

Modeling of resonance phenomena in self-oscillating system of agricultural machines

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Abstract — Factors affecting the interaction of the active cultivator paw with the soil environment are considered in this article. It was established that the active oscillating effect of tillage machines on the environment leads to reduction in its traction characteristics, and the design of symmetrical movable paw is capable of self-oscillation in a horizontal plane. It was developed the theoretical model of oscillatory movements that takes into account the main structural parameters of working bodies represented by symmetrical cultivator paws of the active type. For laboratory studies, a circular soil channel is used, which ensures the movement of the soil relative to the stationary working body for an unlimited time, as well as recording the results using computer, photo and video equipment.

Keywords — active cultivator hoe, cultivation process, draught, plant and ground removing, combined tillage unit.

I. INTRODUCTION

The catastrophic shortage of food should be separately highlighted among the global problems that humanity has faced in recent centuries, which are caused by increase in the world's population.

As known, Ukraine is located in a natural climate zone, which allows it to lead the world's top twenty leaders in the gross harvest of agricultural products in grain and cereal crops production. Corn, wheat and other crops are exported to Europe, Asia and the Middle East.

The demand for food products is growing. Along with the need to increase yields in conditions of energy instability [1], the introduction of energy-saving tillage technologies in crop production is relevant.

Optimal conditions for growing the agricultural plants can be created with sufficient moisture and high-quality cultivation of the soil surface in the area of sowing, which is an important technological process.

Tillage cultivator working bodies that perform these operations are clogged with roots and plant residues and plastic wet soil. This phenomenon is confirmed by the practice of their use and relevant scientific studies [2,3,4].

The accumulation of soil and plant residues on the paws leads to a decrease in the quality indicators of weed cutting

and deterioration in the stability of paw movement in the vertical plane [5,6,7]. This negatively affects the accuracy of the processing depth and the energy indicators of performing this process [8,9].

II. PURPOSE AND OBJECTIVES OF RESEARCH

The scientific research is to develop and justify a model of movement of the cultivator paw in the soil with the reduced specific traction resistance based on the use of resonant shock phenomena in the system "environment – tool blade".

III. MATERIALS AND METHODS OF RESEARCH

Theoretical researches were conducted using methods and techniques of mathematical analysis, vector algebra, differential and integral calculus. Special software Mathcad 15.0 [10] was used at obtaining the final results.

The experimental part of researches was carried out on specially manufactured laboratory equipment, which is based on the developed cultivator paw with the system of elastic and damping elements (fig.1) [3]. When using this construction, resonant shock phenomena become possible in the zone: "paw blades – treated soil". The experimental cultivator paw consists of a hollow riser pipe 1 (fig.1), inside which is a solid shaft 2. In the lower part, the hollow riser pipe and the solid shaft are connected by a self-centered ball bearing 3. In the upper part, the solid shaft is balanced by four springs 9,10,11,12, which are located at an angle of 90° to each other, and are installed on the table 8. The table, in turn, is rigidly fixed to the hollow riser pipe 1 by means of clamps. The springs are placed at the guides 13,14,15,16 and are in contact with the solid shaft by means of spring guides. The spring stiffness is selected in such a way that under the influence of the maximum load on the solid shaft, it can deviate from the vertical position by an angle φ of no more than ± 50 . Potentiometers 17,18 are installed on table 8. By means of levers, the potentiometers are connected to the guide 13 of the spring 11 and the solid shaft 2, correspondingly.

A rectangular opening is made in the hollow riser pipe 1 (fig.1.). Restrictive plate 4 with two holes with a diameter of 16 mm is installed in this opening. The width of the

restrictive plate is less than the width of rectangular opening of the hollow riser pipe. The resulting gap A provides limited movement of the plate 4 in the horizontal plane. Shaft 2 is made step-shaped. The larger diameter of the shaft 2 in its lower part has a milled plane, on which the restrictive plate 4 is attached. The fastening is performed using the bolted connection through the holes of restrictive plate into the corresponding holes with M16 thread. There was also manufactured a restrictive plate 4, the width of which is equal to the width of the rectangular opening of the hollow riser pipe 1 and it corresponds to the complete restriction of its movement in the hollow riser pipe 1.

