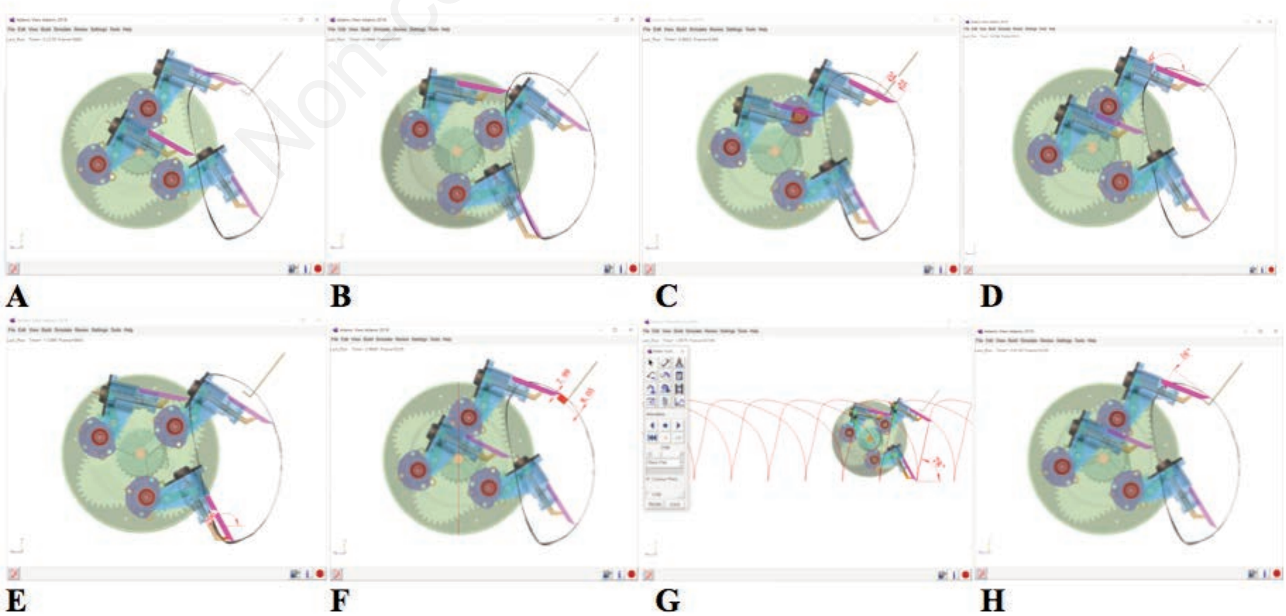
A B C D
E F G H

Figure 10. Analysis of optimisation objectives.

development of the transplanting industry.

Taking the angular displacement of the planetary carrier and the non-circular internal gear in the transplanting mechanism as the known variables, the kinematic model of the transplanting mechanism was established, and the optimal design software of the transplanting mechanism was developed. The digital optimisation objectives were embedded in the software, and a set of mechanism parameters was obtained. From kinematic analysis to the development of optimal design software to the mechanism parameter optimisation of the transplanting mechanism design method, the development cycle of the transplanting mechanism can be significantly shortened, and this method can also be applied to mechanical design in other fields.

Based on the design requirements of the transplanting mechanism, the design theory and method of non-circular internal gear pair were proposed. Pure rolling along the non-circular gear pitch curve by the pitch circle of the series gear cutter and the non-circular gear tooth profile was enveloped by the tooth profile of the gear cutters. The design theory and method of the non-circular inner gear pair are universal and can theoretically be used for non-circular gear pairs with arbitrary unequal transmission ratios.

The trajectory and attitude of the transplanting mechanism were analysed, and the multi-objective achievement degree of the transplanting mechanism was analysed. Under the preferential parameters, the three planting arms can complete the picking, conveying and planting stages in turn, to meet the requirements of the multi-objective optimisation design of the transplanting mechanism. The results of the virtual experiment show that the design method of the transplanting mechanism with differential non-circular gear trains was feasible, which provides a new design scheme for the design of the transplanting mechanism.

References

Chen H.T., Zhao Y., Hou S.Y., Cong G.B., Xu Y. 2016. Kinematics simulation and parameter optimization experiment for transplanting synchronous puncher. Trans. Chin. Agric. Eng. 32:25-30.

Felezi M.E., Vahabi S., Nariman-Zadeh N. 2015. Pareto optimal design of reconfigurable rice seedling transplanting mechanisms using multi-objective genetic algorithm. Neural Comput. Appl. 27:1-10.

