DYNAMIC-ENERGY ANALYSIS OF A ROTARY TILLER WITH A VERTICAL ROTATION AXIS

ՈՒՂՂԱՁԻԳ ՊՏՏՄԱՆ ԱՌԱՆՑՔՈՎ ՀՈՂԱՄՇԱԿ ՖՐԵԶԻ ԴԻՆԱՄԻԿԱԿԱՆ-ԷՆԵՐԳԵՏԻԿԱԿԱՆ ՀԵՏԱԶՈՏՈՒԹՅՈՒՆ

Arshaluys TARVERDYAN[1)], Artur ALTUNYAN[1*)], Gevorg HARUTYUNYAN[2)], Albert GRIGORYAN[1)]

[1)]Scientific Research Institute for Agricultural Mechanization and Automation, Armenian National Agrarian University, Yerevan, Armenia

[2)]Aspirant at Agricaltural Engineering Faculty

Tel: +37455327325; E-mail: arturaltunyan@gmail.com

Corresponding author: Artur Altunyan

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ABSTRACT

The article considers the issue of dynamic and energy analysis of the rotary tiller with planetary transmission gear and vertical rotation axis designed for orchards and vineyards. In the result of investigations, analytical expressions have been derived, which enable to determine the resistance force factors of the cultivated medium affecting the rotary tiller and to set up their changing patterns during a single rotation of the rotor in conditions of forward movement of the aggregate/working unit. It has been confirmed that in case of binary symmetric blades located at 120 degree, the resistance forces transmitted from the tiller satellites to the rotor shaft mutually balance each other, as a result of which the traction resistance force of the unit receives the possible minimum value. As regard to the resistance moments transmitted from the tiller satellites to the rotor shaft and to their main moments, it should be noted that they change significantly during a single rotation of the rotor. Particularly, the extreme values of the main moment for each tiller satellite are obtained at the finite points of the rotor's diameter perpendicular to the forward movement of the tillage machine. Based on the driving and resistance force factors of the machine, as well as the differential equation of the motion, the term has been identified by means of the energy research of the rotary tiller, in case of which the angular acceleration of the rotor's shaft will be equal to zero and the machine will operate without oscillations and vibrations.

ԱՄՓՈՓԱԳԻՐ

Հոդվածում դիտարկվում է պլանետարային հաղորդակով, ուղղաձիգ պտտման առանցքով պտղատու և խաղողի այգիների հողամշակ ֆրեզի դինամիկական և էներգետիկ վերլուծության խնդիրը: Հետազոտությունների արդյունքում ստացվել են անալիտիկ արտահայտություններ, որոնք հնարավորություն են տալիս որոշելու հողամշակ ֆրեզի վրա ազդող մշակվող միջավայրի դիմադրության ուժային գործոններն ու սահմանելու դրանց փոփոխման օրինաչափությունները ռոտորի մեկ պտույտի ընթացքում, ագրեգատի համընթաց շարժման պայմաններում: Հաստատվել է, որ 120 աստիճանի տակ տեղակայված երկուական սիմետրիկ դանակների դեպքում սատելիտ ֆրեզներից ռոտորի լիսեռին փոխանցվող դիմադրության ուժերը փոխադարձ հավասարակշռում են իրար, ինչի հետևանքով ագրեգատի քարշային դիմադրության ուժը ստանում է հնարավոր նվազագույն արժեքը: Ինչ վերաբերվում է ֆրեզ-սատելիտներից ռոտորի լիսեռին փոխանցվող դիմադրության մոմենտներին և դրանց գլխավոր մոմենտին, ապա դրանք ռոտորի մեկ պտույտի ընթացքում փոփոխվում են զգալի չափով: Մասնավորապես գլխավոր մոմենտն իր էքստրեմալ արժեքները, յուրաքանչյուր ֆրեզ-սատելիտի համար, ստացվում են ֆրեզ-մեքենայի համընթաց շարժմանն ուղղահայաց ռոտորի տրամագծի ծայրակետերում: Առաջարկվող պլանետարային հաղորդակով հողամշակ ֆրեզի էներգետիկական հետազոտության միջոցով, հիմք ընդունելով մեքենայի շարժող և դիմադրության ուժային գործոնները, ինչպես նաև շարժման դիֆերենցիալ հավասարումը, սահմանվել է պայմանը, որի դեպքում ռոտորի լիսեռի անկյունային արագացումը հավասար կլինի 0-ի, մեքենան կաշխատի առանց ցնցումների և վիբրացիայի:

INTRODUCTION

The inter-row cultivation of gardens, as well as inter-trunk and near-trunk cultivation of trees in accordance with the agrotechnical requirements is an essential prerequisite for the yield capacity increase and quality improvement of the product in orchards. Furthermore, the latter are ensured by the accurate and timely implementation of the mentioned agrotechnical activities *(Manaenkov, 2017; Kupryashkin & Gusev, 2020, Khort et al., 2020; Bordoni et al., 2019).*

The operational practice of rotary tillers in the inter-trunk and inter-row spaces has enabled to develop their main agrotechnical requirements *(Panov & Tokushev, 2005; Koval, 2010; Tarverdyan et al., 2024)* and at the same time to state that the machines with active working parts are more effective. These are mainly rotary tillers with horizontal or vertical rotation axis equipped with mechanical or hydraulic transmission gears *(Parkhomenko,2012; Kupryashkin & Gusev, 2020; Tarverdyan et al., 2024; Panov, 1963; Byshov, 2017; Tarverdyan & Sargsyan, 2015, Hegazy et al., 2014; Abo-Habaga et al., 2018).*

