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Development of the theory of root crop head cleaner movement in the longitudinal-vertical plane mounted behind a wheeled tractor

Abstract. The relevance of developing analytical methods for studying the stability of root crop cleaners on tractors is determined by the need to improve the efficiency and quality of root crop head cleaning, as this affects the quality of the final product and its marketable properties. The research aims to increase the stability of the cleaner movement by constructing a mathematical model of its oscillatory motion in the longitudinal-vertical plane and its numerical solution to determine the influence of parameters in response to external disturbances. To achieve this goal, a new theory of the movement of a root crop head cleaner in the longitudinal-vertical plane, mounted behind a wheeled tractor, under the influence of the disturbing effect of irregularities in the longitudinal profile of the soil surface on the amplitude and phase frequency characteristics of its angular oscillations was developed. Using numerical calculations carried out on a personal computer, the conditions were found under which the stability of the cleaner's movement in the specified plane will increase when the stiffness coefficient of the pneumatic tyres of the doubler wheels is 315 kN·m^{-1}.· This result is achieved when the pressure in the pneumatic wheel tyres is 135 kPa. As for the damping characteristics of the doubler wheel tyres, which are determined by the coefficient μ, it was found that when it changes from 350 N·s·m^{-1} to 1350 N·s·m^{-1}, there is an invariance of the delay in the reaction of the cleaner to the disturbing effect when its frequencies change from 0 to 24 s^{-1}. It was also found that the influence of the geometric dimensions of the cleaner is insignificant in the range of frequencies of oscillations of the ordinates of the longitudinal profile of soil surface irregularities from 0 to 24 s^{-1}. This follows from the nature of changes in the obtained amplitude-frequency and phase-frequency characteristics. The methodology for constructing a mathematical model of the plane-parallel oscillatory motion of the cleaner can be used in similar analytical studies of other agricultural machines mounted on a wheeled aggregating tractor

Keywords: ground irregularities; equivalent circuit; oscillations; differential equations; amplitude-frequency; phase-frequency characteristics

INTRODUCTION

In the cultivation of most root crops, the most common technology is separate harvesting, which involves cleaning the heads from the remains of the tops on the root. Most heads contain both short green and strong petioles (uncut parts of the green stalks of the tops), which are located on the top, and dry and lying residues in the form of threads of a certain length, which are firmly connected to the side parts of the heads and are located between the rows of crops or between the roots in the row itself, their high-quality removal is a difficult technical task. The conditions for a successful solution to this problem include not damaging the spherical surfaces of the heads and not knocking the root bodies out of the soil. It is possible to guarantee the separation of such residues by applying forces to the surface of each head to separate these types of residues.

The use of rubber cleaning pairs of blades, which cover the row of root crops on both sides, allows you to successfully perform this task. However, when the operating speeds of the cleaners increase, certain difficulties arise, one of the main ones being the oscillations that necessar-

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ily occur for the cleaning tines in the longitudinal-vertical plane. These oscillations cause the ends of the rubber blades to lose contact with the spherical surfaces of the root crop heads, which can lead to a deterioration in the quality of the cleaning process.

According to V. Bulgakov *et al.* (2023), these oscillations are caused by uneven soil surfaces, which in itself is an external disturbance for the cleaner if it is considered a complex dynamic system. Furthermore, the source of such oscillations can be the pneumatic doubler wheels that the cleaners are equipped with.

R. Hevko *et al.* (2014; 2018) and M. Budzanivskyi (2022) studied bunchers, cleaners, and root harvesters under modelling conditions that involve the use of significantly simplified equivalent circuits. This, in turn, leads to the use of initial equations based on the basic law of dynamics. This does not provide the appropriate accuracy of the built mathematical models. The issues of potato tops removal are discussed by H. Kainuma *et al.* (2013). Theoretical studies are given only conditionally and do not reflect the actual technological process of potato tops removal. Some results of solving this problem are presented by F. Wang (2021) and K.P. Titiwa *et al.* (2019). Attempts to solve this problem were made by Ye. Ihnatiev (2017). However, when building the computational mathematical model, an extra generalised coordinate was introduced here, which distorted the accuracy of the final result. These studies do not contain a modern representation of oscillatory processes carried out by complex dynamic systems. Namely, they do not contain either amplitude-frequency or phase-frequency characteristics, which in no way confirms the approximation of the modelled process to the real process.

Analytical studies of the oscillatory movements of agricultural machines rear-mounted on tractor units equipped with support wheels with pneumatic tyres were investigated by V. Nadykto *et al.* (2020). The theoretical models were based on a method that is based on the representation of support and doubler wheels that have elastic-damping properties characterised by stiffness and damping coefficients. In general, these studies do not consider the exact configuration of the oscillating system under study. Namely, the closest location to the tractor's rear hitch mechanism of the machine is aggregated doubler wheels, and behind them are the cleaning working bodies that form the working loads. In general, the point of contact of the doubler wheel with the soil can be the point around which this oscillatory system can rotate.

Although a detailed consideration of solving similar problems has been carried out in the study of car oscillations (Deng *et al.*, 2016; Friedl & Mangerig, 2017) and wheeled tractors (Zheng *et al.*, 2019; Yoo *et al.*, 2021), the study of oscillatory movements of root crop head cleaners requires new approaches related primarily to the specifics of the operating conditions associated with the application of combing forces directed downwards (to the row of root crops located on the soil surface) by external forces, that have different physical characteristics and directions, which will largely determine the nature of these oscillations, as well as the configuration of the location of the suspension points to the aggregating wheeled tractor, the installation of the doubler wheels, and the places where the working forces are formed. Therefore, the research aims to determine the conditions of stable movement of the cleaner based on the constructed mathematical model of its oscillatory movement in the longitudinal-vertical plane and the numerical solution of this model to determine the influence of parameters in response to an external disturbance.

