FORCE ANALYSIS OF A ROTARY TILLER WITH A VERTICAL ROTATION AXIS

ՈՒՂՂԱՁԻԳ ՊՏՏՄԱՆ ԱՌԱՆՑՔՈՎ ՀՈՂԱՄՇԱԿ ՖՐԵԶԻ ՈՒԺԱՅԻՆ ՎԵՐԼՈՒԾՈՒԹՅՈՒՆ

Arshaluys TARVERDYAN, Artur ALTUNYAN*), Albert GRIGORYAN
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

ABSTRACT

The article considers the issue of updating the machines designed for the cultivation of near-trunk and near-vine spaces. From the prospect of complete satisfaction of the agrotechnological and technical requirements of the mentioned machines, the rotary tillers with vertical rotation axis are preferable, which, however, need to get structurally developed. On this account, it is recommended to select planetary mechanism with external gear as a driving mechanism, which enables to ensure high rotation numbers of the tiller in case of relatively low rotation numbers of the rotor. The kinematic analysis of the driving mechanisms has been conducted, analytical expressions have been derived, which enable to determine the trajectory, speeds and accelerations of the tiller's blades depending on the frequency of rotor's rotation and the gear ratio of the mechanism. Based on the analytical expressions resulted from the kinematic research of the mechanism force analysis of the rotary tiller has been conducted. The estimated expressions of the general resistance moment value applied to the shaft of the moving rotor of the machine, as well as the value of resistance moments generated on the planet gear (satellite) rotors have been derived, which enable to identify the optimal kinematic and geometric parameters for the mechanism and rotary through the theoretical calculation method ensuring the optimal values of traction resistance force and other force factors in conditions of the stated working rules.

ԱՄՓՈՓԱԳԻՐ

Հոդվածը նվիրված է պտղատու և խաղողի այգիների մերձբնային և մերձվազային տարածքների մշակման մեքենաների կատարելագործման հիմնախնդրին: Այդ մեքենաներին ներկայացվող ագրոտեխնոլոգիական և տեխնիկական պահանջների լիակատար բավարարման տեսանկյունից, գերադասելի են ուղղաձիգ պտտման առանցքով հողամշակ ֆրեզները, որոնք, սակայն կառուցվածքային կատարելագործման կարիք ունեն: Այդ նկատառումներից ելնելով առաջարկվել է հողամշակ ֆրեզի համար, որպես հաղորդակ ընտրել արտաքին կառչմամբ պլանետարային մեխանիզմ, որը հնարավորություն է տալիս ռոտորի համեմատաբար փոքր պտույտների դեպքում ապահովել ֆրեզի մեծ պտույտներ: Կատարվել է հաղորդակային մեխանիզմի կինեմատիկական վերլուծություն, ստացվել են անալիտիկ արտահայտություններ, որոնք հնարավորություն են տալիս որոշելու ֆրեզի դանակների հետագիծը, արագությունները և արագացումները կախված ռոտորի պտտման հաճախությունից և մեխանիզմի փոխանցման թվից: Մեխանիզմի կինեմատիկական հետազոտության արդյունքում ստացված անալիտիկ արտահայտությունների հիման վրա կատարվել է հողամշակ ֆրեզի ուժային վերլուծություն: Ստացվել են ինչպես սատելիտ-ֆրեզների վրա առաջացող դիմադրության մոմենտների, այնպես էլ մեքենայի տարիչ-ռոտորի լիսեռին կիրառվող ընդհանուր դիմադրության մոմենտի մեծության հաշվարկային արտահայտություններ, որոնք հնարավորություն են տալիս տեսական հաշվարկային եղանակով սահմանելու մեխանիզմի, ինչպես նաև ֆրեզի կինեմատիկական ու երկրաչափական օպտիմալ պարամետրերը՝ ապահովելով քարշային դիմադրության ուժի և այլ ուժային գործոնների օպտիմալ արժեքներ՝ սահմանված բանելակարգի պայմաններում:

INTRODUCTION

Cultivation of inter-row and inter-trunk spaces is very important from the prospect of mechanization of agro-technical activities taken for the orchards and vineyards treatment. By its specific volume it takes up 40% of the implemented overall activities.

The mentioned works are of priority activities among the agro-technical requirements, since due to their accurate and timely implementation soil moisture, thermal and air regimes are ensured, its physico-mechanical and chemical properties get improved, biological processes in plants and soil get activated, efficient weed and pest control is carried out and last, but not the least, the orchard yield capacity and quality is increased *(Monaenkov K.A., 2017; Zhilitsky Ya. Z. Gerasimov N.I., 1973)*.

From technical standpoint, cultivation of inter-trunk and near-trunk spaces is related to a number of difficulties *(Gerasimov N.I., 1979; Dmitriy Khort et al., 2020)*, particularly the centrifugal installation of the tree inter-trunk and near-trunk cultivation device against the tractor's longitudinal axis, the periodic changes of longitudinal and latitudinal components in soil resistance during the cultivation process, especially under soil and relief conditions of orchards and vineyards of the Republic of Armenia, entail to instability of tractor's operation, which in its turn causes breakdown in technological procedures. And this ends up with either uncultivated area or tree trunk damage.

The above stated circumstances indicate that when developing such kind of machines all agro-technical, technological and technical factors and requirements should be taken into account.

The operational practice of such machines has enabled to develop the main agrotechnical requirements and it has been figured out that the machines with active working parts are more effective *(Gerasimov N.I., 1979; Parkhomenko G.G., 2012; Matin M.A. et al., 2014, 2015)*. They are mostly rotary tillers with horizontal or vertical rotation axes and with mechanical or hydraulic gears *(Panov I.M., 1963; Tarverdyan A.P. et al., 2017; Dmitriy Khort et al., 2020)*.