Their structural peculiarities enable to maximally adapt the machine to the cultivation of inter-trunk spaces observing the implementation rules of agrotechnical requirements. The distinctive advantages of machines of this series are introduced in many research works *(Kupryashkin & Gusev, 2020; Byshov, 2017; Tarverdyan & Sargsyan, 2015; Mosatovskiy, 1980; Ahmadi, 2017; Raparelli et al., 2021).*

However, the operational practice of the tillers with vertical axis and the analysis of literature data signify that they are not free from defects either *(Panov & Tokushev, 2005; Panov, 1963; Tarverdyan & Sargsyan, 2015)*.

Particularly, in this case the thrown out mass of cultivated soil is rather large, and at the same time it is mentioned that the way of elimination of this shortcoming is the correct selection of the shape and geometrical parameters of the tiller's blades *(Mosatovskiy, 1980)*.

Considering also the circumstance that the operational reliability of the existing machines is low, especially in the soil types of the Republic of Armenia, which stand out for high content of rocks and gravel, an attempt was made to develop rotary tillers with vertical rotation axis possibly free from the above stated shortcomings and suitable for the cultivation of inter-tree, inter-trunk and inter-vine spaces in orchards and vineyards.

The attempts to solve the mentioned problem through optimization of the shape and profile of the existing tiller's blades with vertical rotation axis failed to bring out the expected results, though partial improvement in some indicators has taken place *(Kupryashkin & Gusev, 2020; Koval, 2010; Byshov, 2017; Chatkin, 2007; Wusong et. al, 2024; Li S.T. et al, 2016)*. Based on the afore stated, the goal and objectives of the presented research work has been developed.

The key objective of the current work was to develop a rotary tiller with vertical rotation axis, which could enable to cultivate the inter-row and inter-trunk spaces in orchards with minimum energy consumption, meanwhile ensuring minimum horizontal movement of the soil mass throughout the cultivation process.

It is apparent that the only way to solve the problem is to make fundamental changes in the tiller's structure.

MATERIALS AND METHODS

Based on the results of our previous investigations *(Tarverdyan et al, 2024; Tarverdyan & Hayrapetyan, 2019)*, an attempt was made to develop a transmission gear of a tiller with vertical axis, which will be able to provide high rotation numbers of working tillers (200–250 rpm) in case of relatively lower rotation numbers of the rotor (50–80 rpm). It is a planetary mechanism of external gear with immobile/fixed annular/sun wheel (Fig. 1).

It consists of an annular fixed wheel (3), mounted to the support (2) attached to the tiller's frame (1), satellites (4) rolled thereto, the rotation axis of which is installed in the grooves of rotor housing/casing (5) with bearings. The tillers discs (6) are rigidly attached to the free ends of the rotation axes of the satellites (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 via belt or chain transmission (8). The rotational motion is transmitted from the power take-off shaft of the tractor or hydraulic motor to the rotor (5) of the rotary tiller which in this case acts as a driver of planetary mechanism; due to the rotor's rotation the satellites (4) roll over the ring belt of the fixed annular wheel (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, blades' linear speeds/velocities of the rotary tillers (working parts) with small diameter in case of relatively small rotation numbers of the rotor.

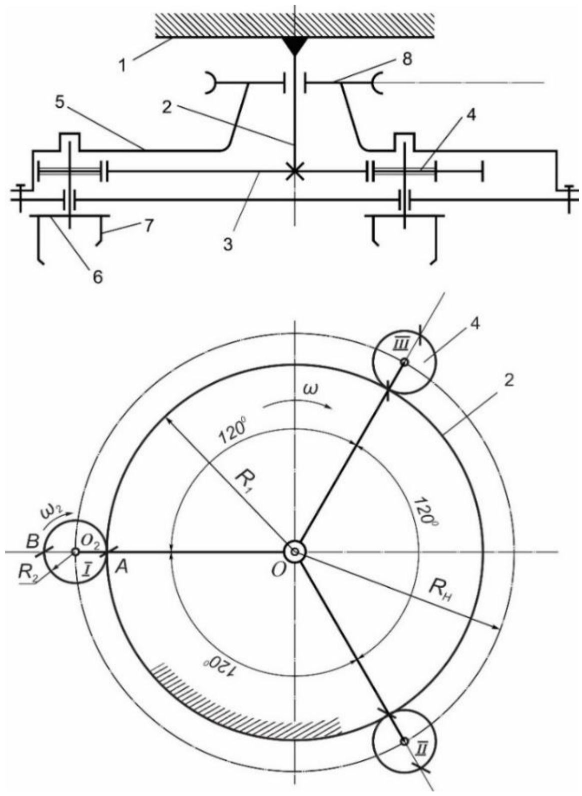


Fig. 1 – Schematic diagram of a rotary tiller with a planetary drive for the cultivation of near-trunk and inter-trunk areas of orchards