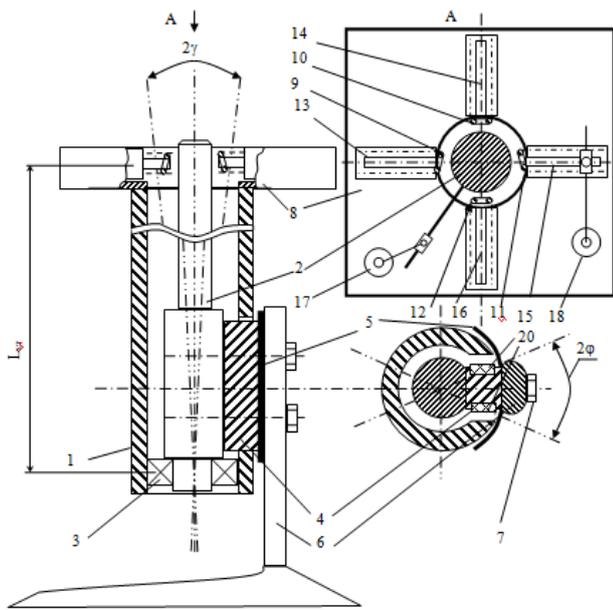


Fig. 1. Design of the working body used in the laboratory unit: 1 – hollow riser pipe; 2 – solid shaft; 3 – self-centered bearing support; 4 – intermediate replaceable plate; 5 – protective plate; 6 – riser pipe with the pointed paw; 7 – bolt M16; 8 – table; 9,10,11,12 – balancing spring elements; 13,14,15,16 – guides of spring elements; 17 – potentiometer for determining the angle of limited rotation; 18 – potentiometer for determining the radial movement of the shaft, 20 – damping plates.

Balancing spring elements 9,10,11,12 (fig.1) are installed on a rigid table 8 in the form of 4 guides 13,14,15,16, which are located at an angle of 90^0 and are in a horizontal plane. The solid shaft 2 is in a vertical position under the action of spring elements 9. The stiffness of the spring elements 9 is chosen in such a way that the possible deviation from the vertical of the solid shaft 2 in the hollow riser pipe 1 corresponds to the maximum force applied to the cultivator paw in the horizontal plane. The value of this force is 600N. The compression of the spring 11 depends on the traction resistance on the paw.

Potentiometers 17 and 18 are installed on the table 8. They are mechanically connected to the guide of the spring element 15 and a solid shaft 2 by means of kinematic chain.

III. RESEARCH RESULTS AND THEIR DISCUSSION

Special requirements for symmetrical working bodies of the cultivator for continuous tillage arise during their application in the machine designs that are used in the technologies of minimum tillage. More stringent requirements are established for the constant depth and uniformity of tillage [2, 3].

Active working bodies can include constructions that provide for the movement of working paws in the soil in different planes under the action of forces that occur on the working planes as a result of interaction of paws with the soil. The expediency of dynamic movement of such working bodies has a double advantage. Firstly, the soils of Europe are full of stones, which makes it necessary to prevent the overloading of working bodies. Secondly, vibration motion is formed, which allows to reduce the tractive effort of the working bodies [11,12]. Such working bodies, under the influence of uneven load and due to elasticity, perform oscillatory movements in the soil, and when their amplitude increases to the critical, they provide the working bodies with the opportunity to go deeper into the soil. We have a positive effect on the one hand, because the riser pipe does not break, and on the other – a negative effect, because the uniformity of the depth of tillage worsens. Such working bodies are widely used in the designs of modern domestic and foreign cultivators, but their common drawback is non-compliance with the uniformity of working depth.

Foreign tillage machines with elastic working bodies include cultivators of the following companies: Nordsten, Lemken, Claas, Pottinger, Reform, Jone Deere, Amazone, Kuhn, Regent, etc.

Working bodies, in which the movement of the riser pipe occurs due to the interaction of elements of working bodies with the soil, have a number of significant disadvantages:

- the uniformity of the depth of tillage worsens;
- require the use of special structural materials that can withstand dynamic loads.

The above-mentioned working bodies for continuous tillage do not fully meet modern agro-technical requirements.

We will also consider the original designs that are offered for movable working bodies of tillage machines.