Structural peculiarities of the rotary tillers with vertical rotation axis enable to maximally adjust the machine to the cultivation conditions of the trees inter-trunk spaces ensuring the observance of agro-technical requirements. The remarkable advantages of the machines belonging to this category are reported in multiple research works, nevertheless, the operational practice of the tillers with vertical axis and the analysis of the literature data show that they are not void of shortcomings either *(Panov I.M., 1963; Byshov N.V., 2017; Chudak S.V., 1975; Mosatovskiy V.B., 1980)*. This made us consider the rotary tiller with vertical rotation axis as the research object.

Considering that rotary tillers with vertical axis have more perspective in view of updating and technical-technological improvements, long-term efforts have been exerted to study the operational indices of the existing tillers, to analyze the literature sources and data and to disclose their shortcomings *(Tarverdyan A.P. et al., 2017; Matin M.A. et al., 2014; Yiwen Tarverdyan A.P. et al, 2022)*.

Taking into account the circumstance that the operational reliability of the existing machines is rather low, particularly in the soil types containing huge amount of rocks and ballast, such as most horticultural areas of the Republic of Armenia, an attempt has been made to develop a rotary tiller with vertical rotation axis, possibly free from the above mentioned flaws, for the cultivation of inter-tree, inter-trunk and inter-vine spaces in orchards and vineyards *(Tarverdyan A.P. et al, 2020; Tarverdyan A.P. et al, 2022; Tarverdyan A.P. et al, 2024)*.

The efforts, aimed at the solution of the current problem through the optimization of the blade form and profile in the existing rotary tillers with vertical axis, haven't provided the expected outcomes *(Byshov N.V., 2017; Mosatovskiy V.B., 1980; Yiwen Yuan et al., 2023)*. Thus, the only way of problem solving is to make fundamental changes in the tiller's structure.

The aim of the current work is to identify the kinematic and force descriptors of the active working parts driven through the structurally new planetary mechanism via analysis and theoretical research, to determine the effect of the main factors on such technically vital parameters, as the values of the main moment and main vector of resistance forces of the tiller's moving shaft in order to set up the optimal working and structural parameters.

MATERIALS AND METHODS

Based on the results of previous research *(Matin M.A. et al., 2015; Tarverdyan A.P. et al., 2019; Tarverdyan A.P. et al., 2017; Iman Ahmadi, 2017)* it was initiated to develop a gear for a tiller with vertical axis, which would be able to ensure high rotation numbers (200 -250 rpm) of the working tiller in case of rotor's relatively lower rotation numbers (e.g. 50-80 rpm); as a matter of fact, it is a planetary mechanism with external gear and with a fixed sun gear (Figure 1).
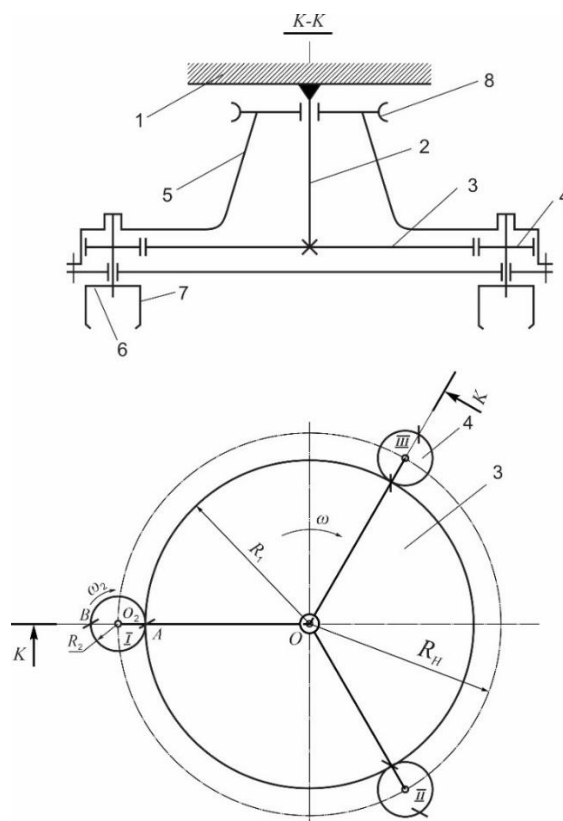
Fig. 1 - The basic diagram of the rotary tiller with planetary gear designed for the cultivation of inter-trunk and near-trunk spaces in orchards and vineyards

The rotary tiller with planetary gear and vertical rotation axis designed for the cultivation of inter-trunk and near-vine spaces of orchards and vineyards consists of a fixed sun gear (3), mounted to the support (2) attached to the rotor's frame (5), planet gears (4) rolled thereto, the rotation axis of which is installed in the grooves of rotor housing/casing. The tillers discs (6) are rigidly attached to the free ends of the rotation axes of the planet gears (4), on which the tiller's blades (7) are mounted in a symmetrical pattern.

The rotor (5) of the tilling machine receives the rotational movement from the hydraulic motor or through the chain transmission (8). The rotational motion is transmitted from the power take-off shaft of the tractor or hydraulic motor to the rotor of the rotary tiller, which in this case acts as a driver of planetary mechanism; due to the rotor's rotation the planet gears (4) roll over the ring belt of the fixed sun gear (3) with the multiplied angular velocity of the gear ratio of the mechanism. In terms of qualitative evaluation, the selected mechanism enables to solve the raised problem: to get the large angular, and hence, linear speeds/velocities of the rotary tillers (working parts) with small diameter in case of relatively small rotation numbers of the rotor.

In the discussed work, per the goals and objectives set up, the trajectories of the tillers' blades, the relation between the geometric and kinematic parameters is of primary interest. To reveal the above discussed questions, let's consider the trajectory of any blade in the tiller, mounted to the axis of planetary mechanism planet gear (4) with the account of forward movement of the machine. The calculation scheme of the problem is presented in Figure 2.