The kinematic analysis of the proposed mechanism was conducted in the first article of the series related to the discussed problem *(Tarverdyan et al, 2024)*, and the expressions of the trajectory Equation (1), velocities Equation (2) and accelerations Equation (3) for the arbitrary point of the tiller's blades of the machine were derived:

$$\begin{cases} x = V \cdot t \pm (R_1 + R_2) \cdot sin\omega t - R_2 sin(i+1)\omega t \\ y = (R_1 + R_2) \cdot cos\omega t - R_2 cos(i+1)\omega t \end{cases} \tag{1}$$

$$\begin{cases} V_x = V \pm \omega[(R_1 + R_2) \cdot cos\omega t - R_2(i+1) \cdot cos(i+1)\omega t] \\ V_y = -\omega[(R_1 + R_2) \cdot sin\omega t - R_2(i+1) \cdot sin(i+1)\omega t] \end{cases} \tag{2}$$

$$\begin{cases} a_x = \mp\omega^2[(R_1 + R_2) \cdot sin\omega t - R_2(i+1)^2 \cdot sin(i+1)\omega t] \\ a_y = -\omega^2[(R_1 + R_2) \cdot cos\omega t - R_2(i+1)^2 \cdot cos(i+1)\omega t] \end{cases} \tag{3}$$

where: V_x, V_y are the speed components of any fixed point for the tiller's blade [m/s], a_x, a_y - acceleration components of any fixed point for the tiller's blade [m/s^2], *V* - velocity of forward movement of the aggregate [m/s], *t* - time of movement [s], R_1 - radius of relatively fixed sun gear, [m], R_2 - planet gear radius, [m], $i = \frac{R_1}{R_2}$ – transmission ratio, ω - angular speed of the planetary mechanism driver [min^{-1}].

Based on the results of kinematic studies of the mechanism, a force analysis of the mechanism was performed, analytical expressions were obtained, which enable to determine the resistance force factors affecting the blades of the tilling machine depending on the kinematic and geometric parameters of the machine *(Tarverdyan et al, 2024).*

In particular for the option of the rotary tiller with planetary transmission gear equipped with three satellites at 120^0, based on the principles of mechanics *(Zinoviev, 1975; Loytsyansky & Lurie, 2006)*, the scheme of resistance force factors (Fig.2) affecting the rings of the tilling machine and their determination expressions *(Tarverdyan et al, 2024)* have been developed.

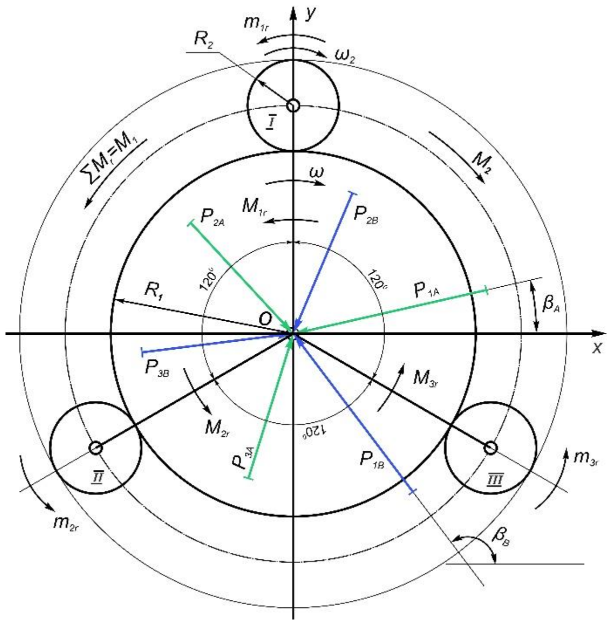


Fig. 2 – Scheme of resistance force factors affecting the rotary tiller with planetary mechanism

Since the resistance moments (m_r^k *and* M_r^k) applied to the satellites and drivers act in one plane, hence, the total resistance or the main resistance moment applied to the drive shaft will be determined in the following way *(Tarverdyan et al., 2024, eq. 17)*:

$$M_1 = m_r^I + M_r^I + m_r^{II} + M_r^{II} + m_r^{III} + M_r^{III}, \tag{4}$$

where m_r is the the moment of resistance forces towards the planet gear axis [N.m], M_r – moment of resistance forces towards the planetary mechanism drive axis [N.m].

Having the value of the resistance moment (M_1) applied to the driving rotor's shaft of the rotary tiller, it will be possible to determine the torque/moment (M_2) applied to the drive shaft and the power of the engine.

The resistance moment ($m_r^{(k)}$) applied to one satellite (k) and the resistance moment ($M_r^{(k)}$) applied to the rotor's shaft therefrom are determined through the following expressions *(Tarverdyan et al., 2024)*.

$$m_r^{(k)} = -P_A \cdot R_2 cos[(i+1)\varphi + \beta_A] - P_B \cdot R_2 cos[(i+1)\varphi + \beta_B]. \tag{5}$$

$$M_r^{(k)} = P_A \cdot R_2 (i+1) cos(\varphi + \beta_A) - P_B \cdot R_2 (i+1) cos(\varphi + \beta_B). \tag{6}$$

For the A and B blades of conditional I, II and III satellites $tg\beta_A$ and $tg\beta_B$ are determined with the following expressions (in the formulae the sign of numerator refers to A blade, and denominator - to B blade):

I satellite

$$\left.\begin{aligned} &\text{A } blade\text{:}\ \ tg\beta_A = -\frac{\omega[R_H sin\varphi - R_2(i+1)sin(i+1)\varphi]}{V \pm \omega[R_H cos\varphi - R_2(i+1)cos(i+1)\varphi]} \\ &\text{B } blade\text{:}\ \ tg\beta_B = -\frac{\omega[R_H sin\varphi + R_2(i+1)sin(i+1)\varphi]}{V \pm \omega[R_H cos\varphi + R_2(i+1)cos(i+1)\varphi]} \end{aligned}\right\} \tag{7}$$

