Epi-and hypo-cyclic spindle drives of a cotton harvester

A A Rizaev¹, D A Kuldoshev¹, N B Dzhuraeva¹, M K Normatov²

¹ Academy of Sciences of the Republic of Uzbekistan, 100125, Tashkent, Uzbekistan
² Tashkent State Technical University named after Islam Karimov, 100095, Tashkent, Uzbekistan

E-mail: rizayev52@mail.ru

Abstract. The article is devoted to the development of a more efficient technological scheme of a vertical-spindle cotton harvester with pair wise and sequentially located spindle drums. The analysis of the existing technological process of this harvesting device with hypo- and epicyclic spindle drives is conducted. The fact of insufficient orientation of the active surface of the spindle to the cotton capture from cotton plants when it enters the working chamber is stated. On the basis of a priori information, the process of removing cotton from the spindles is studied when the direction of rotation of the spindles is changed; and a technological scheme with epi-and hypo-cyclic planetary drives for the rollers of the working bodies is proposed. In order to model the process in the working chamber of the apparatus, a calculation scheme and analytical expressions were drawn up to determine the values and directions of the vectors of the absolute and active velocities of the spindle teeth since the activity of the spindle surface is one of the main criteria for assessing the performance of a cotton harvester during its production.

1. Introduction

As it is known, the selective and quality cotton-picking from open cotton bolls by a harvester with a vertical spindle requires reversible rotational movements of the spindle. The spindle teeth in the working chamber of the harvesting device move along a hypo-cyclic trajectory, and in the zone of cotton removal from the spindle, the spindle performs a reversible epi-cyclic rotational movement [1].

The capture and picking of cotton by spindle harvesting devices are performed only under a certain deformation of the porous cotton boll by the spindle and the active state of the spindle surface. By M. V. Sablikov definition [2], the activity of the spindle surface during cotton picking has coincidences within certain limits of the wedge-shaped tooth tip with the direction of the vector of spindle absolute velocity at a given point. The limiting values of the angle at which the spindle tooth is in the active state are: \( \beta_1 = 90^\circ - \varphi_f \), \( \beta_2 = 90^\circ - \varphi_f - \kappa \), where \( \kappa \) is the point angle of the tooth tip of the spindle, \( \varphi_f \) is the angle of friction between the cotton and the surface of the spindle tooth, \( \beta_1 \), and \( \beta_2 \) are the upper and lower boundaries of the angle of tooth activity. The design of the spindles and the kinematic modes of operation of the harvesting devices of modern horizontal and vertical-spindle machines are mainly based on the above criteria.
To ensure high productivity, reliability and stability of the cotton harvester operation, research and innovation studies were widely conducted by the designers and manufacturers of these machines [3-11].

Modern semi-hitchhacked tractor-mounted cotton pickers with vertical spindles are widely used in farms due to the simplicity of design, low energy and material consumption, and mobility. A number of cotton-growing countries are currently working on improving machine designs [1-4, 7-9]. Let us dwell on the features of the technological process in a harvesting device with sequentially located pairs of vertical-spindle drums.

1. Pairs of counter-rotating drums (planetary gear carrier) with spindles and mechanisms installed along their generatrices, roll over and press out cotton plants with bolls to the size of a slot (24-40 mm) in the working chamber. At the same time, the rollers (planetary gear satellites), pressed into the spindle stems, roll over the outer flexible belts, providing points on the surfaces of the spindles with a hypo-cyclic trajectory in order to capture cotton. But this rotation is carried out counter the drum rotation and contributes to the bending of the cotton plants in the direction of machine motion. The rotating drum brings the spindle with cotton out of the working chamber, and the rollers lose their connection with the belts, continuing hypo-cyclic rotation by inertia.

2. The drum brings the spindle to the removal zone, where the spindle roller is decelerated, hitting the reverse rotation block (epicyclic planetary gear), and there the cotton is removed from the spindle surface under the action of inertial force. Further, due to the spindle reversal, the bulk of the cotton (more than 75 ... 80%) is self-thrown and the brushes of the rotating doffer, capturing the cotton, throw it into the receiving chamber of the harvesting device.

3. The spindle, freed from the cotton, along with the drum in its orbit, goes into the working chamber and the picking cycle is repeated.