Fu J.H., Shi B.Q., Yu G.Q., Li G.G. 2019. Evolution-based uncertainty analysis of separating-planting mechanism trajectory in transplanter. Machine. Design Manufact. 2019:1-4.

Guo L.S., Zhang W.J. 2001. Kinematic analysis of a rice transplanting mechanism with eccentric planetary gear trains. Mech. Mach. Theory. 36:1175-88.

Imran M.S., Abdul M., Khalil A.N.M., MdNaim M.K. 2017. Design of transplanting mechanism for system of rice intensification (SRI) transplanter in Kedah, Malaysia. Iop Conf. 226:012036.

Li G., Ying K.Y., Zhang J.Z., Li J.F., Yu G.H., Ye Y.S., Wang C., Li S.M. 2016. Computation method of non-circular gear based on seedling needle tip point's static trajectory in transplanting mechanism. Chin. J. Mech. Eng. 52:64-71.

Liu F.X., Wu C.Y., Sun L. 2016. Analysis and test of influence of revolute joint clearance on performance of crank-rocker style transplanting mechanism. Trans. Chin. Soc. Agric. Eng. 32:9-17.

Sun L.J., Yin J.J. 2015. Design on a kind of mechanism for compensating side clearance of non-uniform gear transmission in separating-transplanting mechanism. J. Agric. Mech. Res. 37:59-63.

Sun L., Zhu J.B., Zhang G.F., Fang Z., Yu G.H. 2015. Wide-narrow distance transplanting mechanism with special shaped non-circular bevel gears for rice transplanter. Trans. Chin. Soc. Agric. Mach. 46:54-61.

Thomas E.V. 2002. Development of a mechanism for transplanting rice seedlings. Mech. Mach. Theory. 37:395-410.

Wu C.Y., Jin Y.Z., He L.Y. 2008. Numerical algorithm of tooth profile of noncircular gear based on the characteristics of cutter envelope. China Mech. Eng. 19:1796-9.

Ye B.L., Yi W.M., Yu G.H., Gao Y., Zhao X. 2017. Optimization design and test of rice plug seedling transplanting mechanism of planetary gear train with incomplete eccentric circular gear and non-circular gears. Int. J. Agric. Biol. Eng. 10:43-55.

Yin J.J., Wu C.Y., Yang S.X., Li Y. 2012. Kinematic analysis of separating-transplanting mechanism with differential eccentric gear train base on inequality model. Trans. Chin. Soc. Agric. Mach. 43:53-9.

Yu G.H., Jin Y., Chang S.S., Ye B.L., Gu J.B., Zhao X. 2019. Design and test of clipping-plug type transplanting mechanism of rice plug-seedling. Trans. Chin. Soc. Agric. Mach. 50:100-8.

Yu X.X., Zhao Y., Chen B.C., Zhou M.L. Zhang H. Zhang Z.C. 2014. Current situation and prospect of transplanter. Trans. Chin. Soc. Agric. Mach. 45:44-53.

Zhang M., Zhang W.Y., Ji Y., Qi B. 2016. Parameter optimization and experiment of transplanting mechanism of rice transplanter based on virtual response surface analysis. J. China Agric. Univ. 21:114-21.

Zhang Q., Xiao L.P., Cai J.P., Liu M.H. 2015. The design and validation of transplanting mechanism on narrow spaced walking-type transplanter. J. Agric. Mech. Rese. 37:95-100.

Zhao X., Chu M.Y., Ma X.X., Dai L., Ye B.L., Chen J.N. 2018. Research on design method of non-circular planetary gear train transplanting mechanism based on precise poses and trajectory optimization. Adv. Mech. Eng. 10:1-12.

Zhao Y., Gao L.D., Chen J.N., Zhang G.F. 2011. Design and parameters optimization of deformed eccentric non-circular gears transplanting mechanism. Trans. Chin. Soc. Agric. Mach. 42:74-7.

Zhao Y., Jiang H.Y., Wu C.Y., Yin J.J. 2000. Structure analysis and parameter optimization of separating-planting mechanism with eccentric sprockets. Chin. J. Mech. Eng. 36:37-40.

Zhu D.Q., Yao Y.F., Yang S., Song Y., Zhang J.M., Wang Y.Q. 2016. Kinematics analysis and optimization design on separating-planting mechanism of narrow row spacing transplanter. J. Mach. Design. 33:73-7.