II satellite

$$\left.\begin{aligned} &\text{A } blade\text{:}\ \ tg\beta_A = \frac{-\omega R_2(i+1)\{cos(60-\varphi) + cos[(i+1)\varphi - 60]\}}{V \pm \omega R_2(i+1)\{sin(60-\varphi) - sin[(i+1)\varphi - 60]\}} \\ &\text{B } blade\text{:}\ \ tg\beta_B = -\frac{-\omega R_2(i+1)\{cos(60-\varphi) - cos[(i+1)\varphi - 60]\}}{V \pm \omega R_2(i+1)\{sin(60-\varphi) + sin[(i+1)\varphi - 60]\}} \end{aligned}\right\} \tag{8}$$

III satellite

$$\left.\begin{aligned} &\text{A } blade\text{:}\ \ tg\beta_A = -\frac{\omega R_2(i+1)\{cos(\varphi+30) - cos[150 - \varphi(i+1)]\}}{V \mp \omega R_2(i+1)\{sin(\varphi+30) + sin[150 - \varphi(i+1)]\}} \\ &\text{B } blade\text{:}\ \ tg\beta_B = -\frac{\omega R_2(i+1)\{cos(\varphi+30) + cos[150 - \varphi(i+1)]\}}{V \mp \omega R_2(i+1)\{sin(\varphi+30) - sin[150 - \varphi(i+1)]\}}; \end{aligned}\right\} \tag{9}$$

As regard to the forces applied to the drive shaft – $P^{I}_{A_0'}$; $P^{I}_{B_0'}$; $P^{II}_{A_0'}$; $P^{II}_{B_0'}$; $P^{III}_{A_0'}$; $P^{III}_{B_0'}$, they form a planar convergent force system, the vector sum of which comes forth as the main vector R applied to the drive shaft.

In case of small differences in the resistance forces applied to the tiller blades, which is confirmed via empiric data, the forces applied to the drive rotor are balanced (R=0), which practically leads to the abrupt reduction in the traction resistance of the unit.

Thus, the energy indicators of the machine are mainly and primarily related to the resistance moment loads. Taking into account the hypothesis about the equality of resistance forces applied to the tillers' blades, the resistance moment applied to the rotor's shaft of the rotary tiller can be practically determined in the following way:

$M_1 = \sum_{k=I}^{III}(M_r^k + m_r^k)$ or inserting the values of $m_r^{(k)}$ *and* $M_r^{(k)}$ from Equations (5) and (6), it will be obtained:

$$M_1 = PR_2 \cdot \sum_{k=I}^{III}\left\{(i+1)cos\left(\varphi+\beta_{\mathrm{A}}^{(k)}\right) - (i+1)cos\left(\varphi+\beta_{\mathrm{B}}^{(k)}\right) - cos\left[(i+1)\varphi+\beta_{\mathrm{A}}^{(k)}\right] - \cos\left[(i+1)\varphi+\beta_{\mathrm{B}}^{(k)}\right]\right\} \tag{10}$$

It is necessary to take into account that when determining the total resistance moment (M_1) through Equation (10) expression it is required to calculate the values of β_{A}^k and β_{B}^k in the expressions for each satellite and place it in Equations (7), (8) and (9) expressions, respectively.

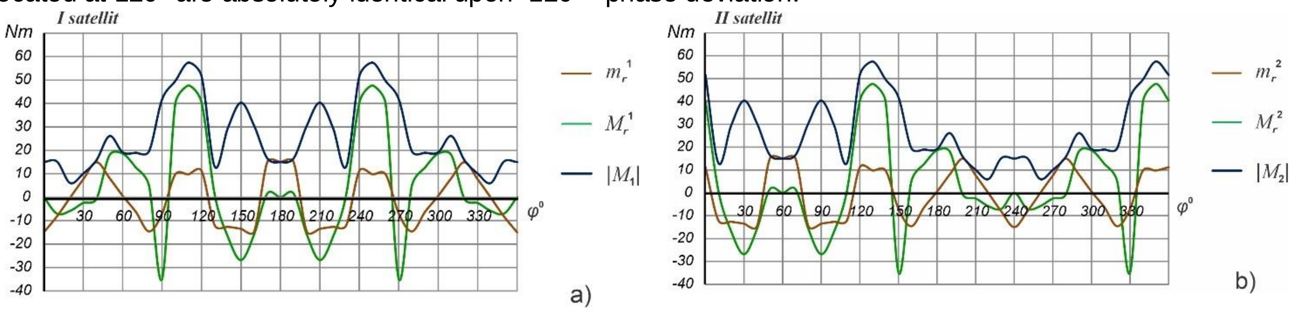
RESULTS AND ANALYSIS

To present the changing patterns of resistance moments applied to the satellite-tiller and drive rotor shaft more vividly, let's build the graphs of their changes during a single rotation of the rotor per the expressions of Equations (5), (6) and (10) in case of the following numerical values: V=1 m/s, ω=5 s^{-1}, ω_2=25 s^{-1}, i=4, R_1=0.2 m, R_2=0.05 m.