It is known that due to the driver's rotation the fixed point of the planet gear circumference of the planetary mechanism with external gear draws a smooth curve – epicycloid. Moreover, depending on the relationship of the fixed sun gear (R_1) and planet gear (satellite) (R_2) radii, the epicycloidal shape and number of arcs are different *(Ossetsky V.M., 1977).*

In the discussed problem the epicycloid shape is important when the fixed sun gear (3) makes a forward movement with a constant speed, since in that case epicycloid is manifested as trajectories of the tiller's (6) blades (7), which enable the determination of optimal geometric and kinematic parameters of the developed rotary tiller.
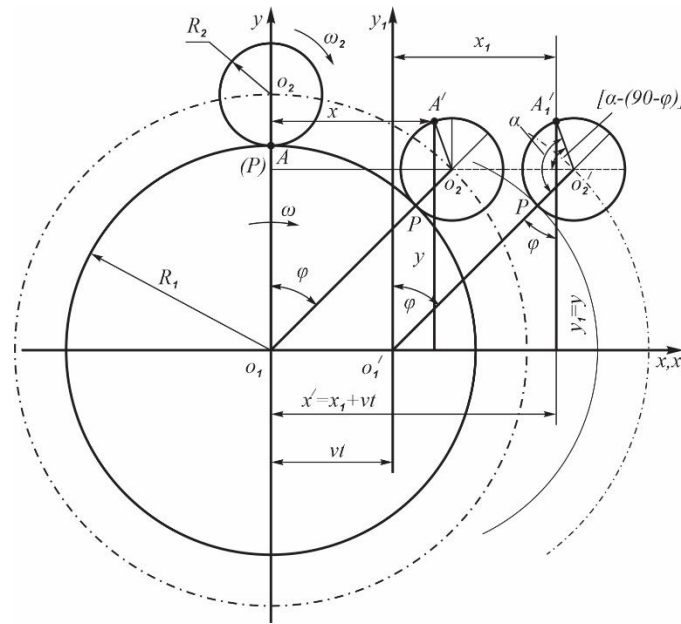
Fig. 2 - The scheme of deriving equations of blade motion in the rotary tiller with planetary driving mechanism

Let's align the initial coordinates point with the center of sun gear (O_1) and assume that in the initial position the center of planet gear (O_2) is on the y axis. O_1O_2 driver gets a rotational motion with ω angular velocity, whereas the planet gear will get a rotational motion with $ω_2$ angular velocity.

In case of the fixed sun gear, at the t moment, when the driver (O_1O_2) rotated with φ angle, the friction A point of sun gear and planet gear would accept $A^{\prime}$ position, but according to the stated term, the sun gear conducts a forward movement with V constant velocity, hence, A point will take on $A^{\prime\prime}$ position, since it also shifts through the direction of X axis with V_t value.

The equation of movement of any fixed A point of the planet gear or tiller's blade with parametric description will look as follows (according to the scheme of figure 2):

$$\begin{cases} x = V \cdot t + (R_1 + R_2)sin\varphi - R_2 sin\left(\frac{R_1+R_2}{R_2} \cdot \varphi\right) \\ y = (R_1 + R_2)cos\varphi - R_2 cos\left(\frac{R_1+R_2}{R_2} \cdot \varphi\right) \end{cases} \tag{1}$$

where R_1 is the radius of relatively fixed sun gear, [m], R_2 is the planet gear radius, [m], $(R_1 + R_2 = R_H)$ is the radius of planetary mechanism driver.

The angular $ω_2$ speed of the planet gear and the rotation $α$ angle of epicycloid-drawing point are determined through the following expressions:

$α = (i + 1)\varphi; \quad ω_2 = (i + 1)ω$, where $i = \frac{R_1}{R_2}$ – is the transmission ratio.

RESULTS AND ANALYSIS

To determine the speed and acceleration of the fixed point (A) of the tiller's blade, the Equation (1) is relevant to introduce in the following form:

$$\begin{cases} X = V \cdot t + R_H\ sinωt - R_2 sin(i + 1)ωt \\ Y = R_H cosωt - R_2 cos(i + 1)ωt \end{cases} \tag{2}$$

When differentiating Eq. (2) per time, the speed components of any fixed (A) point for the tiller's blade will be obtained:

$$\begin{cases} V_x = V + ω[R_H cosωt - R_2(i + 1)cos(i + 1)ωt] \\ V_y = -ω[R_H sinωt - R_2(i + 1)sin(i + 1)ωt] \dots \end{cases} \tag{3}$$

The module of the total velocity for the considered point will be:

$$V_A = \sqrt{V_x^2 + V_y^2} \tag{4}$$

Making respective modifications and considering that $R_H=(i+1)R_2$, for the module of total velocity the following calculated expression will be obtained:

$$V_A = \sqrt{V^2 + 2ω^2 V R_2^2(i + 1)^2[1 - cos(2 + i)ωt] + 2VωR_2(i + 1)[cosωt - cos(i + 1)ωt)} \tag{5}$$

To determine the acceleration components of the considered A point of the blade, let's differentiate the expression Eq. (3) per (t) time and will be obtained:

$$\begin{cases} \alpha_x = -\omega^2[(R_1 + R_2)sin\omega t - R_2(i + 1)^2 sin(i + 1)\omega t\,] \\ \alpha_y = -\omega^2[(R_1 + R_2)cos\omega t - R_2(i + 1)^2 cos(i + 1)\omega t\,] \end{cases} \qquad (6)$$