Due to the counter-directed rotations of the spindles and the drum, cotton plants at the beginning of the working chamber receive oppositely directed velocities. In the narrowest part of the chamber, where the slot is 30 ... 36 mm, the angular velocity of the spindle decreases to 10 ... 15% due to the braking effect of the cotton plants, which negatively affects the activity of the working body.

To determine the orientation of the active surface of the spindle in the working chamber, M V Sablikov in his book [2] writes: “...when the plants pass through the working chamber difficulties arise due to the bending of plants, insufficient capture of cotton in the front of the working chamber. Therefore, it is of great interest to change the rotation direction of the spindles, that is, their rotation in the same direction as the drum rotates”. The activity of the surface of the working body M V Sablikov investigated by the method of graphic plotting and came to the following conclusion “...the active part of the surface when the spindle passes in the front of the working chamber is approximately in the area of possible contact between the spindle and the boll”. Further, he argues: “the disadvantage of the changed direction of rotation is a significant deterioration in the cotton pick-up from the spindles with conventional doffers ... the cotton, knocked down by the brushes on the spindles, which have already passed past the doffer, is thrown, which inevitably leads to a significant deterioration in the doffers operation”. However, he did not take into account the process of self-throwing of cotton from a spindle with an increased diameter due to its reverse and considered cotton removal only along the spindle teeth.

The study by Z. Kh. Izzatov [12] is devoted to the investigation of cotton removal at a similar direction of the drum and spindle rotation in the technological cycle. The technological scheme of the harvesting device included: hypo-epi-cyclic, hypo-cyclic, epi-cyclic (in these options, cotton picking was carried out without spindle reverse), epi-hypo-cyclic spindle drives. In this case, the cotton was removed from the spindles both in the direction of the teeth and in opposite direction. The efficiency of the doffer was evaluated by the coefficient of cotton winding on the spindle, processed with brushes. The amount of cotton removed from the surface of the spindle and the quality of the removed cotton and fiber were determined on the laboratory bench. According to his research, both existing and epi- and hypo-cyclic technological schemes of the harvesting apparatus are rational. But the tests of the harvesting device were not conducted in the field laboratory conditions. The use of the kinematic
scheme of a pair of spindle drums operation with hypo-and epi-cyclic and epi- and hypo-cyclic drives of the spindles and their consecutive counter-arrangement in the harvesting apparatus were considered by the researchers of the Institute of Mechanics and Seismic Stability of Structures of the Academy of Sciences of the Republic of Uzbekistan (IMSS ASRUz) and the State Specialized Design Bureau (GSKB) for machines in cotton growing [13]. Unfortunately, it was not brought to extensive test trials and required further research to improve the operation of the apparatus in the cotton removal zone.

The above analytical review on the study of the technological scheme of the vertical-spindle harvesting apparatus allows concluding that there are reserves for the modernization and improvement of the technical performance of the cotton picker with sequentially located pairs of spindle drums by changing the direction of rotation of the spindles in the chamber of the apparatus.

2. Methods
The materials of the article are based on analytical methods for analyzing the existing design and the development of an efficient technological cycle for the operation of a vertical-spindle picking apparatus, taking into account the change in the kinematic diagram of the drive of the spindle rollers, that is, the epi-cyclic planetary transmission of motion in the working chamber. On the basis of the well-known methods of vector algebra and analytical mechanics, analytical expressions for calculating the values and directions of the absolute and active velocities of the spindle teeth, necessary for assessing the performance of the harvesting device, were developed for this scheme.

3. Results and Discussion
Figure 1 shows a diagram of a cotton picker with a vertical spindle containing several pairs of counter-rotating drums with spindles and drives mechanisms for their rotation installed along their generatrices; drums and spindles in the working chamber are installed with the ability to rotate in the same direction, and when removing cotton from the spindles in the removal zone - with the ability to rotate in the opposite direction. In addition, a solution is proposed, when the spindles are mounted with the ability to rotate in the opposite direction on each subsequent pair of drums (after the first one). In this case, the rollers of the epi-cyclic planetary drive in the working chamber are protected from the outside by a stationary rigid cylindrical arc against cotton and plant elements falling between the rows of spindle drive belts, by the rotating rollers.