Considering the introduced data of *(Konstantinov, 2019*; *Kupryashkin & Gusev, 2020; Tarverdyan & Sargsyan, 2015; Akimov, 2017; Ahmadi, 2017)* in the conducted research works, it is conditionally assumed that P=150 N. The graphs designed with the mentioned data are presented in Figure 3.

It is noteworthy that the resistance forces applied to the satellite blades during the tiller operation change their signs, anyhow, the directions of the resistance moments raised from those forces stay unchanged; this is the reason why the total resistance moments are introduced in absolute values *(Zinoviev, 1975).*

It follows from Fig. 3 that the variation patterns of resistance moments applied to the three satellites located at 120^0 are absolutely identical upon 120^0 –phase deviation.



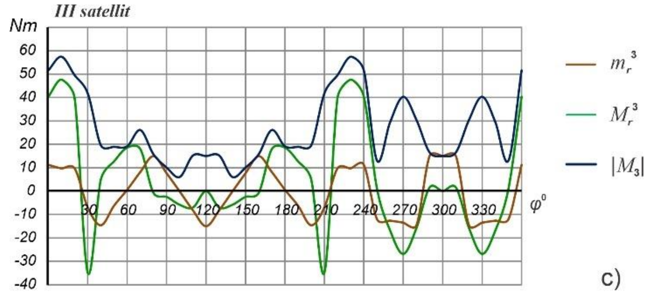


Fig. 3 – Changing graphs of resistance moments affecting the satellite-tiller (m_s^i), rotor's shaft of the satellite-tiller (M_s^k) and the module of total resistance moment ($|M_k|$) of the rotary tiller:
a) –satellite I, b) - satellite II and c) - satellite III.

In Figure 4 the changing graph of total resistance moment $(\sum M)$ applied to the drive-rotor's shaft of the tilling machine is depicted. It follows from the graph that from the standpoint of energy and uniform work of the tiller the extreme values of total resistance moment, being at the top interest point, are formed in the finite diameter points perpendicular towards the forward movement of the aggregate.

After the determination of resistance moment applied to the shaft of the drive rotor, it is crucial to identify the optimal geometrical and kinematic parameters of the transmission gear rings, in case of which the uniformity of the mentioned moment during the working cycle will be maximum or the changing amplitude will be minimum.

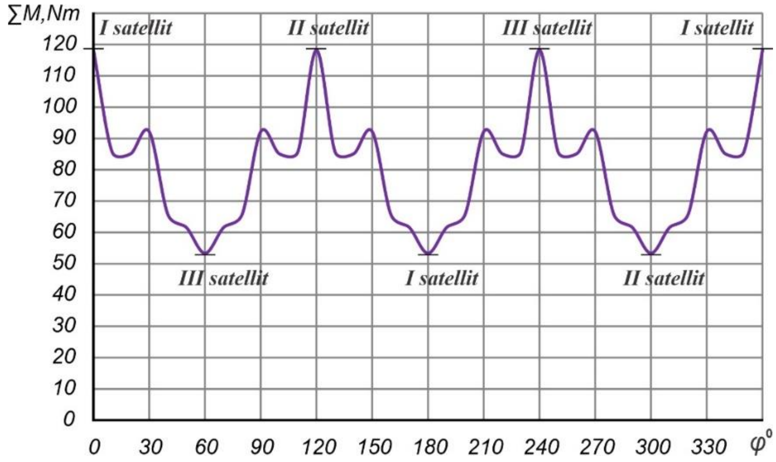


Fig. 4 – Changing graph of total resistance moment affecting the rotor's shaft of the rotary tiller with three satellite tillers located at 120^0

It is obvious that in order to ensure the intended stable work mode of the machine, it is necessary to apply force couple with M_2 moment, which should be equal to the total resistance moment of the soil environment $\sum M_r$, in large or marginal case (for practicality, let's denote $\sum M_r$ as M_1 in further expressions). To solve the proposed problem, it is necessary to perform energy analysis of the rotary tiller with planetary transmission gear. The analysis was conducted according to the scheme presented in Figure 5. The expression of kinetic energy of the mechanism looks as follows according to well-known theories of classical mechanics *(Zinoviev, 1975; Loytsyansky & Lurie, 2006)*:

$$W = (W_1 + 3W_2 + W_V), \tag{11}$$

where $W_1 = \frac{\mathcal{J}_1 \omega^2}{2}$ is the kinetic energy of the drive rotor (1, Fig. 5),

$W_2 = \frac{\mathcal{J}_2 \omega_2^2}{2} + \frac{m_2 V_{02}^2}{2}$ is the kinetic energy of satellite (the satellite performs complex motion: rotational and forward), $W_V = \frac{m_1 V^2}{2}$ is the kinetic energy of the forward movement of the rotor (general mass).

The kinetic energy of the rotary tiller along the driving ring of the mechanism, i.e., drive-rotor's shaft will be:

$$W_r = \frac{\mathcal{J}_r \omega^2}{2} \tag{12}$$

where $\mathcal{J}_r$- is the resulting moment of inertia.

Considering that $\omega_2 = (i+1)\,\omega$, V_{O_2} will be $V_{O_2} = R_2(i+1)\omega$, for the forward movement of the unit, based on the conservation term of kinematic parameters, in case of three satellites installed at 120^0 , the following expression has been obtained (*Tarverdyan et al., 2024)*: $V \leq 6\omega(i+1)R_2$.