In order to reduce the tractive effort, width and slope of the back furrow of the cultivator paw with a rigid riser pipe [13], L. F. Babitsky and A. L. Babitsky proposed a tillage working body. In this case, the possibility of movement in a horizontal plane is provided by means of spring-loaded cutting elements. Reduction of the traction force is achieved by using a tillage working body that has a movable vibrating pawed knife on a guide riser pipe with the cutouts in the form of a logarithmic curve, connected in the lower part with the vibration – impact mechanisms of the right and left movable ploughshares using a movable spacer two-lever mechanism, and in the upper part – vibration-impact mechanism connected to the riser pipe. Along with the advantages of this design in its action on the soil, it should be noted its technical complexity.

In order to improve the quality of the working process of pruning weeds, A. F. Volik, A. A. Volik and G. V. Khotin [14, 15] proposed a working body, which creates vibrations consisting of low and high frequency harmonics. Such working body provides a constant depth, but its design is complicated by additional spring mechanisms.

The original design of the working body of tillage equipment [16] was proposed by R. F. Ilyin, V. V. Tunin, L. I. Gasyuk and N. N. Naumenko. In this design, reduction of the traction force and improvement in the quality of technological process for tillage occurs by means of moving the cutting element in the soil due to three springs. This

design of working body has the ability to clean itself from soil and plants, but it is also structurally complex in production and operation.

An example of a working body that periodically rotates around a vertical axis under the influence of soil resistance is the working body of a cultivator [17,18] proposed by A. V. Kalganov, A. M. Saldaev and V. V. Borodichev. This working body, according to the authors, has the ability to increase the period of its use from sharpening to sharpening, and improves the accuracy of the depth of the paw in the soil, but the possible sticking of soil in the zone of axial bearing will inevitably complicate its deepening into the soil.

Reciprocating or rotating directions of movement of cutting surfaces of the working bodies are used for tillage to different depths. At the same time, combinations of the amplitude and frequency of oscillation of the working surfaces can be different. The mode of motion can be oscillatory, when the amplitude of oscillations is relatively large and the frequency is small. The mode can be vibrational when the specified ratios are opposite.

At the Tavrichesky state agro-technological University, S. G. Kartashov and A.V. Elagin proposed an active torsion riser pipe for research. Structurally, the torsion riser pipe consists of two parts. The lower one is a cultivator paw, the upper one is a mechanism for attaching to the cultivator frame. The upper and lower parts are connected by torsion riser pipes. When the cultivator moves, the paw is affected by the heterogeneous composition of the soil and plant roots. In case of meeting the root, for example, with the left wing of the paw, the paw rotates around the vertical axis, and in the torsion riser pipe arises a restorative force. When the sum of the forces acting on the right wing becomes greater than the sum of the forces acting on the left wing, the system turns in a different direction. The use of two torsion bars on the riser pipe allows, according to the authors, to achieve uniform movement of the paw in depth, eliminating the vertical displacement of the working body, that is, the direction of elastic forces will be exclusively in the horizontal plane. The amplitude and frequency of vibrations of such working body depend solely on the physical and mechanical properties of the soil and the physical characteristics of the riser pipe itself. The proposed design of the working body has its own viability, but requires energy and operational researches.

The shown working body has positive characteristics regarding its active effect on the soil, but the interaction of such design of a riser pipe with plants and soil has not been fully studied. The process of cleaning the working body and riser pipe are not sufficiently studied.

In researches [19], it is proposed to compensate the agro-technical drawbacks in the operation of cultivator paws by installing additional rotary needle harrows. Such solution is one of the alternatives on the subject under study, but it does not solve the problem of removing plant residues from the main cultivator paws.

In researches [20], it is proposed to use a rubber damper instead of the known spring-loaded fastening of the racks of cultivator paws in the frame area in order to form movements between the paw and the rack. Such design is appropriate to prevent the overloading in the operation of the cultivator, but it does not fully solve our scientific problem.

In the work [21], the result of research is the development of the working body for cultivator, which primarily solves the issues of environmental friendliness of technologies and the preservation of soil from wind erosion. The issue of self-cleaning of cultivator paws and the issue of reducing the traction resistance in this case are secondary and require further researches.