The acceleration module will be determined in the following way:

$$\alpha_A = \sqrt{{\alpha_x}^2 + {\alpha_y}^2}$$

The article considers the option of planetary mechanism, in which the number of planet gears is three, installed at 120^0 (Fig. 1, Fig. 3). The trajectories of the fixed points A_0 and B_0) of random blade of the tiller, which are implementing rotational and forward movements throughout one rotation of the tiller are introduced in Fig. 3.
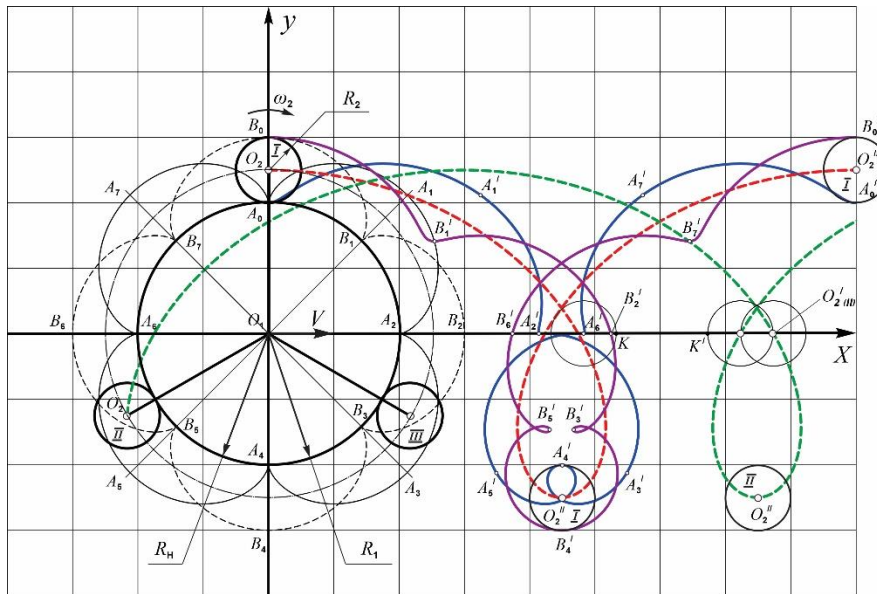

Fig. 3 - The scheme of determining the trajectory of the blades in the rotary tiller with planetary mechanism

The trajectories of the blades attached in the A_0 and B_0 points of the conventional (I) first planet gear without the forward movement of the tiller (V=0) are $A_0\, A_1\, A_2\, A_3\, A_4\, A_5\, A_6\, A_7\, A_0$ и $B_0\, B_1\, B_2\, B_3\, B_4\, B_5\, B_6\, B_7\, B_0$ epicycloids (Figure 3), whereas in case of forward movement of the machine, they are the curves $A_0\, A_1'\, A_2'\, A_3'$ $A_4'\, A_5'\, A_6'\, A_7'\, A_0'$ и $B_0\, B_1'\, B_2'\, B_3'\, B_4'\, B_5'\, B_6'\, B_7'\, B_0'$, respectively (extended epicycloids), which are wrapped around the shortened cycloid (trochoid) $O_2'\, O_2''\, O_2'''\, O_2''''$ drawn by the O_2 center of the planet gear.

In case of one complete rotation of the tiller's rotor casing, which acts as the driver of the planetary mechanism, the length of the soil layer (a segment is shown with a dashed line) cultivated with a tiller attached to a planet gear, is equivalent to the trajectory trochoid length of O_2-center, while the width is equal to the diameter of the tiller attached to the planet gear.

Figure 3 also depicts the trajectory (trochoid) of O_2 center of the second (II) planet gear staying back from the first (I) planet gear by $\frac{\pi}{3}$ phase, which intersects X axis at $0_{2(II)}'$ point. It is obvious that no uncultivated land area will be left if K and K' points coincide ($KK' = 0$). By this term the important kinematic parameter of the machines with rotary working parts is determined, i.e. the relation of forward movement speed (V) and rotor's rotation frequency (ω). In case of 3 planet gear - tillers this relation looks as follows:

$$V \le 6R_2 \cdot \omega$$

In the problems of kinematic and dynamic analysis of the tiller's blade, as mentioned above, the changing parameters of the velocities and accelerations of its fixed point are of great significance, therefore, it is worthwhile introducing the changing graphs of velocity and acceleration components according to the Eq.(3) and Eq.(6) expressions.

It is relevant to present the graphs through some specific example.

Assuming R_1=0.2m, R_2=0.05m, (R_H=0,25m, i=4): ω=5s⁻¹, V=1m/s, ω_2=25s⁻¹.

The changing graphs for velocity components (V_x and V_y) of fixed point of blade attached to the planet gear disc of the rotary tiller and the module of total velocity (V_A) depending on the rotation angle (φ) of the

rotor-driver, as well as the changing graphs of the given point's regular acceleration (a_A) and its components ($a_{A(x)}^n$ and $a_{A(y)}^n$) are introduced in Figure 4.
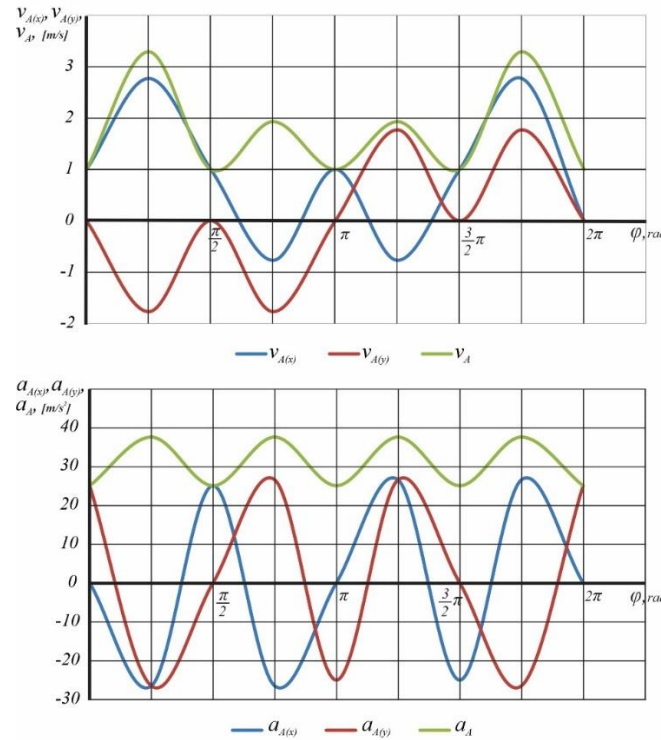

Fig. 4 - The changing graphs of velocities (V_A) and accelerations (a_A) of the fixed point of the tiller blade

From the prospect of the discussed problem, the changing parameters of total velocity module (V_A) and total accelerations module (a_A) are of primary interest. The changing parameters of (V_A) and (a_A) during one rotation period of the driver are related to the rotational or circumferential frequencies of both the driver (ω) and planet gear (ω_2), moreover, the view of their graphs is related to the transmission numbers (i) of the planetary mechanism.