Placing the values of the above-mentioned quantities in Equation (11) and equating the right parts of Equations (11) and (12), it is obtained:

$$\frac{\mathcal{J}_\text{p}\omega^2}{2} = \frac{\mathcal{J}_1\omega^2}{2} + \frac{3\mathcal{J}_2(i+1)^2\omega^2}{2} + \frac{3m_2R_2^2(i+1)^2\omega^2}{2} + \frac{36m_1R_2^2(i+1)^2\omega^2}{2},$$

wherefrom:

$$\mathcal{J}_\text{p} = \mathcal{J}_1 + 3(i+1)^2(\mathcal{J}_2 + m_2R_2^2 + 12m_1R_2^2), \tag{13}$$

where $\mathcal{J}_1$ is the moment of rotor's inertia: $\mathcal{J}_1 = \frac{P_1 R_\text{D}^2}{2g}$ ($m_1 = \frac{P_1}{g}$ is the rotor's mass, P_1 is the total rotor's weight, R_D is the radius of rotor-drive case), $\mathcal{J}_2$ is the satellite's inertia moment: $\mathcal{J}_2 = \frac{P_2 R_2^2}{2g}$ ($m_2 = \frac{P_2}{g}$ is the satellite mass).

Taking into account the above stated expressions and denotations, the kinematic energy of the mechanism will be determined through the following expression:

$$W = \frac{\omega^2}{4g}[P_1 R_D^2 + 6(i+1)^2 \cdot (P_2 R_2^2 + 6P_1 R_2^2)]. \quad (14)$$

To get the equation of the motion of the mechanism, the Lagrange's equation is used for the generalized coordinate φ *(Zinoviev, 1975; Loytsyansky & Lurie, 2006).*

$$\frac{d}{dt}\frac{\partial W}{\partial \dot{\varphi}} - \frac{\partial W}{\partial \varphi} = P_\varphi. \quad (15)$$

In the discussed problem, the position of any ring of the mechanism is determined through the generalized φ coordinate, and in the result of force analysis the resistance moment M_1 was already determined, hence the driving force factor M_2.

Obviously in Equation (15) expression, it can be accepted that the moment M is risen by the active forces as a generalized force factor (P_φ). Since the active forces come forth as both driving and resistant ones, the resulted moment can be presented as the difference in the force factors of driving and resistance moments (*Zinoviev, 1975).*

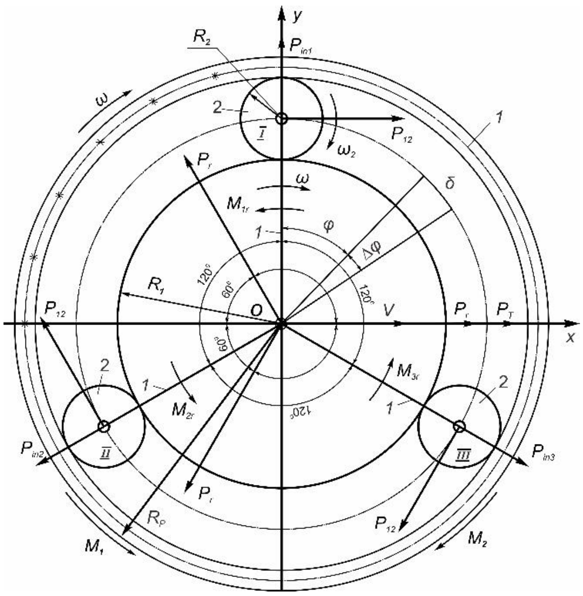


Fig. 5 – Scheme for the dynamic study of the planetary driving mechanism of a rotary tiller.

Besides, the traction resistance force P_T (Figure 5) of the aggregate and satellites' inertial forces P_{in} should be considered as important force factors. Inertial forces of the satellites (centrifugal forces), $P_{in1} = P_{in2} = P_{in3} = m_2\omega^2 \cdot (R_1 + R_2)$, which are equal in modulus, are directed towards each other at 120^0 and practically balance each other relative to the axis of the driver.

To receive the analytical expression of the generalized force (P_φ), it is given the generalized coordinate (φ) an elementary increment ($\delta\varphi$) and the elementary work (δA)performed on that elementary displacement is determined.

The work of the traction resistance force performed on the elementary displacement is determined in the following way:

$$\delta A_{(P_T)} = P_T \cdot (R_1 + R_2)\delta\varphi \cdot \, cos\varphi. \quad (16)$$

Since the traction resistance force is directed towards x axis, and $(R_1 + R_2)\delta\varphi \cdot \, cos\varphi$ is the projection of the elementary displacement on the same axis.

As regard to the satellites' inertial forces, they mutually balance each other relative to the driver's axis and besides, their work performed on the elementary displacement $(R_1 + R_2)\delta\varphi$ is equal to zero, since those forces are perpendicular towards elementary displacement.

Not only the inertial but also the resistance forces (P_r) applied to the rotor shaft are mutually balanced. This circumstance is paramount importance in terms of uniform operation of the rotary tiller and abrupt decline in the traction resistance of the unit. This is one of the most important findings of the planetary transmission gear.

The elementary work performed by the active force factors will be determined through the following expression in the discussed case:

$$\delta A = [M_2 - M_1 - P_T \cdot (R_1 + R_2) \cdot \, cos\varphi] \cdot \delta\varphi \tag{17}$$

.