The device consists of a plant lifter 1, a plant guide 2, a front pair of drums 3 with spindles 4 and their drive rollers 5, an epi-cyclic V-belt drive arc 6 (block), a hypo-cyclic belt arc drive 7 (belts), a pair of doffers 8, a rear pair of drums 9 with spindles and their doffers 10, the side door of the apparatus 11, the receiving chamber 12. When the cotton picker moves along the rows of cotton plants, the front 3 and rear 9 spindle drums and their spindles 4 with rollers 5, located pair wise, make counter-rotational movements with angular velocities \( \omega_{dr} \) and \( \omega_{spi} \) in the working chamber of the apparatus due to the epi-cyclic drive 6. As a result, plants with bolls are more readily captured by spindle drums without additional inclination in the direction of machine travel. Cotton tufts from the open bolls are reliably captured by the spindle teeth and wound onto its surface. Further, drums 3, 9 take the spindles 4 with cotton out of the working chamber, and bring them into the removal zone of the apparatus. Before the contact with doffers 7 and 9, the drive rollers of spindles 4, contacting with the hypo-cyclic belt arc (drive), change the direction of rotation (to the reverse one) and this contributes to self-resetting and separation of the cotton wound on the surface of the spindles, and the brush doffers, capturing cotton, transfer it at a speed 4-5 times faster than the spindle speed into the narrow corridor between doffers 8, 10 and the side door 11 of the apparatus. Subsequently, the cotton is directed to the receiving chamber 12, from where it is blown into machine hopper by the suction air flow. The successful implementation of the technological process in spindle cotton harvesters is directly related to their geometric, kinematic and dynamic parameters. Therefore, the calculation of the operating modes of the apparatus for a given technological scheme ultimately determines the quality of cotton picking.
1 - plant receiver, 2 - side shields, 3 - drums, 4 - spindle, 5 - roller, 6 - block, 7 - belt, 8 - doffers, 9 - rear drum, 10 - rear doffer, 11 - side door, 12 - receiving chamber

Figure 1. Technological diagram of a vertical-spindle apparatus with epi- and hypo-cyclic spindle drive

The considered epi- and hypo-cyclic planetary motion of a vertically located spindle with specially located teeth on the surface can be modeled based on the methods of the kinematics of plane motion of a rigid body. Previously, similar modeling was performed by the authors in [14, 15]. In this study, we consider the movement of any given point on the spindle relative to a stationary cotton boll located on the lower layer of the plant. For this, we use four interconnected systems of Cartesian coordinates, as in [14]. Figure 2 shows the interconnected Cartesian coordinates, where the first stationary coordinate system $O_0\xi\eta\zeta$ is related to the ground on which a cotton plant with open bolls is located, the second coordinate system $O_1X_1Y_1Z_1$ is related to a cotton harvester moving at a given speed $V_m$ along the fixed axis $O_0\xi$, the third system $O_2X_2Y_2Z_2$ is related to the axis of rotation of the drum (planetary carrier) and it rotates at angular velocity $\omega_{dr}$ relative to system $O_3X_3Y_3Z_3$, and the fourth coordinate system $O_4X_4Y_4Z_4$ is related to the axis of rotation of the spindle located at a distance $R_4$ from the center of rotation of the drum, and the spindle rotates about the axis $O_4Z_4$ at angular velocity $\omega_{sp}$, and also rotates with the drum about the axis $O_2Z_2$ at angular velocity $\omega_{dr}$. Consequently, the spindle performs complex movements: two rotations - about its own axis and the drum axis, and moves with the machine along the axis $O_0\xi$ at velocity $V_m$.