In the phases of $\frac{\pi}{4}$ and $\frac{7}{4}\pi$ V_A gets the maximum value (2.77 m/s), while in those of $\frac{\pi}{2}$, π and $\frac{3}{2}\pi -$ the minimum value (1.0 m/s), which is the speed of forward movement of the aggregate (tractor).

In the intervals of $0 \leq \varphi \leq \frac{\pi}{2}$ and $\frac{3}{2}\pi \leq \varphi \leq 2\pi$ the changing amplitude of (V_A) is 1.145 m/s, while in the interval of $\frac{\pi}{2} \leq \varphi \frac{3}{2}\pi$ it makes 0.465 m/s.

Regarding the changing graph of total acceleration module (a_A) it is presented as a sinusoid of the general view (Figure.4), which is shifted in the direction of y axis to the extent of ω^2, and the amplitude of which in the discussed example is 6.25 m/s², while the period is $T = \frac{2\pi}{\omega_2}$.

To ensure possibly maximum uniform rotation moment for the rotor's shaft of the rotary tiller, it is necessary to select such number and location of planet gears of the planetary mechanism, in case of which the amplitude in the general graph of total accelerations could be minimum. It is apparent that this can be achieved by adding the number of planet gears and their symmetric installation considering the term of planet gears vicinity and their optimal number.

To determine the energy indicators of the recommended machine, it is necessary to disclose the nature of force factors (resistance forces) in the interaction of tiller's blades and the cultivated soil mass throughout the working process of the machine, their values and the patterns of directional changes. At this stage of problem discussion, let's assume the value of the resistance force for a tiller's blade as known and approximately constant (they are factually changing values), set up the changing parameters for the direction of that force effect, since the main, momentum force factors generated during the tiller's operation are mostly related to the directions of resistance force effect lines.

In this regard the analysis of the movement trajectory of the tiller's blade is pushed forth, since during the machine operation the interaction force (resistance) of the blade and cultivated medium (soil) is directed to the tangent in the given point of the trajectory and has a direction opposite to the total velocity vector *(Bat*

M.I. et al., 1975). It is obvious that the problem solution leads to the $y'_{(x)}$ function or to the identification of the equation of the tangent for the trajectory (curve) of the blade's fixed point. As in the discussed case the movement equation is presented in the parametric form Eq. (2), hence, it is known that $\frac{dy}{dx} = \frac{y'(t)}{x'(t)}$, where $y'(t)$ and $x'(t)$ are determined per (2) expressions:

$$y'(x) = -\frac{\omega[(R_1 + R_2) \cdot sin\omega t - R_2(i+1) \cdot sin(i+1)\omega t]}{V\omega[(R_1 + R_2) \cdot cos\omega t - R_2(i+1) \cdot cos(i+1)\omega t]} \qquad (7)$$

Anyhow, it is worthwhile mentioning that the trajectories of fixed points (A and B) of the tiller's arbitrary blades in the rotary machine during one rotation of the driver have complex form in case of $i = 4$ and the derived expression Eq. (7) is not applicable along the whole length of the trajectory. In case of one rotation of the driver, A and B points of the conventional planet gear (I) have quadruple reversal points $A_2'A_4'A_6'A_0'$ and $B_1'B_3'B_5'B_7'$ (Figure 3) of epicycloid or extended epicycloid ($A_0 A_1' A_2' A_3' A_4' A_5' A_6' A_7' A_0'$ and $B_0 B_1' B_2' B_3' B_4' B_5' B_6' B_7' B_0$), which are special points of parametric function, in which the graph of the function breaks and in those points Eq. (2) the expression is not applicable. In the vicinity of those points the behavior of the function is researched with the help of Taylor's formula.

In order to present the changing picture of the blade's resistance forces during the operation of the rotary machine, let's consider the trajectories of A and B fixed points of the two symmetric blades in I planet gear in the rotation domain of $0 \leq \varphi \leq \frac{\pi}{2}$ for the planetary mechanism driver - $O_1O_2 = R_1 + R_2$ (Figure 5).
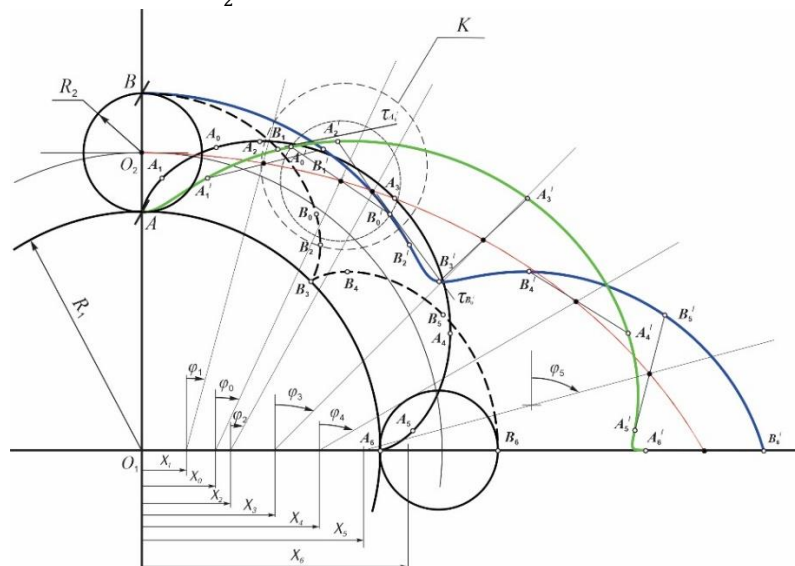

Fig. 5 - **Synchronous scheme of the trajectory of fixed symmetric points in A and B blades attached to the planet gear of the rotary tiller in** $0 \leq \varphi \leq \frac{\pi}{2}$ **domain**

The trajectory of A blade is $A A_0' A_1' A_2' A_3' \dots A_6'$, and for the B blade - $B B_0' B_2' \dots B_6'$; besides, in the mentioned domain B blade has a reversal point - B_3', the consecutive positions of the blades' attachment diameter (AB) are $A_1' B_1'$, $A_2' B_2'$, $A_3' B_3'$, $A_4' B_4'$, $A_5' B_5'$ and $A_6'B_6'$, which correspond to the driver's rotation angles φ_1; $\varphi_2; \dots \frac{\pi}{2}$. The problem of determining the direction of interaction forces in the blades and cultivated soil is relevant to consider in the domain, where the trajectories of the A and B blades (points) are simultaneously missing the reversal points, e.g. when the driver rotated at φ_0 angle which is conformed to $A_0' B_0'$ position of the blades attachment diameter.