Since according to definition $\delta A = P_\varphi \cdot \delta\varphi$, hence:

$$P_\varphi = M_2 - M_1 - P_T (R_1 + R_2) \cdot \, cos\varphi. \tag{18}$$

The partial derivative of the kinetic energy expression Equation (14) per the generalized ($\dot{\varphi}$) speed will be:

$$\frac{\partial W}{\partial \dot{\varphi}} = \frac{1}{2g}[P_1 R_D^2 + 6(i+1)^2 \cdot (P_2 R_2^2 + 6P_1 R_2^2)] \cdot \dot{\varphi}. \tag{19}$$

When deriving the obtained expression Equation (19) per time, it will be obtained:

$$\frac{d}{dt} \cdot \frac{\partial W}{\partial \dot{\varphi}} = \frac{1}{2g}[P_1 R_D^2 + 6(i+1)^2 \cdot (P_2 R_2^2 + 6P_1 R_2^2)] \cdot \ddot{\varphi}. \tag{20}$$

By placing the right parts of Equations (18) and (20) expressions in the Lagrange expression Equation (15) and considering that $\frac{\partial W}{\partial \varphi} = 0$ (the kinematic energy of the mechanism according to Equation (14) expression is not dependent on the generalized coordinate) the differential equation for the movement of the proposed mechanism will be obtained:

$$\frac{1}{2g}[P_1 R_D^2 + 6(i+1)^2 \cdot (P_2 R_2^2 + 6P_1 R_2^2)] \cdot \ddot{\varphi} = M_2 - M_1 - P_T (R_1 + R_2) \cdot \, cos\varphi,$$

wherefrom the angular acceleration of the drive rotor will be:

$$\ddot{\varphi} = \frac{2g[M_2 - M_1 - P_T (R_1 + R_2) \cdot \, cos\varphi]}{P_1 R_D^2 + 6(i+1)^2 \cdot (P_2 R_2^2 + 6P_1 R_2^2)} \tag{21}$$

For the uniform work of the rotor's drive, and hence, for that of rotary tiller, it is required that its angular acceleration-$\ddot{\varphi} = 0$, i.e., the driving moment (M_2) applied to the rotor, should balance the resistance forces, that is:

$$M_2 = M_1 - P_T (R_1 + R_2) \cdot \, cos\varphi. \tag{22}$$

To use the received expression Equation (22) in practice, it is necessary to consider not only the theoretical research results Equation (10) for the definition of M_1 changing patterns, but also first of all the real changing patterns and quantities resulted from the scientific experimental research. In this regard, it is worthwhile mentioning once more, that in case of rotary tiller the real resistance force factors of the medium is possible to determine only empirically then comparing them with the theoretically obtained results. Only in this way is it possible to apply this expression Equation (22) and determine the minimum value of the driving power in case of which the work would be implemented uniformly without any undesired shocks and vibrations.

CONCLUSIONS

1. In the result of the dynamic analysis of the rotary tiller with planetary transmission gear, analytical expressions have been derived, which make it possible to determine the resistance force factors of the medium theoretically, which affect the shaft of satellite tillers and rotor drive and particularly the total resistance moment and its changing pattern throughout a single rotation of the rotor.

2. As a result of dynamic and energy analysis, the term has been set up, in case of which the ratio of total resistance and driving moment/torque should compensate the lack of angular acceleration ($\ddot{\varphi} = 0$) of the rotor's shaft.

Only then it is possible to identify the minimum value of driving moment or the power, in case of which the working mode of the rotary tiller will be uniform without any fluctuations and vibrations.

3. The transmission gear mechanism of the rotary tiller with three satellites located at 120^0 enables not only to ensure uniform work of the machine, but also most importantly, in this condition, the main vector of the cutting resistance forces in the soil environment equals to zero on the rotor's shaft, as a result of which the traction resistance force of the working unit possibly gets its minimum value.

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REFERENCES

[1] Abo-Habaga M., Imara Z., Okasha M. (2018). Development of a Combine Hoeing Machine for Flat and Ridged Soil, *Journal of Soil Sciences and Agricultural Engineering*, 9(12), pp.817-820. https://doi.org/10.21608/jssae.2018.36548

[2] Akimov A.P., Koketantinov Yu.V., Turovskiy B.V. (2017). Mathematical model of interaction of milling knives with soil (Математическая модель взаимодействия фрезерного ножа с почвой). *Bulletin of the Kazan State Agrarian University,* №*4(46),* pp.67-71. https://doi.org/10.12737/article_5a5f06808b59a5.62332052

[3] Bordoni M., Vercesi A., Maerker M., Ganimede C., Reguzzi M.C., Capelli E., Wei X., Mazzoni E., Simoni S., Gagnarli E., Meisina C., (2019), Effects of vineyard soil management on the characteristics of soils and roots in the lower Oltrepò Apennines (Lombardy, Italy), *Science of The Total Environment*, Vol. 693, https://doi.org/10.1016/j.scitotenv.2019.07.196

[4] Byshov N.V., (2017), Modern technical facilities for working in gardens (Современные технические средства для работы в садах). *Scientific journal of Kuban SAU (Научный журнал Куб ГАУ)*, №134, https://doi.org/10.21515/1990-4665-134-082

[5] Chatkin M.N., (2007) *Kinematics and dynamics of the working parts of the rotary tillers with screw elements (Кинематика и динамика ротационных почвообрабатывающих рабочих органов с винтовыми элементами)*. Publishing house of Mordovia State University, Saransk, 316 p.

[6] Hegazy R.A., Abdelmotaleb, Z.M. Imara, Okasha M.H. (2014), Development and Evaluation of Small-Scale Power Weeder, *Journal of Agricultural Engineering (MJAE)*, Vol. 31, Issue 3, pp. 703-728, https://doi.org/10.21608/MJAE.2014.98430

[7] Iman Ahmadi, (2017), A torque calculator for rotary tiller using the laws of classical mechanics, *Soil and Tillage Research*, Volume 165, Pages 137-143, https://doi.org/10.1016/j.still.2016.08.009

[8] Khort D., Filippov R., Smirnov I., Kutyrev A. and Semichev S., (2020), Automated section for milling soil in near-trunk zones of orchard, *E3S Web of Conferences* 161,01068. https://doi.org/10.1051/e3sconf/20 2016101068

[9] Konstantinov Yu. V., (2019) Calculation method of resistance and resistance moment to the soil cutting with a straight disk blade of the cutter (Методика расчета сопротивления и момента сопротивления резанию почвы прямым пластинчатым ножом фрезы). *Tractors and Agricultural Machinery, vol. 86 №5, p. 31-39*. https://doi.org/10.31992/0321-4443-2019-5-31-39

[10] Koval Zh.L.. (2010) *Development of a rotary tiller with energy-saving cutting working parts (Разработка почвообрабатывающей машины с энергосберегающими фрезерными рабочими органами).* Thesis for the Candidate of Mechanical Sciences, Moscow.

[11] Kupryashkin V.F., Gusev A.Yu., (2020) Justification of the parameters and operating modes of a self-propelled rotary tiller with combined rotation of active working parts (Обоснование параметров и режимов работы самоходной почвообрабатывающей фрезы с комбинированным вращеним активных рабочих органов). *Scientific Review. International scientific and practical journal*. Saransk, -pp. 42-48. №2,

[12] Li S.T., Yang L., Niu P., et al (2016). Design and study on the edge curve of blade of a handheld tiller's rotary blade, *INMATEH-Agricultural Engineering*, 51(3): 5-12, Bucharest, Romania.

[13] Loytsyansky L.G., Lurie A.I., (2006), *Course of theoretical mechanics (Курс теоретической механики)*, volume 2., Moscow

[14] Manaenkov K.A., (2017), Improving soil cultivation in near- trunk strips of intensive gardens (Совершенствование обработки почвы в приствольных полосах интенсивных садов). *Technologies of food and processing industry.(Технологии пищевой и перерабатывающей промышленности)*, №3 (17) pp. 28-34

[15] Mosatovskiy V.B., (1980), *Study of the process of near-trunk spaces cultivation in intensive gardens with vertical tillers and justification of the types and parameters of their working parts (Исследование процесса обработки приствольных полос в интенсивных садах вертикальними фрезами и обоснование типов и параметров их рабочих прганов).* Thesis for the candidate of technical sciences, Chisinau.

[16] Panov I.M., (1963), *Rotary tillage machines and tools (Ротационные почвообрабативающие машины и орудия)*. Moscow.

[17] Panov I.M., Tokushev Zh.E., (2005) *Theory, design and calculation of rotary tilling machines (Теория, конструкция и расчет ротоционных почвообрабатывающих машин).* Publ. House of Kokshetau University, Kokshetau, 314 p.

[18] Parkhomenko G.G., (2012), *Tillage in rows of orchards and vineyards*. *Processes, devices*. *(Обработка почвы в рядах садов и виноградников. Процесси, устройства)*. LAP LAMBERT Academic Publishing, ISBN 978-3-659-30811-6, p.148, Saarbrücken, Germany

[19] Raparelli T., Ivanov A., Pepe G. and Eula G. (2021), “Analysis of the energy consumption of a rotary harrow”, *Journal of Agricultural Engineering*, 52(2). https://doi.org/10.4081/jae.2021.1132

[20] Tarverdyan A., Sargsyan S. (2015) The Dynamics of Tiller with Vertical Axis for The Cultivation of Inter-Trunk Spaces in Orchards / *Bulletin of National Agrarian University of Armenia*, Armenian National Agrarian University, Yerevan №4, pp 75-84.

[21] Tarverdyan A.P., Altunyan A.V., Grigoryan A.S. (2024) Force analysis of a rotary tiller with a vertical rotation axis, *INMATEH-Agricultural Engineering* №72(1), p. 454-465, https://doi.org/10.35633/inmateh-72-40

[22] Tarverdyan A.P., Hayrapetyan H.H., (2019), Kinematic and dynamic Study of the rotary crusher transmission gear in the potato digger. *Agriscience and Technology*, Armenian National Agrarian University, Yerevan, №4(68),

[23] Wusong XIAO, Po NIU, Pan WANG, Yingjie XIE, Fei XIA, (2024) Simulation analysis and optimization of soil cutting of rotary blade by ANSYS/LS-DYNA, *INMATEH-Agricultural Engineering*, 72(1): 22-32, Bucharest, Romania https://doi.org/10.35633/inmateh-72-02

[24] Zinoviev, V.A., (1975) *Course on the theory of mechanisms and machines (Курс теории механизмов и машин),* Moscow, “Science”.