Based on the mentioned above, in our opinion, it is reasonable to expand the scientific researches in this direction on the design of a new working body in the form of a double-sided pointed paw, which is attached to the frame on a rigid riser pipe and has a spring-damping connection between the paw and the rack. Such working body is active, performs oscillations in the horizontal plane with relatively inconsiderable frequency, performs the main functions of the cultivator paw – pruning weeds and loosening the upper layers of the soil, and it also has a reduced traction resistance compared to the known samples.

To reduce the phenomenon of accumulation of plant residues, it is sufficient to provide the symmetrical cultivator paws with the possibility of slight free rotation in the horizontal plane [2]. It will allow the accumulated plants to slide off the pointed working body at the moment of its deviation in the direction of rectilinear movement. Under the influence of resistance forces of the soil and the difference in torques applied to opposite blades of the paws, the position in the direction of movement of the unit is restored. This algorithm for oscillation of symmetrical paw is confirmed by previous researches.

By studying the rheological heterogeneity of the soil surface in the horizontal plane and taking into account the oscillatory motion, it can be stated that the proposed design of a movable flat-cut paw will reproduce random amplitude-frequency movements. Such character, with certain amplitude-frequency characteristics, will positively affect the overall traction resistance of the working body due to the impact on the soil environment. That is, it becomes relevant to improve the methodology for determining the design parameters of tillage machine with an active effect on the soil.

At the same time, it is reasonable to conduct the analytical confirmation of hypothetical assumptions about reducing the traction resistance of such machines due to the use of resonant and shock phenomena in the system "environment – tool blade".

Let's consider theoretical modeling of parameters that affect the intensity of actions between the soil and the tool blade. In order to make the mathematical model of the system, here is proposed the scheme and representation of points, angles, and force vectors (fig.2).

The symmetrical tillage equipment (as the example is flat-cut paw) we will schematically consider as an equilateral triangle BAC with an angle at vertex A that is equal to 2φ . In practice, this angle is regulated by researches and is in the range of 35-45°. The center of mass (Mc) is a conditional point, which concentrates all the masses affecting the rotational movement of the scheme in the horizontal plane relative to the point (O) of the center of rotation of the body. The distributed load from the ground resistance is conventionally applied to the points (H, E, S). Accordingly, the vectors of resistance forces are formed along the x and y axes in the horizontal plane. The torques of these forces act

in accordance with the kinematic arms of points U, D₁, K. In the process of movement, the equipment in the horizontal plane deviates from rectilinearity by an angle θ . In order to obtain the torque ratios, we consider the triangles OKE, H_UO and OSD₁. Analytical calculations are performed using Mathcad 15.0 [18].

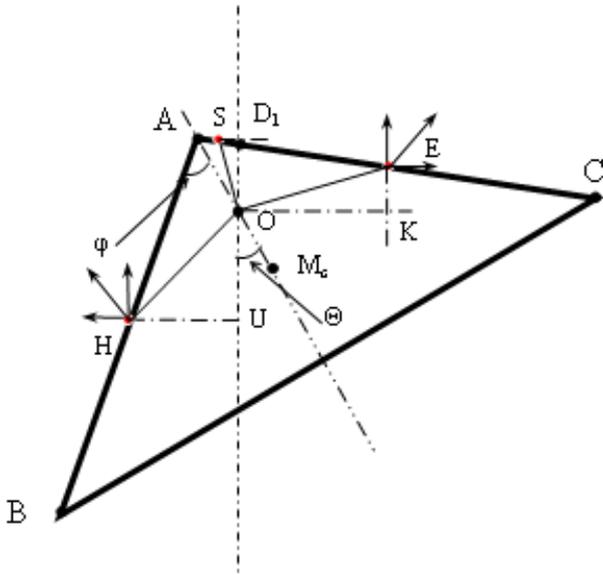


Fig. 2. The scheme of points, angles, and force vectors acting on a symmetrical tillage paw in the horizontal plane.

Accordingly, we have the dependence from the triangle OKE

$$DE = \frac{L_1 - OA \frac{\sin(\theta)}{\cos(\varphi)}}{2} \quad (1)$$

$$OE = \sqrt{OD^2 + DE^2 - OD \cdot DE \cos(\varphi + \theta)}$$

According to the obtained algebraic transformations, the dependences of the vectors OK, EK and OE, the change of which are in accordance with the values of θ , are shown in Figure 3.

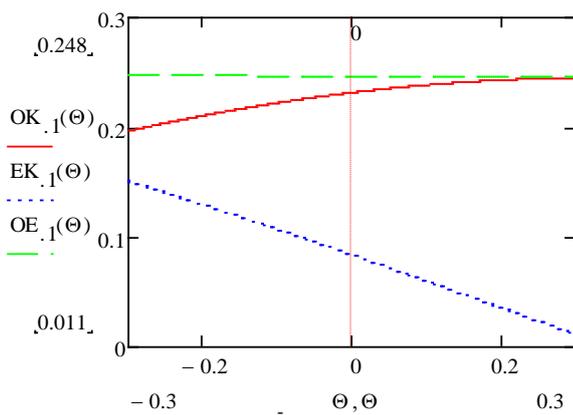


Fig. 3. Analytical study of the changes in vectors of angle functions θ for programming in Mathcad 15.0.

A visual interpretation of the vectors of angle functions θ indicates a certain symmetry relatively to the zero position, which corresponds to the rectilinear motion of the symmetric body. It should also be noted that the range of changes in the angle θ varies within $\pm 30^\circ$ ($\pm 0,5\text{rad}$).

From triangle OSD, using a similar method, we obtain dependence of OD and SD vectors (fig.4).

The analytical values of OD and SD vectors, which are functions of the angle θ , are also symmetric relatively to the position at $\theta=0$.

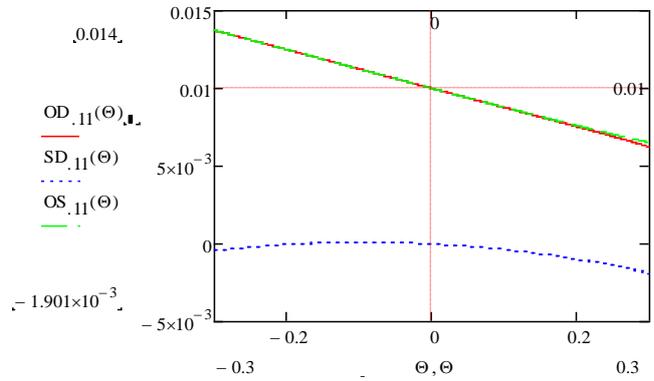


Fig. 4. Analytical study of changes in the vectors OD and SD of the angle functions θ .

$$SD_{11}(\theta) := \left[\sqrt{OA^2 + \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right)^2} - 2 \cdot OA \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi) \right] \cdot \sin \left[\theta - a \sin \left[\frac{\left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi)}{\sqrt{OA^2 + \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right)^2} - 2OA \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi)} \right] \right] \quad (2)$$

$$OD_{11}(\theta) := \left[\sqrt{OA^2 + \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right)^2} - 2 \cdot OA \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi) \right] \cdot \cos \left[\theta - a \sin \left[\frac{\left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi)}{\sqrt{OA^2 + \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right)^2} - 2OA \left(\frac{OA \cdot \sin(\theta)}{2 \cos(\varphi)}\right) \sin(\varphi)} \right] \right] \quad (3)$$

Accordingly, from triangle AOH

$$OU_1(\theta) := \left[\cos \left[2\pi - \theta - a \sin \left[\frac{\frac{L_1}{2} \sin(-\varphi)}{\left(\frac{L_1}{2}\right)^2 + OA^2 - L_1 \cdot OA \cdot \cos(-\varphi)} \right] \right] \right] \quad (4)$$

$$HU_1(\theta) := \left[\sin \left[2\pi - \theta - a \sin \left[\frac{\frac{L_1}{2} \sin(-\varphi)}{\left(\frac{L_1}{2}\right)^2 + OA^2 - L_1 \cdot OA \cdot \cos(-\varphi)} \right] \right] \right] \quad (5)$$

Accordingly, the analytical image of these vectors is shown in Figure 5.