The approved designs scheme (Figure 2) allows deriving the equation of motion for any given point $A$ on the spindle surface, determined by the radius of vector $\rho$ in the following form:

$$\rho = x + R_{dr} + r + z,$$ \quad (1)
where \( \vec{x} \) is the vector that determines the position of the harvesting device (coordinate system \( O_1X_1Y_1Z_1 \)) relative to the origin, that is, to the ground \( (O\xi\eta\zeta) \), \( \lVert \vec{x} \rVert = V_m \cdot t \), m;

\( \vec{R}_{dr} \) is the vector that determines the position of the spindle center (coordinate system \( O_2X_2Y_2Z_2 \)) relative to the drum axis \( O_2Z_2 \), that is \( \lVert \vec{R}_{dr} \rVert = R_{dr} \) is the drum radius, m;

\( \vec{r} \) is the vector that determines the position of point A on the cross-section of the spindle in the coordinate system \( O_3X_3Y_3 Z_3 \), that is \( \lVert \vec{r} \rVert = r \) is the radius of the spindle, m;

\( \vec{z} \) - is the vector that determines the position of point A along the spindle surface in the system \( O_3X_3Y_3 \), \( \lVert \vec{z} \rVert = z_s \), m.

Let us proceed to the definition of the projection of the vector equation (1) on the coordinate axes \( \Omega \). To do this, first, we define the position of point A in the moving coordinates in the following form:

\[
\begin{align*}
\xi &= r \cdot \cos \varphi \\
\eta &= r \cdot \sin \varphi \\
z_1 &= z_s
\end{align*}
\]

(2)

where \( r \) - is the radius of the spindle teeth tip, m;

\( \varphi \) is the angle between the axis \( O_1X_1 \) and vector \( \vec{r} \), that is, the initial position of the considered point A relative to the axis \( O_1X_1 \), rad;

\( z_s \) is the initial position of the considered point A relative to the axis \( O_1Z_1 \), m.

Further, on the basis of the well-known method of coordinates trans form, the position of point A is considered in coordinates \( O_2X_2Y_2Z_2, O_3X_3Y_3Z_3, O_2\xi\eta\zeta \), the method of their description is given in [14]. Based on this, we write down the position of the considered point A in coordinates \( O_2\xi\eta\zeta \) in the following form:

\[
\begin{align*}
\xi &= R_{dr} \cos (\psi_0 + \omega_{dr} t) + r \cos (\varphi + \Omega t) + V_m \\
\eta &= R_{dr} \sin (\psi_0 + \omega_{dr} t) + r \sin (\varphi + \Omega t) \\
z_1 &= z_s
\end{align*}
\]

(3)

where \( \psi_0 \) - is the angle between the axis \( O_2X_2 \) and vector \( \vec{R}_{dr} \), that is, the angle that determines the initial position of the considered point A relative to the drum axis \( O_2X_2 \), rad. \( (\psi_0 = const) \);

\( \omega_{dr} \) is the angular velocity of rotation of the drum around the axis \( O_2Z_2 \), \( s^{-1} \) (\( \omega_{sp} = const \));

\( \Omega = \omega_{sp1} + \omega_{dr} \) is the absolute angular velocity of rotation of the spindle relative to the axis \( O_2\xi \), \( s^{-1} \) ("+" sign - for epi-cyclic trajectory of points on the spindle, sign "-" - for hypo-cyclic trajectory of points on the spindle [14, 15]);

\( t \) is current time, s;

\( \omega_{dr} \) is the angular velocity of the spindle around the axis \( O_2X_2 \), \( s^{-1} \) (\( \omega_{sp} = const \)).
Differentiating the system of equations (3) in time, we determine the projection of the absolute velocity vector of point A on the axis $O\xi$, $O\eta$, $O\zeta$ in the coordinate system $O\xi\eta\zeta$ in the following form:

$$V_{O\xi} = V_m - r\Omega \sin(\varphi + \Omega t) - R_{dr} \omega_d \sin(\psi_0 + \omega_d t)$$
$$V_{O\eta} = r\Omega \cos(\varphi + \Omega t) + R_{dr} \omega_d \cos(\psi_0 + \omega_d t)$$
$$V_{O\zeta} = 0$$
(4)

Based on the method of kinematics of a point performing a complex movement, we determine the modulus of the absolute velocity vector of point A on the spindle surface in the form:

$$V_a = \sqrt{V^2 + U^2 + V_m^2 - 2[V_m U \cos(\omega_{sp1} t + \varphi) + V_m U \sin(\Omega t + \varphi) + V_m U \sin(\omega_{dr} t + \psi_0)]}$$
(5)

$V = r \cdot \Omega$ is the linear velocity of point A on the spindle, m/s;

$U = R_{dr} \cdot \omega_d$ is the linear velocity of the drum along the circle of spindle location, m/s.

It is known from vector algebra and theoretical mechanics that a vector is characterized by both magnitude and direction determined by the cosines of the direction angles with respect to the accepted coordinates. Proceeding from this, we determine the magnitude of the absolute velocity of point A on the spindle from equation (5); the direction of vector $V_a$ is determined relative to the tangent to the circle of the spindle section since the spindle surface captures the cotton tufts by teeth. In [14], a method for calculating normal $V_{an}$ and tangential $V_{at}$ components of $V_a$ is given, and on its basis the angle between vector $V_a$ and the tangent to the circle of the spindle section is determined taking into account its epi-cyclic motion in the following form:

$$\tan \beta = \frac{V_{an}}{V_{at}} = \frac{V_m \cos(\Omega t + \varphi) - U \sin(\omega_{sp1} t + \psi_0 + \varphi)}{V - V_m \sin(\Omega t + \varphi) - U \cos(\omega_{sp1} t + \psi_0 + \varphi)}.$$  
(6)

By M V Sablikov [2] definition, an active component of the absolute velocity vector $V_a$ ensures the capture of cotton tufts from the bolls. He proposed a technique for graphical plotting of diagrams of active velocities on the spindle surface in the working chamber of the harvesting apparatus; using an oblique projection $V_a$ on the direction of the bisector of the spindle tooth point angle at an angle $90^\circ - \varphi$, where $\varphi = \arctg f$, $f$ is the coefficient of friction between the cotton and the spindle.

In contrast to this technique, we have developed an analytical method for determining the active component of the tooth absolute velocity vector, taking into account the oblique projection $V_a$ on the bisector of the tooth point angle in the form [14, 15]:

$$V_{at} = \kappa \cdot V_a$$
(7)

where $\kappa$ is the coefficient of the direction angles, determined by the oblique projection of vector $V_a$ on the bisector of the tooth point angle, and determined by the following system:
\[
\kappa = \begin{cases} 
\pm \cos(\beta - \theta) - \sin(\beta - \theta) \cdot t g (\chi + \varphi_i) & \text{at } \beta \geq \theta \text{ and } (+) \text{ at } V_{ar}, V_{an} > 0, \\
\quad (-) \text{ at } V_{ar}, V_{an} < 0; \\
\pm \cos(\theta - \beta) - \sin(\theta - \beta) \cdot t g (\chi + \varphi_i) & \text{at } \beta < \theta \text{ and } (+) \text{ at } V_{ar}, V_{an} > 0, \\
\quad (-) \text{ at } V_{ar}, V_{an} < 0; \\
\pm \cos(\theta + |\beta|) - \sin(\theta + |\beta|) \cdot t g (\chi + \varphi_i) & \beta \leq 0 \text{ and } (+) \text{ at } V_{ar} > 0, V_{an} < 0, \\
\quad (-) \text{ at } V_{ar} < 0; V_{an} > 0; 
\end{cases}
\]

(8)

where \( \theta \) is the angle between the bisector of the tooth point angle and the tangent to the circle of the spindle section, rad;
2 \( \chi \) is the angle of the spindle tooth point, rad;
\( \varphi_i \) is the angle of friction between the cotton and the spindle material, rad.

The analytical expressions (5) - (8) obtained, make it possible to determine the values and directions of the vectors of the absolute and active velocities of the spindle tooth, which are the main criteria for the efficiency of the harvesting device.

4. Conclusion
1. A more efficient orientation of the active surface of the spindle in the working chamber was proposed for the technological cycle of the operation of vertical-spindle picking apparatus due to the change in the kinematic diagram of the drive of the working body roller and the epi-cyclic planetary gear.
2. A method for calculating the magnitudes and directions of the vectors of absolute and active velocities of the spindle teeth tips during its complex plane-parallel movement was developed based on the well-known methods of vector algebra and analytical mechanics.

The analytical expressions (5) - (8) obtained, make it possible to determine the values and directions of the vectors of the absolute and active velocities of the spindle tooth, which are the main criteria for the efficiency of the harvesting device.

5. Acknowledgments
This study was funded from the budget of the Republic of Uzbekistan.

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