KINEMATICS OF A ROTARY TILLER EQUIPPED WITH A TURNING MECHANISM FOR TREE-TRUNK BYPASS
/
ՇՐՋԱԴԱՐՁԱՅԻՆ ՄԵԽԱՆԻԶՄՈՎ ՅՈՂԱՄՇԱԿ ՖՐԵԶԻ ԿԻՆԵՄԱՏԻԿԱՆ ԾԱՌԱԲՆԵՐԻ ՇՐՋԱՆՑՄԱՆ ԺԱՄԱՆԱԿ

Albert GRIGORYAN, Artur ALTUNYAN*), Seryozha PAPYAN, Arman SIMONYAN, Aram MAZMANYAN, Arshaluys TARVERDYAN
Scientific Research Institute for Agricultural Mechanization and Automation, Armenian National Agrarian University, Yerevan, Armenia
Tel: +37455327325; E-mail: arturaltunyan@gmail.com
Corresponding author: Artur Altunyan
DOI: https://doi.org/10.35633/inmateh-79-100

***Keywords:** rotary tiller, near-trunk zone, bypass kinematics, protection zone, untreated area, motion trajectory.*

ABSTRACT
To prevent mechanical damage to tree trunks during cultivation of orchard inter-row and near-trunk areas, agrotechnical requirements prescribe leaving a certain uncultivated soil area around each trunk as a plant protection zone. When the working unit is periodically displaced relative to the tree-row axis by forced motion, an additional untreated soil area remains outside the plant protection zone. Its size is influenced by several factors, including the diameters of the rotary tiller and tree trunk, the lengths of the swing arm carrying the tiller and the sensing probe that actuates the hydraulic system, the tractor travel speed, and the piston-rod velocities of the hydraulic cylinder. Among these factors, the present study considers the principal kinematic factor: the influence of the backward motion of the hydraulic-cylinder piston rod toward the inter-row space and its forward motion toward the tree row, which provide and regulate the transverse displacement of the tiller relative to the tree-row axis, on the size of the untreated soil area. Using a grapho-analytical method for drive mechanism analysis, the equations of motion of the vertical-axis rotary tiller during tree-trunk bypass were derived for the backward motion of the hydraulic-cylinder piston toward the inter-row space and the forward motion toward the tree row. The resulting motion trajectories are cycloids. To characterize them, the concept of a bypass kinematic parameter was introduced: $\Lambda = V_c/V_m$, representing the ratio of the circumferential velocity of the tiller's transverse-radial motion to the tractor forward velocity. The size, shape, and position of the untreated soil area outside the plant protection zone relative to the tree trunk are determined by the value of the bypass kinematic parameter, which, for a given tractor speed, depends on the backward and forward velocities of the hydraulic-cylinder piston rod. It was established that, for given structural and kinematic parameters of the tillage machine, regulating the volume of oil supplied to the hydraulic cylinder makes it possible to minimize the untreated soil area outside the plant protection zone around the tree trunk.

ԱՄՓՈՓԱԳԻՐ
Պտղատու այգիներում միջբնային և մերձբնային տարածությունների մշակման ընթացքում ծառաբների մեխանիկական վնասումը բացառելու նպատակով, ագրոտեխնիկական պայմաններով նախատեսված դրանց շուրջ թողնվում է որոշակի չափով անմշակ հողատարածք (բուսապաշտպան գոտի): Ծառաշարքի առանցքի նկատմամբ աշխատանքային օրգանի պարբերական տեղաշարժը հարկադրական իրականացնելու դեպքում բուսապաշտպան գոտուց դուրս մնում է որոշ չափի չմշակված հողամակերես: Դրա մեծության վրա ազդում են մի շարք գործոններ. ֆրեզի և ծառաբնի տրամագծերը, ֆրեզը տեղափոխող շարժաթևի և հիդրոհամակարգի գործարկման ազդանշանային շոշափոցի երկարությունները, տրակտորի արագությունը, հիդրոգլանի մխոցակոթի շարժման արագությունները: Նշված գործոններից դիտարկվել է հիմնական կինեմատիկական գործոնը՝ ծառաշարքի առանցքի նկատմամբ ֆրեզի լայնական տեղափոխություններն ապահովող և կարգավորող հիդրոգլանի մխոցակոթի հետընթաց (դեպի միջշարք) և առաջընթաց (դեպի ծառաշարք) շարժումների արագությունների ազդեցությունը չմշակված հողամակերեսի մեծության վրա:

Շարժաբեր մեխանիզմի գրաֆո-անալիտիկական վերլուծության եղանակով ստացվել են ծառաբների շրջանցման ընթացքում ուղղաձիգ առանցքով հողամշակ ֆրեզի շարժման հավասարումները հիդրոգլանի մխոցի հետընթաց (դեպի միջշարք) և առաջընթաց (դեպի ծառաշարք) շարժումների դեպքում: Շարժման հետագծերն իրենցից ներկայացնում են ցիկլոիդաներ, որոնց բնութագրման համար ներմուծել ենք շրջանցման կինեմատիկական պարամետր հասկացությունը՝ $\Lambda = V_c/V_m$*, որն իրենից ներկայացնում է ֆրեզի լայնակի-շառավղային շարժման շրջագծային և տրակտորի համընթաց շարժման արագությունների հարաբերությունը: Բուսապաշտպան գոտուց դուրս չմշակված հողամակերեսի չափը, տեսքը և ծառաբնի նկատմամբ դիրքը պայմանավորված են շրջանցման կինեմատիկական պարամետրի արժեքներով, որը՝ տրակտորի տրված արագության դեպքում, կախված է հիդրոգլանի մխոցակոթի հետընթաց և առաջընթաց շարժումների արագություններից: Սահմանվել է, որ հողամշակ մեքենայի կառուցվածքային և կինեմատիկական տրված պարամետրերի դեպքում հիդրոգլան մղվող յուղի ծավալի կարգավորման միջոցով կարելի է ապահովել ծառաբնի շուրջ բուսապաշտպան գոտուց դուրս անմշակ հողամակերեսի նվազագույն մեծություն:*

INTRODUCTION

The technological process of soil cultivation in the near-trunk and inter-trunk zones of orchards plays an important role in increasing fruit yield, its primary objectives being weed control and loosening of the upper soil layers *(Cabrera-Pérez et al., 2022; Manzone et al., 2020; Hanson et al., 2016).*

Despite the existence of various weed-control methods — including thermal, chemical, laser, and other techniques — mechanical weed control remains the most widely applied approach at present. This method is environmentally friendly, sustainable, and effective; it improves soil aeration and increases crop productivity, which explains its widespread application in horticulture *(Morin, 2020; Gagliardi et al., 2023; Gagliardi et al., 2024; Zawada et al., 2023; Parasca et al., 2024; Liu et al., 2023; Pergher et al., 2019)*.

It should be noted that one of the most important agrotechnical requirements for mechanical orchard maintenance is to prevent damage to the tree trunk and root system by the working unit during near-trunk cultivation. In particular, the risk of uprooting the tree must be completely eliminated (*Manayenkon et al., 2017*). To meet this requirement, a prescribed uncultivated area is maintained around the tree trunk, referred to as the *plant protection zone* (Fig. 1).

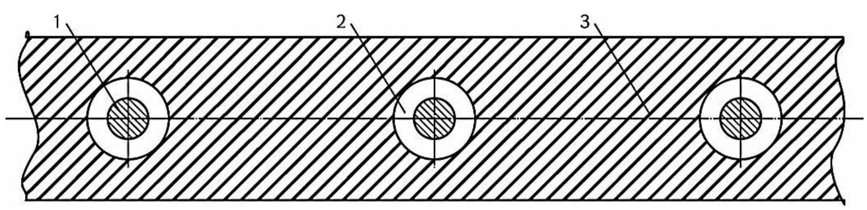


Fig. 1- Schematic representation of the inter-trunk and near-trunk cultivation areas in accordance with agrotechnical requirements

1- tree trunk, 2- plant protection zone, 3- cultivated area.

To implement the above-mentioned process, various types of tillage equipment operating on different principles may be used, including rotary tillers, harrows, sweep blades, and cultivators *(Kostandinov et al., 2016)*. Regardless of the tillage method and the type of technical equipment employed, the working units must be capable of performing periodic transverse displacements relative to the tree-row axis.

Among modern orchard tillage machines, rotary tiller units equipped with a turning section and a working body with a vertical axis of rotation have recently become widely used *(Rzaliev et al., 2025)*. As a rule, these machines cultivate both the inter-trunk and near-trunk zones in a single pass. Operational practice and experimental studies of currently used rotary tiller machines demonstrate that, during the cultivation of inter-trunk and near-trunk zones, an untreated soil area remains between the plant protection zone and the area cultivated by the working unit while bypassing the tree trunk *(Papyan & Harutyunyan, 2024; Inter-Row Machine, EP 3 824 708 A1, 2021; Wang et al., 2023)*.

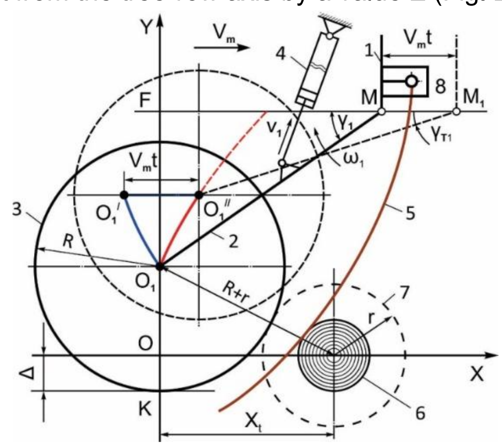
The objective of this study is to determine a trajectory of rotary tiller motion during tree-trunk bypass that prevents the tiller from entering the plant protection zone while simultaneously ensuring a minimum distance between the tiller contour and the plant protection zone boundary. It is evident that such an approach will also ensure the minimum possible area of untreated soil remaining outside the plant protection zone.

MATERIALS AND METHODS

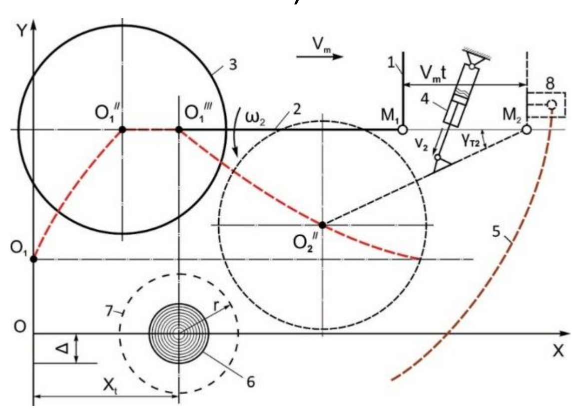
The object of the study was a rotary tiller machine with a vertical axis of rotation, intended for the cultivation of near-trunk and inter-trunk zones in intensive orchards and vineyards, was selected. Its structural and design schemes are presented in Fig. 2 a, b. According to the classification of automatic control systems, the machine may be categorized as an *astatic indirect-action system*, in which the automatic control system maintains the prescribed parameter independently of the magnitude of external disturbances. In the present case, there is no static correspondence between the mutual positions of the working unit and the sensing probe, while the extreme positions of the working unit are determined either by the full stroke of the hydraulic-cylinder piston or by external limiters.

The developed rotary tiller machine consists of a transverse frame (1), a swing arm rotating in the radial direction (2) mounted on the frame by means of a hinge joint, a rotary tiller with a vertical axis of rotation (3) fixed to the swing arm, a hydraulic cylinder (4) actuating the swing arm, the sensing probe (5) mounted on the frame, and an inductive sensor assembly (8).

The operating principle of the rotary tiller machine is as follows. Prior to contact with a tree trunk, the hydraulic cylinder (4) is in its extended (open) position, while the rotary tiller (3) mounted on the swing arm is located within the tree row. In order to ensure overlap between successive passes during field operation, the contour of the tiller is laterally offset from the tree-row axis by a value Δ (Fig. 2a,b).



a)



b)

Fig. 2- Schematics for deriving the equations of motion of the center of the vertical-axis rotary tiller during tree-trunk bypass:

a) backward motion of the hydraulic-cylinder piston; b) forward motion of the hydraulic-cylinder piston

The bypass of the tree trunk by the tiller is conventionally divided into two stages. Stage 1**:** from the moment the signal probe contacts the tree trunk until the end of contact and stage 2: after the probe–trunk contact is terminated, when the tiller, in the free state of the probe, returns again into the tree row. In the first stage, the signal probe (5), mounted on the frame (1), upon encountering the tree trunk (6), rotates and, via inductive sensors, transmits a signal to the electromagnetic hydraulic distributor. Due to the change in the position of the distributor spool, the direction of the oil flow supplied to the hydraulic cylinder is reversed. As a result, the piston of the hydraulic cylinder moves backward, and through the rotation of the swing arm about point M, the tiller is withdrawn from the tree row (Fig. 2a). In the second stage, when the contact between the probe and the tree trunk ceases, the probe returns to its initial position under the action of a spring. A reverse signal is transmitted from the inductive sensor to the hydraulic distributor, and the forward movement of the hydraulic-cylinder piston causes the tiller to re-enter the tree row (Fig. 2b).

During tree-trunk bypass, the turning section of the rotary tiller machine performs a complex motion, consisting of a rotation about point M with radius $L = |O_1M|$ and a translational motion with velocity V_m in the direction of tractor forward motion.

For deriving the equations of motion of the tiller center, the coordinate system is defined such that the X-axis is aligned with the longitudinal direction of the tree row, while the ordinate axis passes through the initial position of the tiller center O_1 .

During the withdrawal phase from the tree row (first stage), over a time interval t_1, the center O_1 of the tiller rotates about point M from its initial position by an angle of $\gamma_1 - \gamma_{\text{т}1}$. As a result, point O_1 moves to position O_1'. However, since during the same time interval t_1 the translational motion of the tractor with velocity V_m also occurs, point O_1' is further displaced by a distance $V_m t_1$ in the direction of motion, reaching position O_1'' . The coordinates of point O_1'' can be written as:

$$\begin{cases} x_1 = V_m t_1 + L(cos\gamma_1 - cos\gamma_{\text{т}1}) \\ y_1 = R - \Delta + L(sin\gamma_1 - sin\gamma_{\text{т}1}), \end{cases} \tag{1}$$

where:

V_m is the tractor forward (translational) velocity, $[m/s]$; t_1-is the time interval during which the tiller center O_1 rotates about point M through the angle $\gamma_1 - \gamma_{\text{т}1}$; $[s]$; γ_1-is the initial installation angle of the tiller relative to the direction of machine motion $(\angle O_1MF)$, $[deg]$; $\gamma_{\text{т}1}$ is the current bypass angle $(\angle O_1''M_1F)$, $\left(\gamma_{\text{т}1} \in [\gamma_1,\ 0]\Big|_{\gamma_{\text{т}1}^{min} = 0}^{\gamma_{\text{т}1}^{max} = \gamma_1}\right)$, $[deg]$; L is the length of the swing arm of the tiller ($|O_1M|$), $[m]$; R is the radius of the tiller, $[m]$; Δ is the overlap width/value between adjacent passes of the tiller, $[m]$.

At the end of the first stage of bypass, the position of the tiller relative to the tree trunk is determined by the length of the sensing probe. In the case of a long probe, the tiller begins entering the inter-tree zone relatively late, thereby increasing the area of untreated soil around the tree trunk. Conversely, if the probe is excessively short, the risk of the tiller entering the plant protection zone and damaging the tree trunks increases.

Assume that the first stage of tree-trunk bypass ends when the centers of the tiller and the tree trunk lie on the same line, i.e., when the contact between the tree trunk and the sensing probe terminates. This means that the extreme point of the probe will also be located on the line connecting the centers of the tree trunk and the tiller. Depending on the kinematic parameters of the technological process, situations may arise in which, after the tiller center O_1 has rotated through a prescribed angle about point M, contact between the probe and the tree trunk is still maintained. In this case, the tiller continues only its translational motion along the segment $|O_1''O_1'''|$| (Fig. 2b). Starting from point O₁"", the second stage of bypass begins through the forward motion of the hydraulic-cylinder piston, and the tiller is displaced back toward the tree row (Fig. 2b).

For the specified section, $|O_1'''O_2''|$, the equations of the tiller motion may be written in the following form:

$$\begin{cases} x_2 = x_1^{max} + V_m\, t_2 + L\left(cos_{\gamma_{\text{т}1}}^{min} - cos\gamma_{\text{т}2}\right) \\ \qquad y_2 = R - \Delta + L(sin\gamma_1 - sin\gamma_{\text{т}2}), \end{cases} \tag{2}$$

where: x_1^{max} is the abscissa of the tiller center at the end of the first stage of bypass, $[m]$; t_2 is the duration of the second stage of bypass, $[s]$; $\gamma_{\text{т}2}$ is the current bypass angle of the tiller during its motion toward the tree row $(\angle O_2''M_2F)$; $\left(\gamma_{\text{т}2} \in [0,\ \gamma_1,]\Big|_{\gamma_{\text{т}2}^{min} = 0}^{\gamma_{\text{т}2}^{max} = \gamma_1}\right)$, $[deg]$;.

The actuation mechanism of the swing arm of the rotary tiller machine may be represented as a slotted-link mechanism with one degree of freedom, in which the rotation of the swing arm of length $L = |MO_1|$ about point M is determined by the variation of the distance $S = |DB|$ (forward and backward motions of the hydraulic-cylinder piston) *(Artobolevkiy, 1988; Smirnov, 2014)* (Fig. 3).

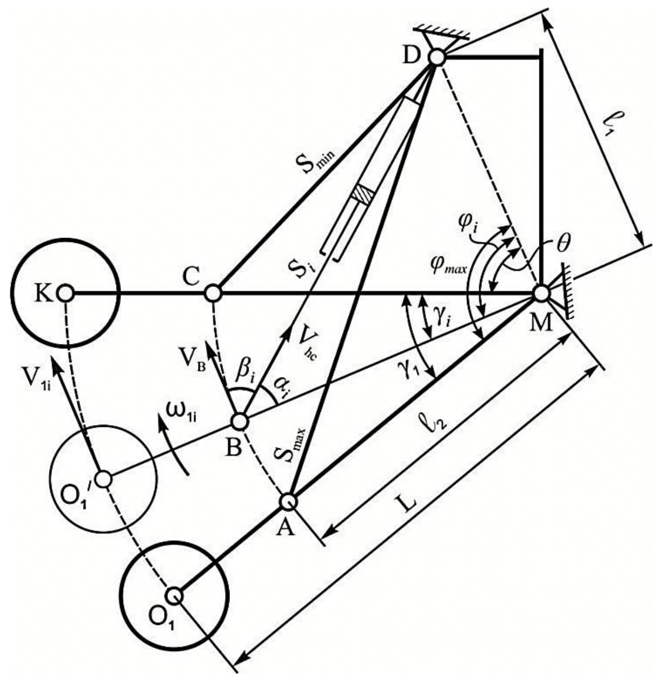


Fig. 3 - Scheme for determining the angular velocity of the swing arm

According to the scheme:

$$\gamma_i = \varphi_i - \theta, \tag{3}$$

where:

φ_i is the current angle formed between the swing arm and the line connecting the fixed kinematic pairs M and D, ($\angle AMD$), $\left(\varphi_i \in [\varphi_{max}, \theta] \middle| \begin{matrix} \varphi_{max} = \gamma_1 + \theta \\ \varphi_{min} = \theta \end{matrix}\right)$, $[deg]$; θ is a structural parameter (constant geometric angle), $[deg]$.

The kinematic relationship between the rotation angle φ_i of the swing arm and the distance S_i may be expressed by the following equation for triangle (ΔDMB):

$$S_i^2 = l_1^2 + l_2^2 - 2l_1 l_2 cos\varphi_i, \tag{4}$$

from which:

$$\varphi_i = arccos\frac{l_1^2 + l_2^2 - S_i^2}{2l_1 l_2}, \tag{5}$$

where:

$S_i = S_{max(min)} \mp \Delta S$ - represents the current extension value of the hydraulic cylinder, $\left(S_i \in [S_{max}, S_{min}] \middle| \begin{matrix} S_{max} = |AD| \\ S_{min} = |CD| \end{matrix}\right)$, $[m]$; $\Delta S = V \cdot t_i$- is the displacement of the hydraulic-cylinder piston per unit time, $[m]$; V is the velocity of the hydraulic-cylinder piston, $[m/s]$; l_1 is the length of the line connecting the fixed kinematic pairs M and D, $[m]$; l_2 is the distance from point M to point A, where the hydraulic cylinder is connected to the swing arm, $[m]$.

The angular velocity analogue of the swing arm is obtained by differentiating Eq. (5) with respect to S. Multiplying the resulting equation by the velocity of the driving link — namely, the hydraulic-cylinder piston velocity $\left(\frac{dS_i}{dt}\right)$, yields the angular velocity of the swing arm:

$$\omega_i = \frac{d\varphi_i}{dS} = \frac{S_i}{l_1 l_2} \cdot \frac{1}{\sqrt{1 - \left(\frac{l_1^2 + l_2^2 - S_i^2}{2 l_1 l_2}\right)^2}} \cdot \frac{dS_i}{dt}. \tag{6}$$

where:

ω_i is the angular velocity, [s^{-1}].

Because the backward velocity V_1 and the forward velocity V_2 of the hydraulic-cylinder piston are different, with $V_1 > V_2$, the angular velocities of the swing arm in the corresponding sections of motion also differ. Accordingly, for a swing arm of length $L = |MO_1|$, the angular velocities of the swing arm during the backward and forward motions of the hydraulic cylinder piston are given, respectively, by:

$$\omega_1 = \frac{S_{max} - \Delta S}{l_1 l_2} \cdot \frac{1}{\sqrt{1 - \left(\frac{l_1^2 + l_2^2 - (S_{max} - \Delta S)^2}{2 l_1 l_2}\right)^2}} \cdot \frac{d(S_{max} - \Delta S)}{dt} \tag{7}$$

$$\omega_2 = \frac{S_{min} + \Delta S}{l_1 l_2} \cdot \frac{1}{\sqrt{1 - \left(\frac{l_1^2 + l_2^2 - (S_{max} + \Delta S)}{2 l_1 l_2}\right)^2}} \cdot \frac{d(S_{min} + \Delta S)}{dt}, \tag{8}$$

where:

$V_1 = \frac{d(S_{max} - \Delta S)}{dt}$ is the backward velocity of the piston, $[m/s]$; $V_2 = \frac{d(S_{min} + \Delta S)}{dt}$ is the forward velocity of the piston, $[m/s]$.

Knowing the angular velocities of the swing arm in the corresponding sections, the respective circumferential velocities of the tiller center may be determined:

$$V_1 = L \cdot \omega_1 \tag{9}$$

$$V_2 = L \cdot \omega_2. \tag{10}$$

The backward and forward velocities of the hydraulic-cylinder piston are also determined by the following equation:

$$v_{1(2)} = \frac{Q \cdot \mu}{F_{1(2)} \cdot 60 \cdot 1000}, \tag{11}$$

where:

$F_{1(2)}$ are the effective piston areas in the rod-side and piston-side chambers of the hydraulic cylinder $[m^2]$; μ is the efficiency coefficient of the hydraulic system, $\mu = 0{,}85 - 0{,}95$.

The effective piston areas on the rod side and piston side of the hydraulic cylinder are, respectively:

$$F_1 = \pi \frac{(D^2 - d^2)}{4}, \quad F_2 = \pi \frac{D^2}{4},$$

where: D is the piston diameter, and d is the piston-rod diameter, $[m]$.

The objective of the study is to determine the optimal relationship between the tractor forward velocity and the circumferential velocity of the swing arm rotating about point M, such that the untreated soil area remaining outside the plant protection zone around the tree trunk is minimized.

RESULTS AND ANALYSIS

The ratio of the circumferential velocity of the tiller center about point M, i.e., the swing-arm mounting point, to the tractor forward velocity is defined as the *bypass kinematic parameter* and expressed as:

$$\Lambda_i = V_i/V_m, \tag{12}$$

where:

$V_i = L\omega_i$ is the circumferential velocity of the tiller center about point M (ω_i is the angular velocity of the swing arm).

During the first stage of tree-trunk bypass, corresponding to the backward motion of the hydraulic-cylinder piston, the angular velocity of the tiller center about point M is $\omega_1 = \frac{|\gamma_1-\gamma_{\text{т}1}|}{t_1}$, whereas during the motion of the tiller toward the tree row, corresponding to the forward motion of the hydraulic-cylinder piston, the angular velocity is $\omega_2 = \frac{|\gamma_{\text{т}1}^{min}-\gamma_{\text{т}2}|}{t_2}$. These angular velocities may also be determined from Eqs (7) and (8). Substituting the relation $\Lambda = V_c/V_m$ into Eqs. (1) and (2), and taking into account that $V_c = \omega \cdot L$, the equations of motion of the tiller center O_1 for the backward and forward motions of the hydraulic-cylinder piston take the following form:

For the backward motion of the piston:

$$\begin{cases} x_2 = x_1^{max} + L\left(\dfrac{|\gamma_{\text{т}1}^{min} - \gamma_{\text{т}2}|}{\Lambda_2} + cos\gamma_{\text{т}1}^{min} - cos\gamma_{\text{т}2}\right) \\ y_2 = R - \Delta + L(sin\gamma_1 - sin\gamma_{\text{т}2}), \end{cases} \tag{13}$$

For the forward motion of the piston:

$$\begin{cases} x_2 = x_1^{max} + L\left(\dfrac{|\gamma_{\text{т}1}^{min} - \gamma_{\text{т}2}|}{\Lambda_2} + cos\gamma_{\text{т}1}^{min} - cos\gamma_{\text{т}2}\right) \\ y_2 = R - \Delta + L(sin\gamma_1 - sin\gamma_{\text{т}2}), \end{cases} \tag{14}$$

where:

$\Lambda_1 = \frac{\omega_1 t_1}{V_m}$ is the bypass kinematic parameter during the first stage of tree-trunk bypass (backward motion of the hydraulic-cylinder piston); $\Lambda_2 = \frac{\omega_2 t_2}{V_m}$ is the bypass kinematic parameter during the second stage of tree-trunk bypass (forward motion of the hydraulic-cylinder piston); x_1^{max} is the abscissa corresponding to the centers of the tiller and the tree trunk at the end of the first stage of bypass.

It is evident that the angles $|\gamma_1 - \gamma_{\text{т}1}|$ and $|\gamma_{\text{т}1}^{min} - \gamma_{\text{т}2}|$ vary over equal intervals, but at different rates, which is due to the difference between the forward and backward velocities of the hydraulic cylinder piston; specifically, the forward velocity is lower than the backward velocity.

From the equation defining the bypass kinematic parameter $\Lambda = V_c/V_m$, it follows that, at a constant tractor forward velocity, its value depends on the circumferential velocity of the rotary tiller's center relative to point *M*. This velocity, in turn, depends on the forward and backward velocities of the hydraulic-cylinder piston.

Since the piston velocities differ during the forward and backward motions of the hydraulic cylinder, the values of the bypass kinematic parameter also differ along the corresponding sections of the tree trunk bypass trajectory; moreover, $\Lambda_s < \Lambda_b$.

In order to prevent the rotary tiller from entering the plant protection zone and damaging the tree trunk and root system, it is necessary that the distance from any point of the tiller center trajectory to the center of the tree trunk be greater than or equal to the sum of the radii of the tiller and the plant protection zone: $R + r$ (Fig. 2).

Thus, for the trajectory section corresponding to the first stage of tree trunk bypass, Eq. (15) can be written as follows:

$$\sqrt{(x_1 - x_t)^2 + y_1^2} \geq R + r, \tag{15}$$

Similarly, for the trajectory section corresponding to the second stage of tree-trunk bypass, Eq. (16) can be written as follows:

$$\sqrt{(x_2 - x_t)^2 + y_2^2} \geq R + r, \tag{16}$$

where:

x_1 and y_1 are the current coordinates of the rotary tiller center during the first stage of bypassing; x_2 and y_2 are the current coordinates of the tiller's center during the second stage of bypassing; $x_t = \sqrt{2R(r+\Delta)}$ is the abscissa of the tree trunk center (Fig. 2a).

The problem is analyzed for a specific set of geometric and kinematic parameters of the rotary tiller.

Consider the kinematics of a rotary tiller with a vertical axis of rotation, equipped with rotary tiller of diameter $D = 0{,}35$ m and a swing arm length $L = 0{,}38$ m during tree trunk bypassing.

It is assumed that the tractor forward travel speed is $V_m = 0{,}75$ m/s, the overlap between adjacent passes of the rotary tiller is $\Delta = 0{,}05$ m, and the angular position of the tiller relative to the direction of the aggregate motion is $(\angle O_1MF)\ \gamma_1 = 35^0$ (Fig. 2a). Therefore, during the first stage of tree-trunk bypass, the angle of rotation of the rotary tiller about point M varies from 35^0 to 0^0, while during the second stage, when returning to the tree row, it varies from $0^0$to 35^0. The maximum displacement of the rotary tiller center along the ordinate axis from the initial position is determined as $|O_1F| = |O_1M| \cdot sin\gamma_1 = 0{,}38 \cdot sin35 \approx 0{,}22$ m (Fig. 2a). The length of the line connecting the fixed kinematic pairs M and D is $l_1 = 0{,}3$ m, while the distance from point M to point A, where the hydraulic cylinder is connected to the swing arm, is $l_2 = 0{,}35$ m (Fig. 3).

The rotation angle of the sensing probe is adjusted so that the inductive sensors generate a signal for the backward motion of the hydraulic cylinder when the distance between the rotary tiller contour and the probe becomes equal to the width of the plant protection zone, i.e., 0.05 m.

To actuate the swing arm, a hydraulic cylinder of type 40-25-160 is selected, with the following principal parameters: piston diameter: $D = 0{,}04$ m, piston rod diameter: $d = 0{,}025$ m, stroke length: $h = 0{,}16$ m, $S_{max} = 0{,}51$ m, $S_{min} = 0{,}35$ m.

Table 1 presents the variation ranges of the swing-arm angular velocities calculated using Eqs. (7) and (8), the circumferential velocities of the rotary tiller center calculated using Eq. (9), and the bypass kinematic parameter calculated using Eq. (11), for the forward and backward motions of the hydraulic cylinder at different volumetric flow rates of oil supplied to the cylinder.

Table 1

Values of the rotary tiller center velocity and bypass kinematic parameter at different hydraulic oil flow rates ($V_m = 0,75$ m/s)

Oil flow rate, Q, L/min	Swing arm angular velocity, s^{-1}		Circumferential velocity of the rotary tiller center, m/s		Bypass kinematic parameter, Λ	
	Backward motion, ω_1 (7)	Forward motion, ω_2 (8)	Backward motion, V_1 (9)	Forward motion, V_2 (10)	Backward motion, Λ_1 (12)	Forward motion, Λ_2 (12)
10	1.025-0.76	0.46-0.63	0.39-0.29	0.17-0.24	0.52-0.38	0.23-0.32
15	1.54-1.14	0.7-0.94	0.58-0.43	0.27-0.36	0.77-0.57	0.36-0.48
20	2.05-1.51	0.93-1.26	0.78-0.57	0.35-0.48	1.04-0.76	0.47-0.64
30	3.08-2.28	1.39-1.89	1.17-0.87	0.53-0.72	1.56-1.16	0.7-0.96
40	4.1-3.03	1.86-2.51	1.56-1.15	0.71-0.95	2.08-1.53	0.94-1.28
50	5.13-3.8	2.3-3.14	1.95-1.44	0.87-1.19	2.6-1.92	1.16-1.61

Based on the system of Eqs. (13) and (14), as well as the data presented in Table 1, the trajectories of motion of the rotary tiller center were plotted for different values of the hydraulic oil flow rate supplied to the hydraulic cylinder and for different values of the bypass kinematic parameter (Fig. 4).

Among the trajectories shown in Fig. 4, the optimal one should be considered the trajectory in which the edge of the rotary tiller moves at the minimum possible distance from the plant protection zone, while simultaneously satisfying conditions (15) and (16). It is evident that, in this case, the uncultivated area outside the plant protection zone will also be minimal.

Fig. 5 shows the variation in the distance between the centers of the rotary tiller and the tree trunk as a function of the bypass kinematic parameter.

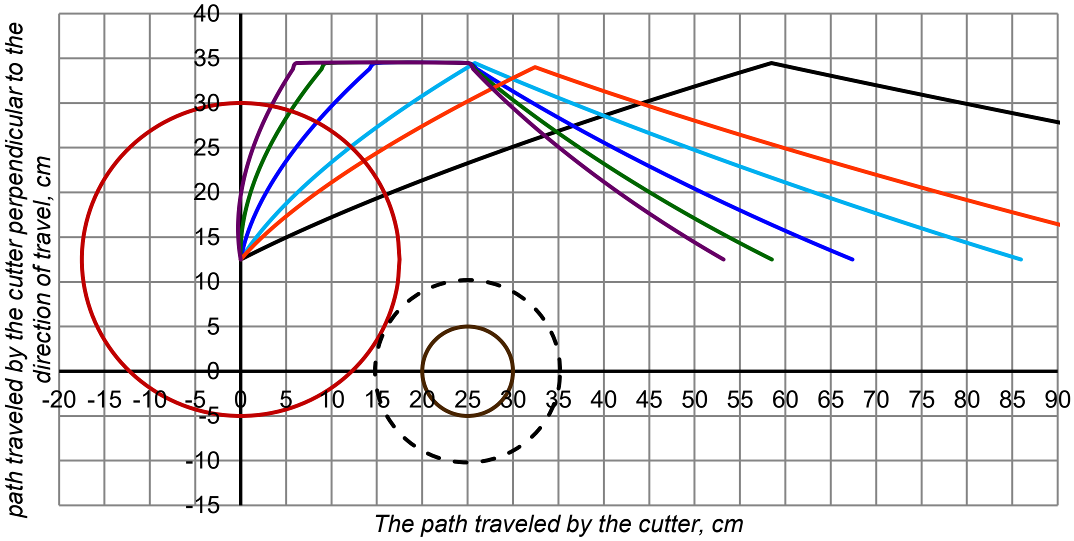


Fig. 4 - Trajectories of the rotary tiller center at different hydraulic oil flow rates, *Q*, and corresponding values of the bypass kinematic parameter, *Λ*

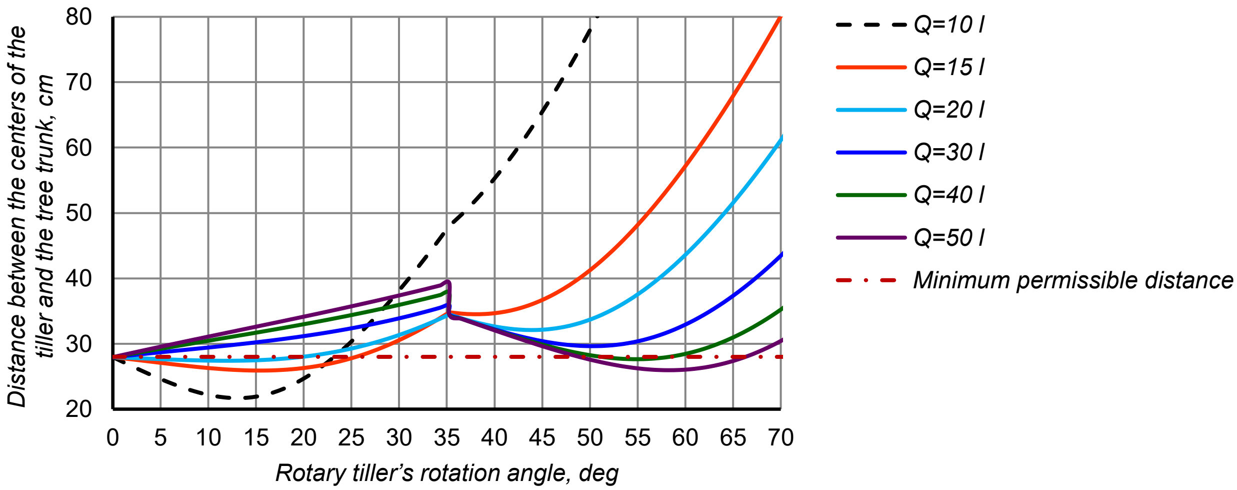
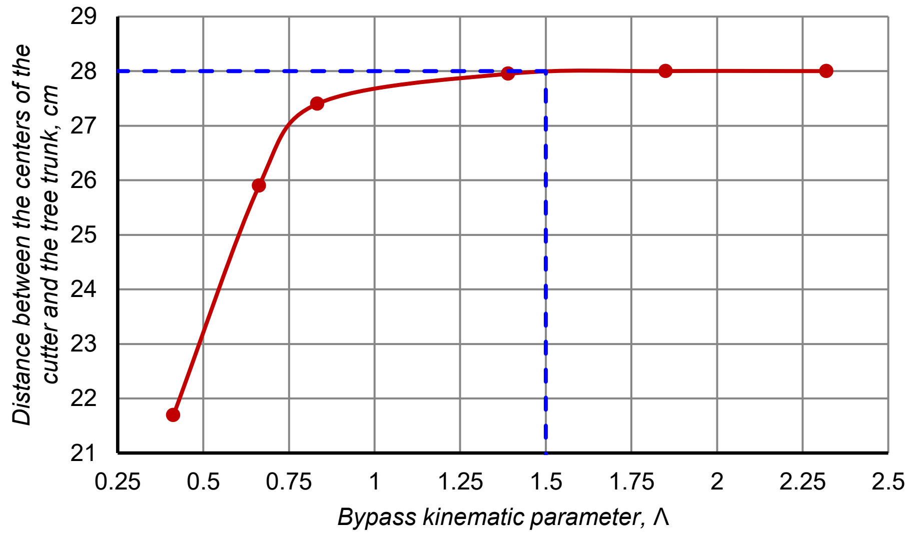


Fig. 5 - Variation in the distance between the centers of the rotary tiller and the tree trunk s a function of the bypass kinematic parameter

The graphs in Figs. 6a and 6b show the variation of the minimum values of the distance between the centers of the rotary tiller and the tree trunk during the first and second stages of tree trunk bypassing, respectively, for different values of the bypass kinematic parameter.



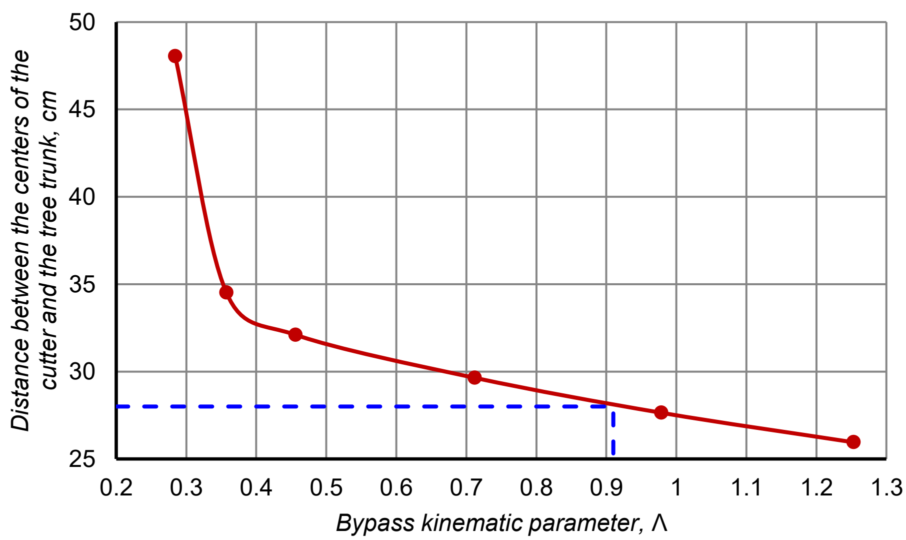


Fig. 6 - Variation in the minimum distance between the centers of the rotary tiller and the tree trunk for different values of the bypass kinematic parameter:
a) first stage, b) second stage

Analysis of the plotted trajectories shows that, during the first stage of tree trunk bypassing, the optimal value of the bypass kinematic parameter is $\Lambda_1 = 1{,}5$, while during the second stage the optimal value is $\Lambda_2 = 0{,}91$.

When selecting the hydraulic system parameters, the value of Λ_2, corresponding to the second stage of tree trunk bypassing, should be taken as the reference, from which the corresponding value of Λ_1 for the first stage is obtained.

In the example considered, the optimal value for the second stage is $\Lambda_2 = 0{,}91$, which corresponds to $\Lambda_1 = 1{,}16$ for the first stage. Therefore, for a machine travel speed of $V_m = 0{,}75$ m/s and the selected hydraulic cylinder of type 40-25-160 the specified values of the bypass kinematic parameters are achieved at a hydraulic oil flow rate of approximately $Q \approx 30$ l/min in the hydraulic cylinder (Table 1).

CONCLUSIONS

The study showed that, for automatically controlled working units operating on an astatic principle during cultivation of near-trunk and inter-trunk areas in vineyards and intensive orchards, the kinematic parameter based solely on the circumferential velocity of the rotary tiller is insufficient to determine the optimal tiller trajectory and minimize the untreated soil area outside the plant protection zone around tree trunks. Therefore, to more adequately characterize the rotary bypass motion, the concept of the *bypass kinematic parameter* was introduced. The analysis showed that this parameter more accurately reflects the kinematic characteristics of the rotary tiller during tree-trunk bypass.

It was established that, for given agrotechnical requirements, operating conditions, and known structural parameters of the machine, regulation of the hydraulic oil flow rate supplied to the hydraulic cylinder makes it possible to vary the forward and backward velocities of the piston and, consequently, the values of the bypass kinematic parameter. In this way, a tree-trunk bypass trajectory can be obtained that minimizes the untreated soil area outside the plant protection zone.

REFERENCES

[1] Artobolevsky I.I., (1988). *Theory of Mechanisms and Machines (Теория механизмов и машин),* Nauka (Наука), Moscow, 640 p.

[2] Cabrera-Pérez, C., Valencia-Gredilla, F., Royo-Esnal, A., Recasens, J. (2022). Organic Mulches as an Alternative to Conventional Under-Vine Weed Management in Mediterranean Irrigated Vineyards. *Plants*, 11, 2785. https://doi.org/10.3390/plants11202785

[3] Gagliardi, L., Fontanelli, M., Luglio, S., Frasconi, C., Peruzzi, A., Raffaelli, M. (2023). Evaluation of Sustainable Strategies for Mechanical Under-Row Weed Control in the Vineyard. *Agronomy* 13, 3005. https://doi.org/10.3390/agronomy13123005

[4] Gagliardi, L., Luglio, S., Peruzzi, A., Fontanelli, M., Frasconi, C., Raffaelli, M. (2024). Comparative Analysis of the Performance of a Chain Mower and Tools That Perform Under-Row Weed Control with Tillage in the Vineyard. *Agronomy* 2024, 14. https://doi.org/10.3390/agronomy14010206

[5] Hanson, B.D., Roncoroni, J., Hembree, K.J., Molinar, R., Elmore, C.L. (2016). Weed Control in Orchards and Vineyards. *Encyclopedia of Applied Plant Sciences*, New York, NY, USA, pp. 479–484. https://doi.org/10.1016/B978-0-12-394807-6.00032-0

[6] Inter-Row Machine, EP 3 824 708 A1, 26.05.2021. RINIERI, Nicola 47121 Forli', Frazione Collina (IT)

[7] Kostadinov G. D., G. Parkhomenko G. G., Tverdohlebov S. A., Ponomarev A. V., (2016). Systematization of Devices for Soil Treatment in The Orchard, (Систематизация устройств для обработки почвы в садах), *Bulgarian Journal of Crop Science (Растениевъдни науки), 2016, 53(1-3), 88-97*. https://agriacad.eu/ojs/index.php/bjcs/article/view/3193/2819

[8] Liu, C., Yang, K., Chen, Y., Gong, H., Feng, X., Tang, Z. (2023) Benefits of Mechanical Weeding for Weed Control, Rice Growth Characteristics and Yield in Paddy Fields. *Field Crops Res.*, 293, 108852. https://doi.org/10.1016/j.fcr.2023.108852

[9] Manaenkov K. A., Koldin M. S., Arkova Z. A., (2017) Improvement of tillage in row strips intensive gardens (Совершенствование обработки почвы в приствольных полосах интенсивных садов), *Food and processing industry technologies (Технологии пищевой и перерабатывающей промышленности)*-N3.28-34 pp. https://cyberleninka.ru/article/n/sovershenstvovanie-obrabotki-pochvy-v-pristvolnyh-polosah-intensivnyh-sadov/viewer

[10] Manzone M., Demeneghi M., Marucco P., (2020) Technical solutions for under-row weed control in vineyards: Efficacy, costs and environmental aspects analysis. *Journal of agricultural engineering* 51(1): pp. 36-42. https://doi.org/10.4081/jae.2020.991

[11] Morin, L. (2020) Progress in Biological Control of Weeds with Plant Pathogens. *Annu. Rev. Phytopathol.* 58, 201–213. https://doi.org/10.1146/annurev-phyto-010820-012823

[12] Papyan S., Harutyunyan G., (2024) Research and Development of a Mechanism for Fixing the Hydraulic Cylinder for Controlling the External Working Element of a Garden Milling Machine and Determining its Optimal Parameters, Scientific journal of ANAU *“AgriScience and Technology*” 4(88), 279-283. https://doi.org/10.52276/25792822-2024.4-279

[13] Parasca, S., Spaeth, M., Rusu, T., Bogdan, I., (2024) Mechanical Weed Control: Sensor-based Inter-row Hoeing in Sugar Beet (Beta vulgaris L.) in the Transylvanian Depression. *Agronomy*, 14, 176. https://doi.org/10.3390/agronomy14010176

[14] Pergher, G., Gubiani, R., Mainardis, M., (2019) Mainardis Field Testing of a Biomass-Fueled Flamer for In-Row Weed Control in the Vineyard. *Agriculture*, 9, 210. https://doi.org/10.3390/agriculture9100210

[15] Rzaliev A.S., Serikbayev A.U., Zhunusbaev A.B., (2025) Theoretical study on the substantiation of the parameters of a vertical-axis rotary tiller for cultivation of the near-trunk zone of fruit trees (Теоретические исследования по обоснованию параметров фрезы с вертикально-роторным валом вращения для обработки приствольной зоны плодовых деревьев), *Journal Izdenister natigeler*, №3 (107), 605-616 pp. https://doi.org/10.37884/3-2025/60

[16] Smirnov D. A., (2014) Kinematic analysis of a one-degree-of-freedom slotted-link mechanism with fixed revolute kinematic pairs (Киематический анализ кулисного механизма с одной степенью свободы с неподвижными вращательными кинематическими парами), *Fundamental research* (*Фундаментальные исследования*), № 8(5). Nizhny Novgorod, 1069-1074 pp. https://fundamental-research.ru/ru/article/view?id=34719

[17] Wang, Y., Kang, J., Peng, Q., Chen, Y., Fang, H., Niu, M., Wang, S. (2023) Design and Experiment of Obstacle Avoidance Weeding Machine for Fruit Trees. *J. Jilin Univ. Eng. Technol. Ed.*, 53, 2410–2420. https://doi.org/10.13229/j.cnki.jdxbgxb.20211121

[18] Zawada, M., Legutko, S., Goscianska-Lowinska J., Szymczyk, S., Nijak, M., Wojciechowski, J. (2023). Mechanical Weed Control Systems: Methods and Effectiveness. *Sustainability*, 15, 15206. https://doi.org/10.3390/su152115206