Determination of the torques of forces acting on the tool

blades depending on the angle Θ (fig. 6) will have the form.

$$M_x = OK \left(DE + \frac{L_1}{2} \right) \cdot k - SD_1 \cdot AD \cdot k - HU \cdot k \cdot L_1 \quad (6)$$

$$M_y = EK \cdot k \cdot \left(DE + \frac{L_1}{2} \right) - AS \cdot k \cdot OD_1 - OU \cdot k \cdot L_1 \quad (7)$$

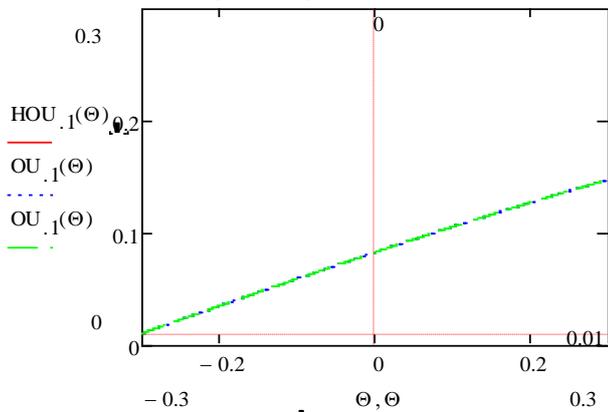


Fig. 5. Analytical study of changes in vectors OU and HU of the angle functions Θ .

Where the total torque is obtained from the vector product.

$$M_o = \sqrt{(M_x)^2 + (M_y)^2} \quad (8)$$

The obtained dependence of the change in torque in the x and y planes functionally depends on the angle Θ of deviation from rectilinearity of movement of the blade of the flat-cut paw. Applying the torques of acting forces, we will have a differential equation taking into account the gradual and oscillatory movement performed by the paw during complex movement in the soil. This model considers only the horizontal plane.

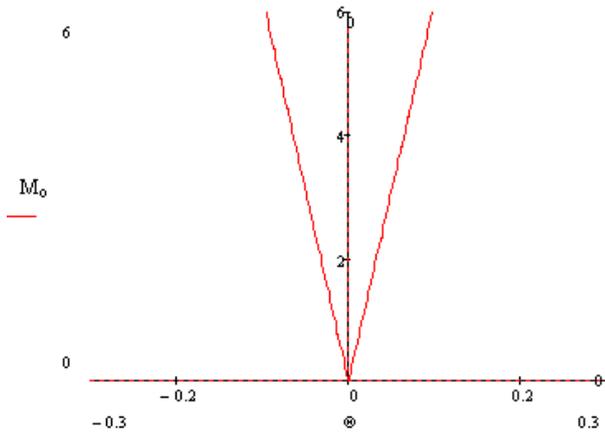


Fig. 6. Analytical study of the change in torque that occurs on a symmetrical working blade.

The initial values are given relatively to the existing cultivator paw with the corresponding design characteristics of the cutting paw of the cultivator SSC-4 with the following indicators: weight – 3.6 kg, angle at the vertex of the blade – 60° , blade length – 0.3 m (fig.7).

The obtained solution of the differential equation describing the oscillatory motion of a symmetric working blade in the form of a functional dependence of $y(t)$ coordinate on the observation time t shows that within the

accepted initial conditions, a synchronous oscillation occurs with an amplitude of 0.2 radians (11.5 deg) and the step of oscillations – 9c. This result is compliant with experimental data.

To confirm the validity of obtained theoretical dependencies, it is necessary to compare them with the experimental ones. An experimental laboratory unit based on a new working body (fig. 8) was developed and manufactured in order to determine the energy indicators of operation of the tillage working body. Typical podzolized chernozem is used as the soil medium.

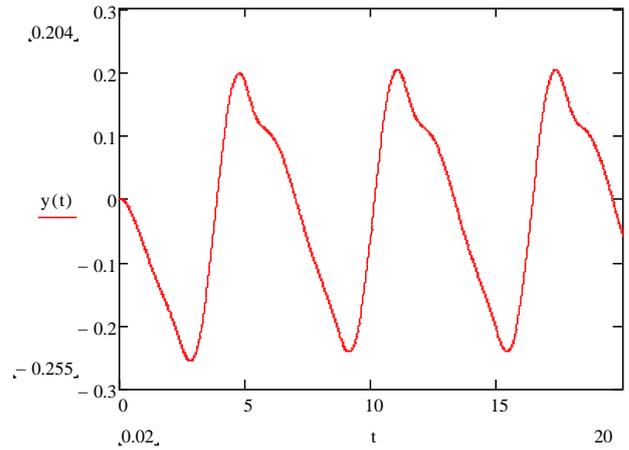


Fig. 7. Graphical representation of the solution of a differential equation in Mathcad 15.0

For laboratory tests, a circular soil channel is used, which ensures the movement of the soil relatively to the stationary working body for an unlimited time. The soil channel has the ability to change the speed of rotation of the tray during research.