MATERIALS AND METHODS

In 2021, at the National University of Life and Environmental Sciences of Ukraine, the Department of Mechanics of the Faculty of Construction and Design, under the leadership of Prof. V. Bulgakov, developed a new root crop head cleaner, in which each row of root crops was cleaned of the residues of tops on the heads at once using two cleaning shafts, on which pairs of rubber blades are hinged on hubs.

The design feature of the developed cleaner, which is shown in Figure 1, is that it is mounted on the rear hitch 2 of the wheeled aggregating tractor 1, has two drive cleaning shafts 6 mounted on the frame 3 and is located horizontally at an acute angle to each other in the transverse plane, covering the row with root crops from both sides. The rows of rubber cleaning blades 7 are fixed on them with the help of clips 8 with an appropriate spacing. Drive shafts 6 have counter-rotating movements provided by the drive 4 (Fig. 1).

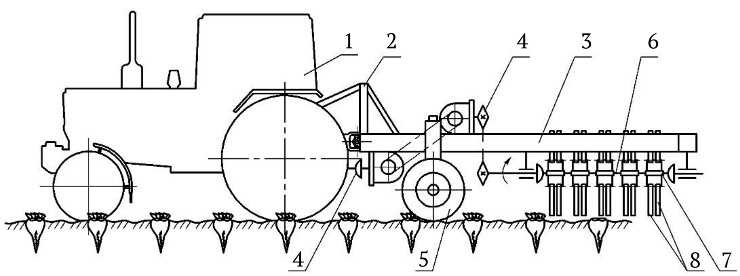

Figure 1. Design and technological scheme of a twin-shaft root crop head cleaner for removing tops from the root
Note: 1 – wheeled aggregating tractor; 2 – rear tractor hitch; 3 – frame; 4 – elements of the cleaning shaft drive; 5 – doubler wheels; 6 – cleaning shafts; 7 – rubber cleaning blades; 8 – blade clips
Source: compiled by the author

The installation of the drive cleaning shafts 6 in the horizontal plane at an angle creates a so-called capture zone for all root crop heads in case they deviate from the row axis. This ensures that they are guaranteed to fall into the cleaning channel formed by the drive cleaning shafts 6 and that they are in contact with the ends of the rubber blades 7.

At the same time, the efficiency of cleaning the heads of root crops from the remains of tops on the root depends on the height of the drive cleaning shafts 6 above the soil surface. For setting and adjusting this parameter, frame 3 of the root crop cleaner is equipped with two doubler wheels 5 located on both sides of frame 3, which have pneumatic tyres.

At the same time, the doubler wheels 5 can be installed in different places on the side parts of frame 3, thanks to the screw mechanisms for their movement and fixation on the specified side parts of frame 3. The doubler wheels 5 move in the rows of root crops, copying the unevenness of the soil surface. This causes oscillations of frame 3 and cleaning shafts 6 in the longitudinal-vertical plane, which can significantly reduce the quality of cleaning the heads of root crops, especially at a sufficiently high speed of the forward movement of the cleaner. In addition, the copying wheels 5 are made in the form of pneumatic tyres, which, with a significant mass of the cleaner, causes oscillatory movements of its frame 3 in the longitudinal-vertical plane. To develop the theory of the movement of the cleaner for cleaning root crop heads from the remains of tops on the root in the longitudinal-vertical plane, considering that it is hitched to the rear of a wheeled aggregating tractor, it is necessary to develop a mathematical model, the first stage of which was to draw up its equivalent scheme.

On this basis, it is necessary to consider the cleaner mounted behind the tractor as a dynamic system consisting of specific elements and moving under the influence of external forces. In this case, as a first approximation, it can be assumed that the aggregating wheeled tractor with which the cleaner is connected moves in a straight line and uniformly (i.e., moves forward at a constant speed $\bar{V}$) and its influence on the movement of the cleaner itself is minimal. It is also important that the movement of the cleaner as a dynamic system must be considered in one plane, i.e., in this case, it is the longitudinal-vertical plane.

The main assumptions in the development of this theory, and, accordingly, the equivalent diagram, are that, based on the specified conditions, the rear-mounted root crop header is an oscillating system that can be represented as independent of the aggregating tractor, which, in the course of its operation, oscillates in the specified plane relative to the elements of the tractor's rear hitch. In terms of construction, the cleaner consists of a frame, two doubler wheels and cleaning tines, and therefore all these elements should be shown in an equivalent diagram.

Given the above, the cleaner was presented in the form of a frame *BE*, which, with the help of the two lower *AB* and one upper *TD* joints is attached with its hitch *BD* to the rear hitch of the tractor. Points *A* and *T* belong to the tractor and as shown above, move in a straight line and uniformly in the direction shown by the arrow with a constant forward speed $\bar{V}$. Cleaner's frame *BE* relies on two copying pneumatic wheels, which, in the case of considering this dynamic system in the longitudinal-vertical plane, can be shown in the equivalent diagram as one common wheel with the centre marked by the point O_1. