Design of flower transplanting mechanisms based on double planet carrier non-circular gear train with complete rotation kinematic pair

Xiong Zhao[1,2], Xingxiao Ma[1], Hongwei Liao[1], Yongsen Xiong[2], Yadan Xu[3], Jianneng Chen[1*]

(1. *Faculty of Mechanical Engineering and Automation, Zhejiang Sci-Tech University, Hangzhou 310018, China*;

2. *Key Laboratory of Crop Harvesting Equipment Technology of Zhejiang Province, Jinhua 321017, Zhejiang, China*;

3. *Hangzhou Vocational & Technical College, Hangzhou 310018, China*)

Abstract: In view of the problems of the existing mechanisms based on 2R open-chain planetary gear train for seedling transplanting, such as the bad tracking flexibility, low positioning accuracy, and high structure design difficulties of the mechanisms based on 3R open-chain planetary gear train for seedling manipulation. In this study, a transplanting mechanism based on the solution domain synthesis of a 3R open-chain-based complete rotation kinematic pair, a gear train with a single cycle integral rotating pair, is designed. The Burmester curve equation is derived from the given transplanting trajectory and four exact poses corresponding to each other on the rotation center. Then, the open-chain road model of the 3R complete rotation kinematic pair is obtained under the constraint governed by the judgment condition of the hinge integral rotating pair. Meanwhile, combined with our developed in-house optimization software, the solution to the optimal parameters for the transplanting mechanism can be optimized according to the target trajectory. Finally, the feasibility of the design method was verified by transplanting testing, where kale seedlings with ages of about 20 d and heights of about 80-120 mm were used. The experimental results showed that the actual motion trajectory of the prototype is basically identical to the theoretical trajectory, validating the feasibility of transplanting mechanism design, parts processing, and test-bed construction. Through the statistical analysis, the average success rate of transplanting is 90.625%, and the reliability of designed mechanism is satisfied. This study provides a promising solution for the seedling transplanting of two-planet scaffold pots.

Keywords: 3R open-chain planetary gear train, complete rotation kinematic pair, precise manipulation, plug seedling

DOI: 10.25165/j.ijabe.20221503.6957

Citation: Zhao X, Ma X X, Liao H W, Xiong Y S, Xu Y D, Chen J N. Design of flower transplanting mechanisms based on double planet carrier non-circular gear train with complete rotation kinematic pair. Int J Agric & Biol Eng, 2022; 15(3): 9–15.

1 Introduction

In recent years, in order to reduce the burden of planting crops, the emergence of mechanical transplanting has got a wide concern[1,2]. At present, many seedling transplanting machines are designed and developed based on the planetary gear train[3]. The transplanting mechanisms with 2R open-chain planetary gear trains are widely used in the automation of agricultural production because of their high transplanting efficiency, stable operation, and adaptivity to various trajectories. However, with the upgrading of agricultural transplanting demands in recent years, challenges exist in more complex and higher accurate transplanting trajectories generation. In the design of 2R mechanisms, the active rod is often not too long, which leads to the problems such as slight displacements of mechanisms and large limitations of the motion law, which also affects the efficiency and applicability of transplanting. Choi et al.[4] studied the sliding and wheel train combined transplanting mechanisms. Liang et al.[5] designed a rice pot seedling transplanting mechanism by using a double-crank five-bar mechanism, but such mechanisms have poor efficiency due to their transmission restriction. Sun et al.[6] realized the "eagle beak" clamp-seedling trajectory design with a single planetary gear train mechanism, but it was difficult to achieve the requirements of trajectories that have large displacement.

Although various novel designs of non-circular gears, such as incomplete non-circular gears and discontinuous non-circular gears, have been proposed to cope with inflexible trajectory generation[7], the configuration limitation of 2R open-chain planetary gear trains cannot be alleviated. In order to solve this issue, Chen et al.[8] proposed a study on double-rack swinging transplanting mechanisms, which added a set of crank-rocker mechanisms to the traditional transplanting mechanism which is based on 2R gear train for performing the trajectories that have more flexible shapes. However, the mechanism selection method used in this design is too subjective to provide sufficient improvements on the double planet rack mechanisms for this issue. On the other hand, a mechanism with 3R open-chain planetary gear train was designed by Zhao et al.[9] was implemented by adding a first-level convex two-wheel pair on the basis of the 2R open-chain mechanism, where the third rotating pair adopts the form of swinging roller push rod cam. Although such a mechanism can generate a larger displacement, the drawback that the needle cannot clamp the seedling during work impedes the success of seedling manipulation.

According to the above problems, this study proposed a

Received date: 2021-07-30 **Accepted date:** 2022-03-22

Biographies: Xiong Zhao, PhD, Associate Professor, research interest: mechanism design, non-circular gear transmissions, and agricultural machinery, Email: zhaoxiong@zstu.edu.cn; **Xingxiao Ma**, Master candidate, research interest: agricultural machinery, Email: 2416771239@qq.com; **Hongwei Liao**, MS, research interest: agricultural machinery, Email: 1153565139@qq.com; **Yongsen Xiong**, PhD, Professor, research interest: agricultural machinery, Email: 791228154@qq.com; **Yadan Xu**, PhD, Associate Professor, research interest: agricultural machinery, Email: 4732420@qq.com.

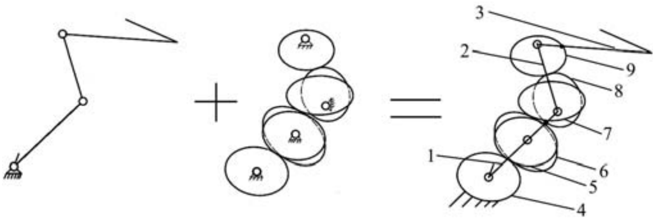
***Corresponding author: Jianneng Chen**, PhD, Professor, research interest: machine design, non-circular gear transmissions and control, and agricultural machinery. Faculty of Mechanical Engineering and Automation, Zhejiang Sci-Tech University, Hangzhou 310018, China. Tel: +86-13065701536, Email: jiannengchen@zstu.edu.cn.

method that combines the planetary gear train and the modern analytical theory on linkages to design a transplanting mechanism[10,11]. Based on a 3R open-chain planetary gear train, given four precise poses (including positions and orientations) on the desired trajectory, a type of transplanting mechanism that has a single-cycle complete revolution is designed. Such a mechanism can meet the requirements that need large displacements, and accurate and flexible trajectories, while ensuring the single-cycle movement of the transplanting arm relative to the planetary rack, greatly simplifying the structural design for various kinds of manipulation.

2 Composition and working principle of 3R open-chain transplanting mechanism

2.1 Structure analysis of 3R open-chain planetary gear train

Mechanisms based on 3R open-chain planetary gear train, also known as double planet carrier planetary gear mechanisms. As shown in Figure 1, it is composed of a 3R open-chain rod set and a non-circular gear set[12-14]. The 3R open-chain rod group has a total of three rotating pairs, and each connecting rod can swing or rotate in a circle driven by a non-circular gear group. The coordination between the two groups of no-circular gear sets makes the design of the end trajectory of such a mechanism possible and also enlarges the transplanting space, which is more suitable for complex manipulation.



1. First planet carrier 2. Second planet carrier 3. Connecting rod 4. Sun gear 5. First middle gear 6. second middle gear 7. Planetary gear 8. Second planet rack driving gear 9. Second planet rack driven gear

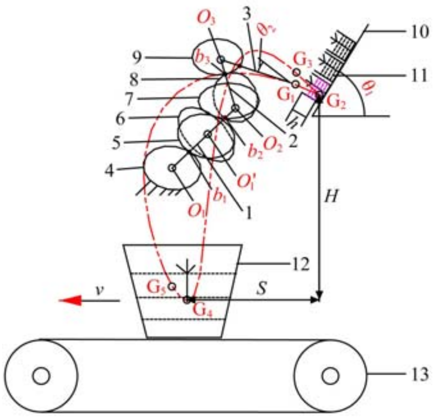
Figure 1 Schematic diagram of double planet carrier planetary gear mechanism

However, the motion of the 3R open-chain rod group is complex and changeable, it is difficult to synchronize the three rotating pairs for integral rotation in one cycle during the design phase. Therefore, a simple structure design is hard to perform seedling manipulation, implying that the complexity of the transplanting mechanism exists.

2.2 Workflow of transplanting mechanism based on open-chain planetary gear train with 3r complete rotation kinematic pair

The workflow of a pot seedling transplanting machine generally includes four steps: taking seedlings, sending seedlings, planting seedlings, and returning trips. The whole process is completed by the coordination of the rack and the transplanting arm[15,16]. The mechanism based on 3R planetary gear train (depicted in Figure 2) consists of two racks and one transplanting arm. The complete rotation kinematic pair means that the first planet carrier should rotate one cycle relative to the ground, while the second planet carrier should rotate one cycle relative to the first planet carrier, and the transplanting arm should rotate one cycle relative to the second planet carrier. In this mechanism, The Angle between the seedling box and the ground is θ_1, which is normally from 50° to 60°. The sun gear is fixed to the ground, the first middle gear engages with the sun gear b_1, the second middle gear engages with the planet gear b_2, and the first planet carrier drives the circular motion around the sun gear. The first middle gear and the second middle gear are hinged at point O_1'. The planetary gear connects to the second planet carrier for driving the second planet carrier to do a single-cycle full rotation relative to the first planet carrier. The driving gear of the second planet carrier connects to the first planetary planet carrier and is hinged at point O_2. The driven gear of the second planet carrier connects to the transplanting arm to enable the transplanting arm to execute a single-cycle full rotation relative to the second planet carrier. The angle between the seedling claw and transplanting arm is θ_2. The transplanting arm can trigger the seedling claw through a cam, so as to complete the capturing and planting manipulation of flower seedlings. During operation, as long as the motor is added to the rotation center O_1, the transplanting mechanism can complete the preset movement trajectory and realize the transplanting. The whole transplanting trajectory contains four stages. G_1G_2 stage is the seedling taking stage, G_2G_4 is the seedling feeding stage, G_4G_5 stage is the planting seedlings stage. G_5G_2 stage is preparing for the seedling taking stage.

The speed of the conveyor belt is corresponding to the distance between flowerpots on belt and the efficiency of transplanting seedling. For example, when the distance between flowerpots on belt is 200 mm, and the efficiency for transplanting one seedling is 40 plants/min, the speed of the conveyor belt is 0.134 m/s.