The soil rotates, but the models of working bodies and measuring equipment remain stationary. The model of the working body 4 is fixed motionlessly to the riser pipe 5, which regulates the depth of immersion of the working body. The riser pipe by means of carriage is moved radially along the holder, on which the support frame holds two or more working bodies. The unit has support frame 6, working bodies with side mounting of racks, and the depth of immersion of working bodies in the soil is regulated by the mechanism of side riser pipes within the range from 0 to 50 cm. The channel receives rotational motion from the drive shaft, to which mechanical variable-speed drive transmits rotational motion from the gear motor 7. The mechanical variable-speed drive, during operation, smoothly changes the speed of rotation of the channel within the range from 0.01 to 333 m/min.

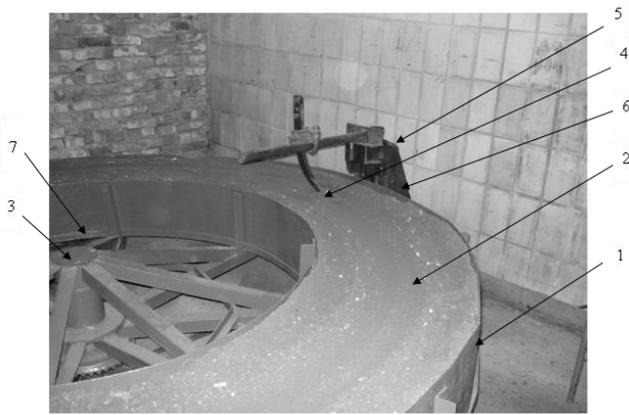


Fig. 8. Circular soil channel: 1 – tray; 2 – soil; 3 – rotation axis; 4 – tillage working body; 5 – riser pipe; 6 – support frame; 7- gear motor

Conditions for conducting the experiment, physical and mechanical properties of the soil were determined at the beginning of research. The working body is immersed in the soil environment to the appropriate depth in accordance with the experimental program. The movement of the circular soil channel is activated. The traction resistance of experimental working body was determined at the first stage using dynamometric equipment. On the second stage, the maximum torque of such a working body was determined with the help of dynamometric equipment.

By substituting the experimentally obtained values of the physical and mechanical characteristics of the soil to the theoretical model, we obtain the corresponding theoretical data of torque that confirm the similarity with the results of the experiment. The results of the research are presented in table 1.

TABLE I. RESULTS OF DETERMINING THE TORQUE MKP THEORETICALLY AND EXPERIMENTALLY

Angle of maximum deviation, φ_0	Depth of treatment, cm	Torque, Nm		Error, %
		Experimental	Theoretical	
±15	5	5,27	5,0	5,1
	10	7,51	8,0	6,5
	15	10,8	11,4	5,5
±30	5	8,1	7,3	9,8
	10	12	10,9	9,2
	15	14,1	14,6	3,6

The error between the experimental and theoretical values of the torque acting on the cultivator paw at the moment of its maximum deviation is within 10%. It is acceptable for the statement that the theoretical model of the experimental working body with the possibility of limited rotation in the horizontal plane and the results of laboratory experiments are similar [18].

IV. CONCLUSIONS

1. It is determined that the system "environment – tool blade" remains relevant from the point of view of energy components and economic factors in agriculture both in Ukraine and abroad.

2. The considered existing designs of working bodies that use oscillatory movements do not fully meet modern agro-technical requirements, so there is a possibility of their further improvement.

3. The developed cultivator paw with the system of elastic and damping elements is proposed for theoretical and experimental studies. When using this design, resonant shock phenomena become possible in the zone: "paw blades – treated soil".

4. The theoretical model that describes the regularities of gradual and oscillatory movement of the cultivator paw during its complex movement in the soil, depending on the observation time, was developed in this article.

5. The adequacy of the theoretical model was checked by comparing the results, theoretical calculations and laboratory tests of experimental cultivator working body with the weight of 3.6 kg, angle at vertex of 60° and the blade length of 0.3 m. The adequacy of the theoretical model is 90%.

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