The tangent drawn to the trajectory of A blade at A_0' point is $\tau_{A_0'}$, that of the trajectory of B blade at B_0' point is $\tau_{B_0'}$, and to determine their direction it can be used the Eq. (7) expression, but it should be taken into account that this expression has been derived from I planet gear for A blade (point). To use this expression for B point its phasal deviation from A point should be considered.

To present the calculated scheme more clearly, the mentioned sector of Figure 5 (K circle) is depicted in Figure 6 with augmented form, besides, the speeds of A_0 and B_0 points $\left(V_{A_0} \text{ and } V_{B_0}\right)$ are introduced without the forward movement of the unit ($V = 0$, Figure 6a) and with the presence of forward movement $\left(V_{A_0'} \text{ and } V_{B_0'}\right)$

(Figure 6b). In the former case, the total velocity of A point $\bar{V}_{A_0} = \bar{V}_{A_0 O_2} + \bar{V}_{O_2}$ is directed towards the length of τ_{A_0} tangent at the A_0 point of the $A_1 A_2$ fragment in the epicycloid, the total velocity of B point $\bar{V}_{B_0} = \bar{V}_{B_0 O_2} + \bar{V}_{O_2}$ is directed towards the length of τ_{B_0} tangent at the B_0 point of $B_1 B_2'$ fragment in the extended epicycloid. In the second case (Figure 6b) A_0' total velocity $\bar{V}_{A_0'} = \bar{V}_{A_0} + \bar{V}$ is directed towards the length of tangent $\tau_{A_0'}$ at the A_0' point of $A_1' A_2'$ fragment in the extended epicycloid, the total velocity of B_0' point $\bar{V}_{B_0'} = \bar{V}_{B_0} + \bar{V}$ - towards the length of tangent $\tau_{B_0'}$ at the B_0' point of $B_1' B_2'$ fragment in the extended epicycloid.
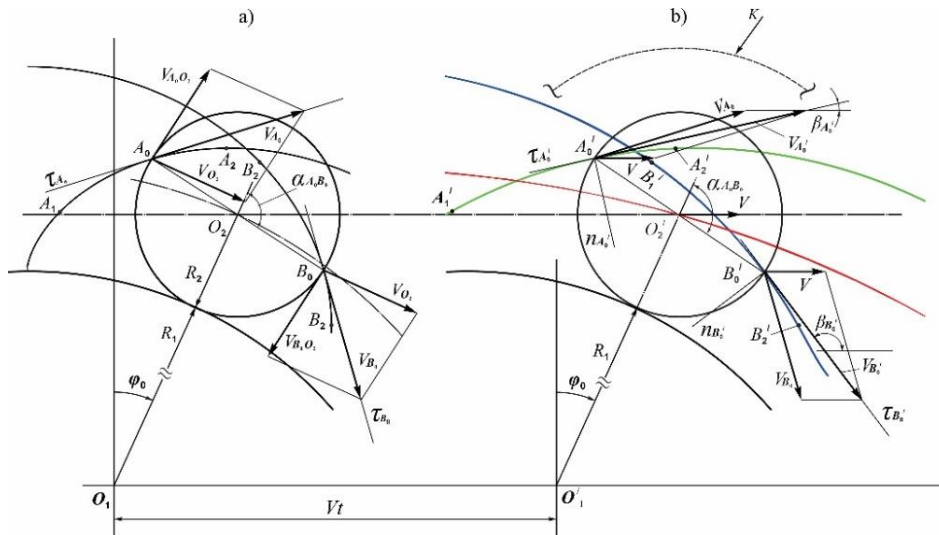

Fig. 6 - The scheme of determination of the velocities of A and B blades attached to the planet gear of the rotary tiller without forward movement (a), with forward movement (b).

Taking into account that during the soil cultivation the total resistance force of the medium affects the tiller's blade in the direction of total velocity vector and has an opposite direction, the scheme of force affecting a planet gear of rotary machine will have the following view (Figure 7). At the observation moment $(\varphi = \varphi_0)$ the line affecting the A blade force $\left(P_{A_0'}\right)$ coincides with the direction of the total velocity $\left(V_{A_0'}\right)$ vector of the fixed point of that blade, has an opposite direction and forms $\beta_{A_0'}$ angle with x axis. The force $P_{B_0'}$ affecting B blade has a direction opposite to $V_{B_0'}$ and forms $\beta_{B_0'}$ angle with x axis.
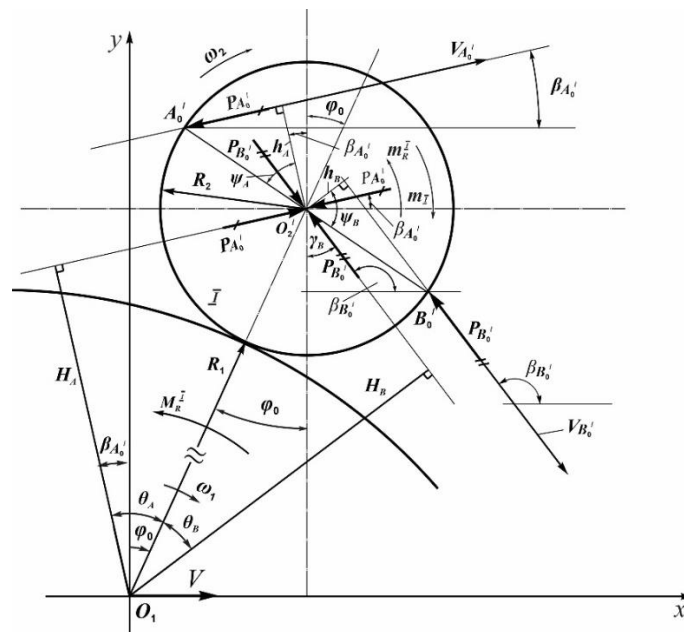

Fig. 7 - The scheme for determining the resistance force factors of the blades attached to the tiller's planet gear

Considering the force system acting on the tiller attached to the planet gear as flat, based on the well-known principles of mechanics *(Loytsyansky L.G., Lurie A.I., 2006)*, it can be stated that the mentioned forces generate force couples, the sum of moments of which will come forth as the resistance moment with respect to the planet gear axis (O'_2) and the forces $P_{A'_0}$ and $P_{B'_0}$ affecting that axis with their respective directions and values. The moment of resistance forces towards the planet gear axis will be:

$$m_R^I = \left(m_{P_{A'_0}}\right) + \left(m_{P_{B'_0}}\right) = P_{A'_0} \cdot h_A + P_{B'_0} \cdot h_B \tag{8}$$

The schemes indicate (Figure 6 and Figure 7) that $h_A = R_2 \cdot cos\psi_A$, $h_B = R_2 \cdot cos\psi_B$
where $\psi_A = \pi - \alpha - \beta_{A'_0} - \varphi_0$; $\psi_B = -\pi + \beta_{B'_0} + \alpha + \varphi_0$.

Considering that $\alpha = i\varphi$, by placing the values in (8), there will be:

$$m_R^I = -P_{A'_0} \cdot R_2 cos\left[(i + 1)\varphi_0 + \beta_{A'_0}\right] - P_{B'_0} \cdot R_2 cos\left[(i + 1)\varphi_0 + \beta_{B'_0}\right] \tag{9}$$

At this stage of problem solution, when assuming that the resistance forces are known, the determination of which depending on the blades geometrical shape and installation parameters, is a matter of individual consideration, the problem of resistance moment identification leads to the determination of $\beta_{A'_0}$ and $\beta_{B'_0}$ angles.

Since the affecting lines of $P_{A'_0}$ and $P_{B'_0}$ resistance forces coincide with the direction of tangents drawn towards the specific point of the blades trajectory, hence:

$$tg\beta_{A'_0} = \frac{dy_A}{dx_A}, \qquad tg\beta_{B'_0} = \frac{dy_B}{dx_B}$$

$tg\beta_{A'_0}$ is determined via Eq. (2) expression. $tg\beta_{B'_0}$ is determined with the same logic. Considering the phasal deviation of A and B points, the equation of the fixed point motion of B blade will have the following form:

$$\begin{cases} x_B = Vt(R_1 + R_2) \cdot sin\omega t + R_2 sin(i + 1)\omega t \\ y_B = (R_1 + R_2) \cdot cos\omega t + R_2 cos(i + 1)\omega t \end{cases} \tag{10}$$

Thus:

$$tg\beta_{B'_0} = \frac{dy_A}{dx_A} = \frac{-\omega[(R_1 + R_2) \cdot sin\omega t + R_2(i + 1)sin(i + 1)\omega t]}{V\omega[(R_1 + R_2) \cdot cos\omega t + R_2(i + 1)cos(i + 1)\omega t]} \tag{11}$$

By placing the identified values of $\beta_{A'_0}$ and $\beta_{B'_0}$ in Eq. (9), the resistance moment towards the planet gear axis can be determined. In their turn, $P_{A'_0}$ and $P_{B'_0}$ forces moved towards the parallel of their acting line and applied to the planet gear rotation axis (O'_2), generate force couples against the driver's rotation axis with $M_{R(P_{A'_0})}^I$ and $M_{R(P_{B'_0})}^I$ moments, which are resistance force factors of affected/impact medium; besides, $P_{A'_0}$ and $P_{B'_0}$ forces are applied to the driver's rotation axis (O) with their preliminary directions.

According to the scheme presented in Figure 7:

$$M_{R(P_{A'_0})}^I = P_{A'_0} \cdot H_A = P_{A'_0} \cdot (R_1 + R_2) \cdot cos(\beta_{A'_0} + \varphi_0)$$

$$M_{R(P_{B'_0})}^I = P_{B'_0} \cdot H_B = -P_{B'_0} \cdot (R_1 + R_2) \cdot cos(\beta_{B'_0} + \varphi_0)$$

The general resistance moment from the *I* planet gear applied in the driver of planetary mechanism will be:

$$M_R^I = P_{A'_0} \cdot (R_1 + R_2) \cdot cos(\beta_{A'_0} + \varphi_0) - P_{B'_0} \cdot (R_1 + R_2) \cdot cos(\beta_{B'_0} + \varphi_0) \tag{12}$$

Similarly for the II and III planet gears, and given that they are shifted against the I planet gear by 120[0] and 240[0], for A and B blades there will respectively be:

II planet gear, A blade:

$$tg\beta_A = \frac{-\omega R_2(i + 1)\{cos(60 - \varphi) + cos[(i + 1)\varphi - 60]\}}{V\omega R_2(i + 1)\{sin(60 - \varphi) - sin[(i + 1)\varphi - 60]\}} \tag{13}$$

B blade:

$$tg\beta_B = \frac{-\omega R_2(i + 1)\{cos(60 - \varphi) - cos[(i + 1)\varphi - 60]\}}{V\omega R_2(i + 1)\{sin(60 - \varphi) + sin[(i + 1)\varphi - 60]\}} \tag{14}$$

III planet gear, A blade:

$$tg\beta_A = \frac{-\omega R_2(i+1)\{cos(\varphi + 30) - cos[150 - \varphi(i+1)]\}}{V \mp \omega R_2(i+1)\{sin(\varphi + 30) + sin[150 - \varphi(i+1)]\}}. \tag{15}$$

B blade:

$$tg\beta_B = \frac{-\omega R_2(i+1)\{cos(\varphi + 30) + cos[150 - \varphi(i+1)]\}}{V \mp \omega R_2(i+1)\{sin(\varphi + 30) - sin[150 - \varphi(i+1)]\}}. \tag{16}$$
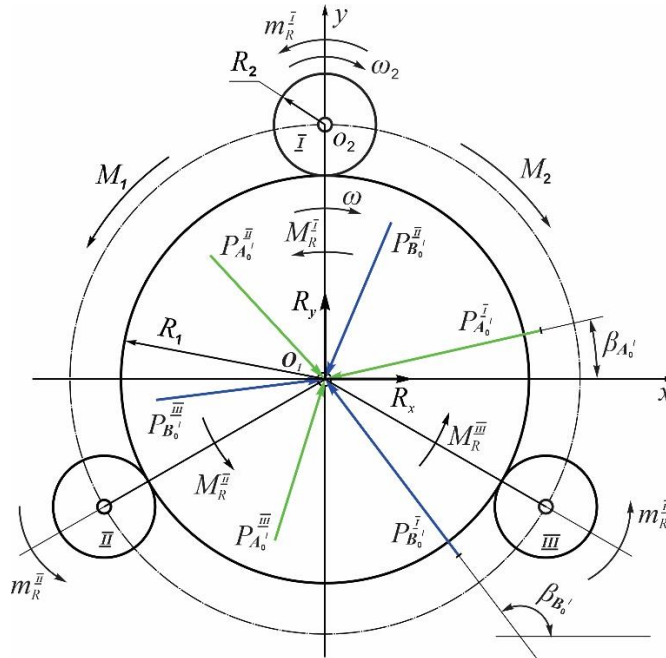

Fig. 8 - The scheme of resistance forces affecting the rotary tiller with planetary mechanism

The resistance moments m_R^{II}; M_R^{II}; m_R^{III} and M_R^{III} are determined through Equations (9) and (12) expressions, by placing therein the values of $tg\beta_A$ and $tg\beta_B$ from Equations (13) (14) (15) and (16) expressions, respectively. The scheme of resistance forces affecting the rotary tiller with planetary mechanism equipped with three planet gear arranged at 120^0 looks as follows (Figure 8).

Since the resistance moments (m_R^i and M_R^i) applied to the planet gears and drivers are acting within one plane, thus the total resistance moment or the main resistance moment applied to the driver's shaft will be determined in the following way:

$$M_1 = m_R^I + M_R^I + m_R^{II} + M_R^{II} + m_R^{III} + M_R^{III} \tag{17}$$

Having the value of the resistance moment (M_1) applied to the shaft of the driving rotor in the rotary tiller, it will be possible to determine the required power of the motor and the moment (M_2) applied to the shaft of transmission gear.

Regarding the forces applied to the driver $P_{A_0}^I$; $P_{B_0}^I$; $P_{A_0}^{II}$; $P_{B_0}^{II}$; $P_{A_0}^{III}$; $P_{B_0}^{III}$, they form a planar convergent force system, the vector sum of which serves as the main vector R applied to the axis of the driver's shaft.

Considering the scheme depicted in Figure 8 and the expressions of Equations (7), (11) and (14), it becomes easily clear that in case of insignificant difference between the resistance forces applied to the tiller's blades, which is empirically approved, the forces applied to the driving rotor are balanced, which practically leads to the abrupt reduction in the traction resistance of the aggregate (tractor).

The aforementioned is one of the most considerable results in the application of the planetary mechanism of the tiller's transmission gear. Thus, the energy indicators of the machine are foremost and mainly related to the resistance moment loading. Taking into account the hypothesis about the equality of the resistance forces applied to the tiller's blade, the resistance moment applied to the rotor's shaft of the rotary machine can be practically determined in the following way per the expression Eq. (17):

$M_1 = (M_R + m_R)$ or by inserting the values of M_R and m_R, it will be obtained:

$$M_1 = 3P \cdot R_2 \left((i+1)[cos(\beta_{A'_0} + \varphi_0) \cdot cos(\beta_{B'_0} + \varphi_0)] - cos[(i+1)\varphi_0 + \beta_{A'_0}] + cos[(i+1)\varphi_0 + \beta_{B'_0}] \right). \tag{18}$$

After the identification of the resistance moment applied to the shaft of the driver's rotor, another priority issue is to determine the optimal geometric and kinematic parameters of the transmission chains, in the case of which the uniformity of the mentioned moment under the operational cycle will be maximum or the changing amplitude will be minimum. The solution to this problem will be presented in the next article.

CONCLUSIONS

1. For the rotary tiller with vertical axis developed for the tillage of inter-trunk and near-trunk spaces of orchards and vineyards a planetary mechanism with immobile sun gear, external planet gears and driving rotor (carrier) has been selected as a driving mechanism, which enabled to provide velocities in multiple times higher than the tiller's blades transmission number in case of relatively smaller rotation numbers of the rotor (e.g. in case of 50 rpm rotation number of the rotor 250 rpm of the tiller has been received.

2. In the result of kinematic analysis of the mechanism, analytical expressions have been derived, which enable to establish a relationship between the transmission number of rotation frequency of driving rotor and tiller's blade velocities and accelerations.

3. In the result of kinematic research of planetary transmission mechanism of the rotary tiller analytical expressions have been derived based on which force analysis of working parts have been conducted. Analytical expressions for the determination of the tiller's and rotor's resistance moments have been received, which enable to solve the problem of ensuring their minimum optimal values.

4. The expressions obtained in the result of dynamic research of the recommended mechanism enable to identify the optimal geometric parameters of the tillers disc blades and their mutual arrangements, in case of which the resistance forces of the cultivated medium will be at minimum.

ACKNOWLEDGEMENTS

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