1. First planet carrier 2. Second planet carrier 3. Transplanting arm 4. Sun gear 5. First middle gear 6. Second middle gear 7. Planetary gear 8. second planet rack driving gear 9. Second planet rack driven gear 10. Seedling delivery mechanism 11. Seedling box 12. Flowerpot 13. Conveyor belt

Figure 2 Schematic diagram of designed transplanting mechanism during operation

3 Mathematical model of designed transplanting mechanism

3.1 Derivation of Burmester curve for 3R open-chain rod group

In this section, a displacement equation will be used to derive the equations of the generalized Burmester curve[17] for the finite separation four-position problem. The plane displacement is depicted in Figure 3.

The two planes of *Bb* and *Mm* rods are π_1 and π_2, respectively. When the two rods move from position 1 to position *i*, it is recorded as from π_1 and π_2 to π_1^i and π_2^i. The specific azimuth is determined by the azimuth α_i of any straight line b_iL_{bi} passing through points b_i on plane π_1^i and by the azimuth β_i of any straight line m_iL_{mi} passing through points m_i on plane π_2^i. Now given the coordinates of m_i (i=1, 2, 3, 4) and b_i (i=1, 2, 3, 4) as well as their azimuths. According to the theory of the rigid-body guidance, the displacement matrices of planes π_1 and π_2 from position 1 to position *i* are $[D_1^{1i}]$ and $[D_2^{1i}]$, respectively.

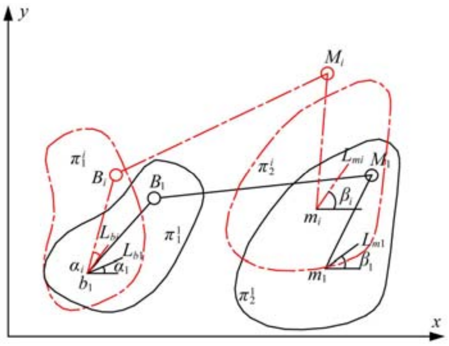


Figure 3 Illustration of rigid-body guidance

Let the hinge points B and M at the first position be $B_1=(x_{B1}, y_{B1})^T$ and $M_1=(x_{M1}, y_{M1})^T$, and the hinge points B and M at the position i be $B_i=(x_{Bi}, y_{Bi})^T$ (mm) and $M_i=(x_{Mi}, y_{Mi})$ (mm). According to the theory of rigid-body guidance, the following equations can be obtained:

$$\begin{bmatrix} B_i \\ 1 \end{bmatrix} = \begin{bmatrix} D_1^{1i} \end{bmatrix} \begin{bmatrix} B_1 \\ 1 \end{bmatrix} \tag{1}$$

$$\begin{bmatrix} M_i \\ 1 \end{bmatrix} = \begin{bmatrix} D_2^{1i} \end{bmatrix} \begin{bmatrix} M_1 \\ 1 \end{bmatrix} \tag{2}$$

According to the constant condition of the rod length, the constraint equation is

$$[B_i-M_i]^T[B_i-M_i]=[B_1-M_1]^T[B_1-M_1] \quad (i=2, 3, 4) \tag{3}$$

It can be obtained by collating:

$$D_i x_{B1} + E_i y_{B1} + F_i = 0, \quad i=2, 3, 4 \tag{4}$$

Take x_{B1} and y_{B1} as unknowns, then the conditions of the above equations construct the following equation

$$\begin{vmatrix} D_2 & E_2 & F_2 \\ D_3 & E_3 & F_3 \\ D_4 & E_4 & F_4 \end{vmatrix} = 0 \tag{5}$$

The cubic equations of x_{M1} and y_{M1} can be obtained by arranging the above equations

$$\begin{aligned} &H_1(x_{M_1}^3 + x_{M_1} y_{M_1}^2) + H_2(y_{M_1}^3 + y_{M_1} x_{M_1}^2) + H_3 x_{M_1}^2 \\ &+ H_4 y_{M_1}^2 + H_5 x_{M_1} y_{M_1} + H_6 x_{M_1} + H_7 y_{M_1} + H_8 = 0 \end{aligned} \tag{6}$$

Finally, the solution to point M is obtained by solving Equation (6), and the solution to point B is obtained by substituting the solution of point M into Equation (4).

3.2 Mathematical model of 3R open-chain rod group and judgment condition of one-cycle complete rotation pair

As shown in Figure 4, the rotation center of the 3R open-chain rod group is denoted as point O, the two intermediate hinge points are denoted as B and M, and the end hinge pose is denoted as P. According to the actual condition of the transplanting mechanism during operation, point O is set as the fixed hinge point, and this point is always located at the origin of the coordinates. Four exact poses P are given by (P_{xi}, P_{yi}, β_i) and the corresponding attitude angle of the OB rod is α_i (i=1, 2, 3, 4).

By substituting the initial conditions into the Burmester curve equation, the solution curves of points B and M are obtained. The coordinates of point B on the solution curve are expressed as a matrix consisting of $[X_b]$ (mm) and $[Y_b]$ (mm), and the coordinates of point M on the solution curve are expressed as $[X_m]$ (mm) and $[Y_m]$ (mm). Then, the lengths of the three bars on the 3R open-chain rod group can be obtained:

$$\begin{cases} L_{ob} = \sqrt{(X_b - O_{x1})^2 + (Y_b - O_{y1})^2} \\ L_{bm} = \sqrt{(X_b - X_m)^2 + (Y_b - Y_m)^2} \\ L_{mp} = \sqrt{(X_m - P_{x1})^2 + (Y_m - P_{y1})^2} \end{cases} \tag{7}$$

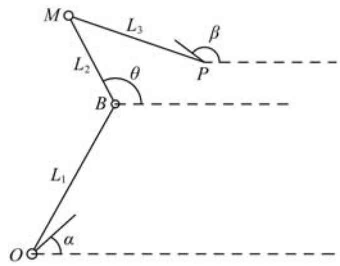


Figure 4 Illustration of 3R open-chain rod group

By solving L_{ob}, L_{bm}, and L_{mp}, which are denoted as L_1, L_2, and L_3, the coordinates of points B and M corresponding to positions 2, 3, and 4 in the initial condition can be calculated by the equations as follows:

$$\begin{cases} B_{xi} = L_1 \cdot \cos(\alpha_i) \\ B_{yi} = L_1 \cdot \sin(\alpha_i) \end{cases}, \quad i=2, 3, 4 \tag{8}$$

$$\begin{cases} M_{xi} = P_{xi} + L_3 \cdot \cos(\beta_i) \\ M_{yi} = P_{yi} + L_3 \cdot \sin(\beta_i) \end{cases}, \quad i=2, 3, 4 \tag{9}$$

The angle θ of point B can be obtained by using the following inverse trigonometric equation:

$$\theta_i = a\cos\frac{(M_{xi} - B_{yi})}{\sqrt{(M_{xi} - B_{xi})^2 + (M_{yi} - B_{yi})^2}}, \quad i=2, 3, 4 \tag{10}$$

To ensure the monotonicity of the angle change of rotating pairs, this study designed a monotone filtering algorithm for the parameters in the solution domain, so as to design the parameters for the transplanting mechanism based on a double planetary rack gear train with single-cycle full rotating pairs. The specific steps of the monotone filtering algorithm are described as follows: first of all, four initial angles of the first and the last two points are constrained. With a rotation of 360° as a period, these four angles must satisfy the angles of point O (i.e., $\alpha_1<\alpha_2<\alpha_3<\alpha_4$ or $\alpha_1>\alpha_2>\alpha_3>\alpha_4$) and point P (i.e., $\beta_1<\beta_2<\beta_3<\beta_4$ or $\beta_1>\beta_2>\beta_3>\beta_4$). When the angle condition is met, the initial condition is substituted into the Burmester curve equation to solve the angle of the rotation pair B (i.e., θ) to meet the monotone condition. Then, the solution is substituted into the next calculation. The B-spline curve fitting can be carried out for the solution to obtain the angle change of 360° in a cycle. Finally, N angles will be inserted between the four angles that originally meet the monotony for generating a new angle matrix. That is, the third judgment of the angle monotony can be made for the new angle matrix. The flowchart for the monotone filtering algorithm is shown in Figure 5.

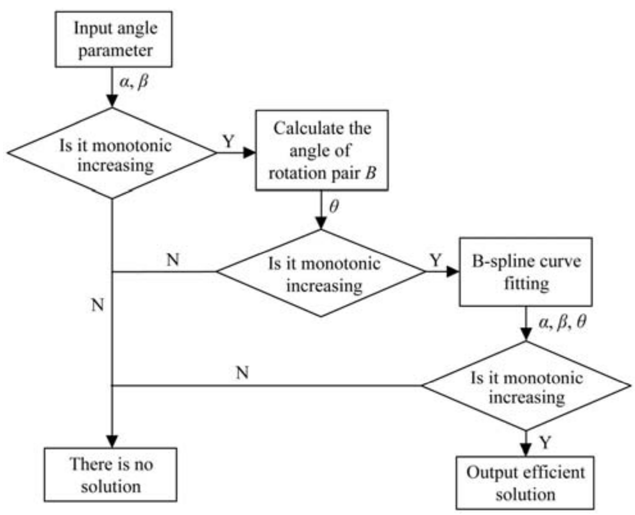


Figure 5 Flowchart of monotone filtering algorithm

4 Compound design of 3R open-chain rod group and non-circular gear train mechanism

4.1 Obtain the transmission ratio between each planet carrier

In order to calculate the angle change within a cycle, the absolute angle between the first planet carrier point O in the 3R open-chain rod group and the positive direction of the X-axis is set as φ_{1i}, where the initial angle is $\varphi^{1''}$. The absolute angle between point B (on the second planet carrier) and the X-axis is set as φ_{2i}, where, the initial angle is φ^2. The absolute angle between the end point P (on the transplanting arm) and the X-axis is set as φ_{3i}, where the initial angle is φ^3. The corresponding calculation can be realized as follows:

1) The relative rotation angle of the first planet carrier around point O is

$$\varphi_1^i = \varphi_i - \varphi_1 , \quad i=1,2,3,4 \tag{11}$$

2) The relative rotation angle of the second planet carrier with respect to the first planet carrier is

$$\varphi_{12}^i = (\varphi_i - \varphi_2) + \varphi_1^i , \quad i=1,2,3,4 \tag{12}$$

3) The relative rotation angle between the transplanting arm and the second planet carrier is

$$\varphi_3^i = (\varphi_{3i} - \varphi^3) - \varphi_{2i} , \quad i=1,2,3,4 \tag{13}$$

Let the end point and the first point be the same, but the abscissa is 360° apart. Then, the angle $\varphi_1{}^i$, $\varphi_2{}^i$, $\varphi_3{}^i$, and the end point are brought into the B-spline curve fitting equation for obtaining the single period angle fitting curve (as shown in Figure 6).

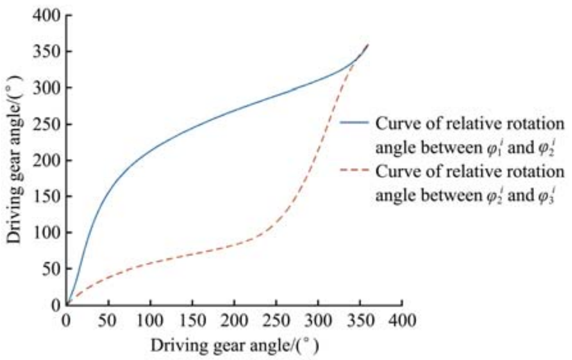


Figure 6 Angle fitting curve in one cycle

Now it is stipulated that the counterclockwise rotation is in the positive direction. According to the relative rotation angle φ_1, φ_2, and φ_3 calculated in the figure, the equation of the transmission ratio between the first planet carrier and the second planet carrier can be obtained by Equation (14).

$$i_1 = \frac{n_1}{n_2} = \frac{\mathrm{d}(\varphi_1)}{\mathrm{d}(\varphi_2)} \tag{14}$$

Similarly, the formula of transmission ratio between the second planet carrier and the transplanting arm is

$$i_2 = \frac{n_2}{n_3} = \frac{\mathrm{d}(\varphi_2)}{\mathrm{d}(\varphi_3)} \tag{15}$$

Through Equations (12) and (13), the relation curve of the transmission ratio between each planet carrier of the 3R transplanting mechanism can be obtained, as shown in Figure 7.

4.2 Solution to non-circular gear transmission ratio

Considering the problem of the mechanical interference and the machining of the non-circular gear pitch curve shape, the first rack adopts two-stage non-circular gear transmission, and the second rack adopts first-stage gear transmission. In order to make the two stages of non-circular gears in the first rack have similar

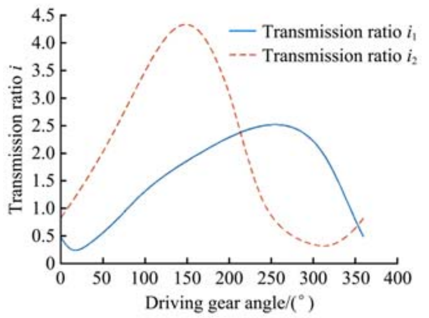


Figure 7 Relation curve of transmission ratio

mechanical and kinematic properties, the distribution scheme in this paper takes $i_1^1 = i_1^2 = \sqrt{i_1}$. Meanwhile, an adjustment coefficient t is introduced to ensure that the pitch curve of the driving wheel has the same length as that of the driving wheel. After an adjustment, the first-stage transmission ratio is $i_1^{1\prime} = i_1^1 / t$, $t = \dfrac{2\pi}{\int_0^{2\pi} \frac{1}{i_1} \mathrm{d}\varphi_1}$, and the second-stage transmission ratio is $i_1^{2\prime} = i_1 / i_1^{1\prime}$. The transmission ratio after the distribution is shown in Figure 8.

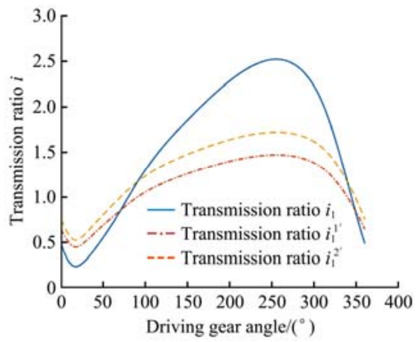


Figure 8 Transmission ratio distribution curve of first planet carrier

According to the obtained transmission ratio, the solution curve of the non-circular gear can be calculated by the expression of the pole diameter and the angle[18].

5 Optimization of mechanism parameters

5.1 Characteristics of transplanting trajectory and kinematic targets

Based on the previous research experience and combined with the advantages and disadvantages of different transplanting tracks and experimental results[19,20], the transplanting trajectory selected in this paper is an "8" shape trajectory as shown in Figure 2. In order to make the transplanting mechanism designed to better meet the needs of flower transplanting, this paper puts forward the following objectives of parameter optimization and design requirements for the transplanting trajectory:

1) The trajectory of the seedling taking stage (G_1G_2) enters from the lower part of the seedling, and the angle between the needle and the bowl wall should not be greater than 5° during taking out.

2) In order to protect the seedlings, the needle must clamp the pot seedlings, and the depth of the needle to the bowl plate h is 30-35 mm.

3) For the seedling feeding stage (G_2G_4), in order to prevent the transplanting arm from touching the seedling box, the trajectory at this stage is designed to be far away from the box enough.

4) For planting seedlings (G_4G_5), in order to ensure the straight strength of the seedlings and avoid interference, the planting angle θ_2 is designed to be about 90°.

5) In order to achieve larger displacement, the transplanting height H is greater than 350 mm, and the transverse distance S is about 100 mm.

5.2 Parameter optimization for transplanting trajectory

Four poses P_1, P_2, P_3, and P_4 are selected on the transplanting trajectory according to the requirements of the kinematics target. The specific data are listed in Table 1. In addition, the corresponding angles of point O were selected as (–50°, 30°, 84°, –110°) according to the angle relation of forward solution in ADAMS.

Table 1 Four exact poses of transplanting trajectory

poses	(x_i, y_i)/mm	θ/(°)
P_1	(−115, −140)	−13
P_2	(212, 100)	150
P_3	(98, 126)	145
P_4	(−40, −300)	92

The above data are then fed into a 3R bar open-chain planetary gear train model design software based on Matlab developed by us, where a series of solutions satisfying the given conditions can be obtained.

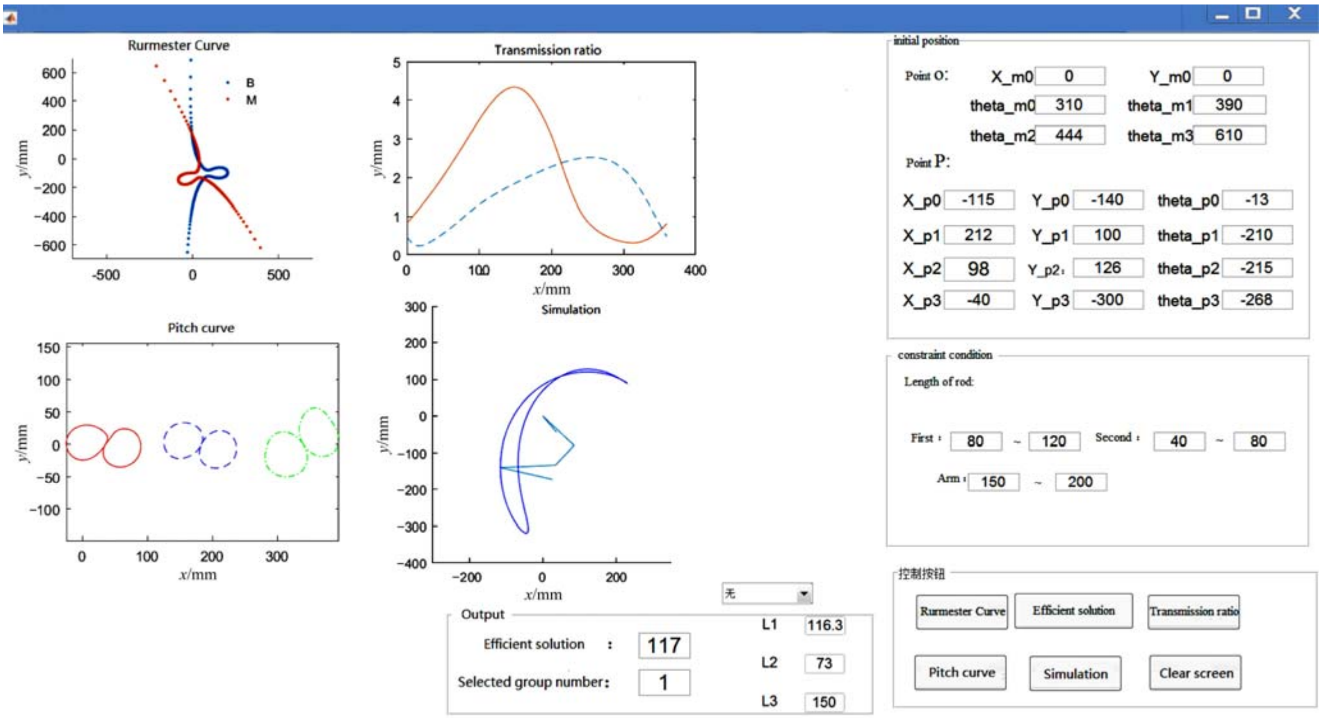


Figure 9 Interface of Transplanting trajectory optimization software

6 Simulation and experiments

6.1 Simulation

After the aforementioned optimization, a group of optimal solutions (L_1=116.3 mm, L_2=73 mm, L_3=150 mm) are selected for the design and simulation of the 3R transplanting mechanism. The motion trajectory of the simulation is shown in Figure 10. It shows that the simulated trajectory is basically consistent with the theoretical trajectory, which proves the feasibility of the mechanism design idea and the proposed method in this study, and also provides a reference and support for the subsequent processing and assembly of the physical prototype.

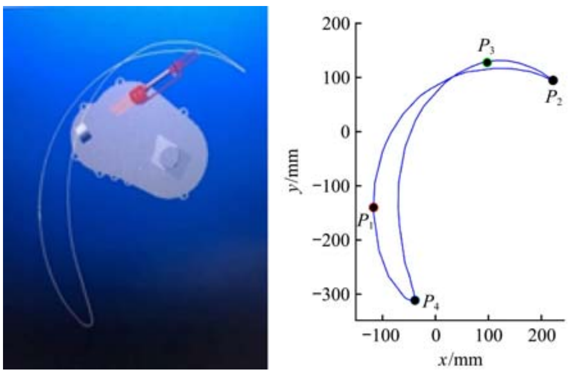


Figure 10 Schematic diagram of static trajectory

6.2 Experiments

6.2.1 Idling Testing

The recorded video stream during the testing is imported into the Adobe After Effect CC2017 analysis software. By converting the video stream to more than 100 images and connecting the end position of the seedling needle in each image, the one-cycle trajectory can be obtained, as shown in Figure 11a.

Since the end of the seedling needle will shrink during actual rotation, the simulation trajectory obtained after the seedling needle is also set to shrink in ADAMS, as shown in Figure 11b. Compared with the two trajectories, these trajectories are basically identical, which proves the feasibility of our transplanting mechanism design, parts processing, and testbed construction.

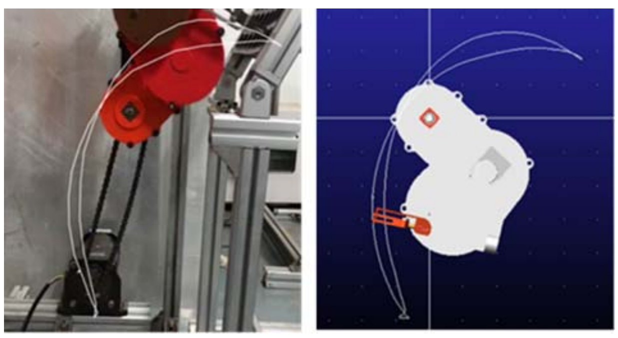


Figure 11 Comparison of actual and simulated trajectories

6.2.2 Transplanting Testing

In the setup of this testing, kale seedlings are planted in the seedling raising base, where the seedling ages are about 20 d, and the seedling heights are about 80-120 mm. The size of the pot seedling is 30 mm×30 mm×44 mm. The planting depth of seedling in the soil is 35mm. During the testing, the motor speed was adjusted to 40 r/min. The efficiency for transplanting one seedling is 40 plants/min. After recording the transplanting process, the obtained video stream is processed by image analysis software to obtain the actual execution of the four poses, as shown in Figure 12, where the angles of the four poses are obtained by measurement as –10.5°, 152°, 145°, and 92°. Compared with the initially given angle in Table 1, all of the error values are less than 3°, which is within the error tolerance range. The actual transplanting height is measured as 397 mm. Therefore, it is considered that the attitude of the seedling satisfies the expectation. Meanwhile, it also verifies the rationality of the design method of the 3R complete rotation pair transplanting mechanism.

A row of eight holes is used as a group in the testing, and 80 groups of runs are carried out, the success rate of the mechanism for transplanting is 90.625%. According to the results obtained by the transplanting mechanism prototype in the experimental platform, both the generated trajectory attitude target and the transplanting success rate meet the requirements.

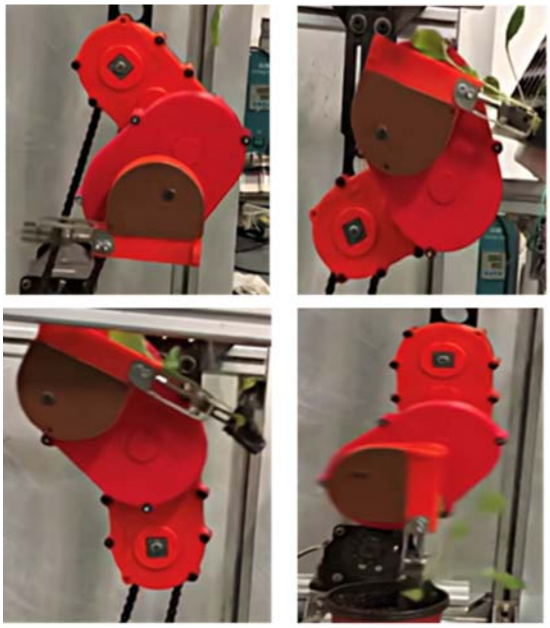


Figure 12 Four poses during transplanting

7 Conclusions

1) Aiming at the shortcomings of the existing 2R and 3R planetary gear train mechanisms, a transplanting mechanism based on a 3R complete rotation pair gear train was proposed. Based on the rigid body guiding theory of the mechanism, the expression of the Burmester curve equation with four poses of finite separation is deduced, the characteristics of the angle in the motion of the linkage group are analyzed, and then the judgment conditions of the complete rotation pair are also established. The 3R open-chain rod group model that satisfies the condition can be obtained by condition determination.

2) The relationship between the rotation angle of the rod group and the pitch curve of the non-circular gear group was established. A set of optimal parameters are obtained through transplanting trajectory optimization software, and the optimized parameters are L_1=116.3 mm, L_2=73 mm, and L_3=150 mm. Then, according to the optimal parameters, the three-dimensional model of the transplanting mechanism is designed and imported into ADAMS software for motion simulation, and the end trajectory of the transplanting arm is obtained. Through comparison, it is found that the simulated trajectory conforms to the design of the theoretical trajectory, which validates the proposed method.

3) Through idling testing, the angles of –10.5°, 152°, 147.5°, and 93° at four precise poses are compared with the initial conditions, where the error is found to be less than 3°, within the tolerance range. Besides, the transplanting testing shows an average success rate of 90.625% for seedling transplanting. Therefore, the feasibility of the proposed method for the designed transplanting mechanism, the 3R single-cycle integral gear train based on solution domain synthesis, can be validated.

Acknowledgements

The authors acknowledge that this work was financially supported by the National Natural Science Foundation of China (Grant No. 32071909, 51975536), the Key Research Projects of Zhejiang Province (Grant No. 2022C02002, 2021C02021), the Basic Public Welfare Research Projects of Zhejiang Province (Grant No. LGN20E050006), the Shanghai Science and technology agricultural Development Project (2021 No 4-1), and the General Project of Agriculture and Social Development in Hangzhou (Grant No. 20201203B92).

[References]

[1] Yang Z N. Current situation and development of flower gardening production and application. Journal of Green Science and Technology, 2019; 17: 142–143. (in Chinese)

[2] Yang Y. Development status, trend and strategy of Chinese flower industry. Modern Horticulture, 2019; 11: 44–45. (in Chinese)

[3] Zhou M F, Huang Z J, Shi L D, Yu G, Zhao X, Liu P. Improvement design and experimental analysis of automatic transplanting mechanism for flower plug seedlings. Journal of Chinese Agricultural Mechanization, 2019; 40 (5): 11–16. (in Chinese)

[4] Choi W C, Kim D C, Ryu I H, Kim K U. Development of a seedling pick-up device for vegetable transplanters. Transactions of the ASAE, 2002; 45(l): 13–19.

[5] Liang X, Li Z, Qi W, Zhao Y. Optimal design and development of a double-crank potted rice seedling transplanting mechanism. Transactions of the ASAE, 2017; 60(1): 31–40.

[6] Sun L, Xu Y D, Huang H M, Wang Z F Zhang G F. Analysis of planetary gear train transplanting mechanism based on pitch curve convexity discrimination. Transactions of the CSAM, 2018; 49(12): 90–99. (in Chinese)

[7] Guo S C, Li D Q, Shang Z Q, Zhang C P, Wei S. Structural design of noncircular planetary gear based on MATLAB. Heavy Machinery, 2016; 4(9): 49–52. (in Chinese)

[8] Chen X. Structural synthesis and application of gear linkage transplanting mechanism based on graph thory. Master's dissertation. Hangzhou: Zhejiang Sci-Tech University, 2018; 79p.

[9] Zhao X, Liao H W, Ma X X, Dai L, Yu G H, Chen J N. Design and experiment of double planet carrier planetary gear flower transplanting mechanism. Int J Agric & Biol Eng, 2021; 14(2): 55–61.

[10] Zhao X, Ye J, Chu M Y, Dai L Chen J N. Automatic scallion seedling feeding mechanism with an asymmetrical high-order transmission gear train. Chinese Journal of Mechanical Engineering, 2020; 33(12): 4690–4694. (in Chinese)

[11] Sun L, Shen J H, Zhou Y Z. Design of vegetable pot seedling transplanting mechanism with non-circular gear and connecting rod combination transmission. Transactions of the CSAE, 2019; 35(10): 26–33. (in Chinese)

[12] Zhou M F, Xu J J, Tong J H, Yu G H, Zhao X. Design and experiment of an integrated automatic transplanting mechanism for flower plug seedlings. Transactions of the CSAE, 2018; 34(20): 44–51. (in Chinese)

[13] Ye B L, Li L, Yu G H, Liu A, Zhao Y. Dynamics analysis and experiment

of vegetable bowl seedling rotary picking mechanism. Transactions of the CSAM, 2014; 45(6): 70–78. (in Chinese)
[14] Yu G H, Jin Y, Chang S S, Ye B L, Gu J B. Design and experiment of rice seedling transplanting mechanism with bowl type. Transactions of the CSAM, 2019; 50: 100–108. (in Chinese)
[15] Wang L, Sun L, Huang H M, Yu Y X, Yu G H. Design of clamping-pot-type planetary gear train transplanting mechanism for rice wide–narrow-row planting. Int J Agric & Biol Eng, 2021; 14(2): 62–71.
[16] Wu G H, Yu G H, Ye B L, Yu Y X. Study on forward and reverse design method of rice seedling transplanting mechanism with planetary gear train. Transactions of the CSAM, 2020; 51(2): 86–102. (in Chinese)
[17] Han J Y, Cui G Z, Yang T. Solution domain synthesis theory and method for four position motion generation of six bar linkage. Journal of BUAA, 2014; 40(9): 1170–1175. (in Chinese)
[18] Wang Y Z, Chen B, Hu C B, Zhang S T, Li T, Liu Y P. Design of third-order non-circular planetary gear. Advanced Materials Research, 2012; 482-484: 305–308.
[19] Sun L, Xing Z Q, Xu Y D, Liu B, Yu G H, Wu C Y. Research on transplanting mechanism of rice pot seedling based on precise multi position analysis. Transactions of the CSAM, 2019; 50 (9): 79–86. (in Chinese)
[20] Jin X, Li D Y, Ma H, Ji J T, Zhao K X, Pang J. Development of single row automatic transplanting device for potted vegetable seedlings. Int J Agric & Biol Eng, 2018; 11(3): 67–75.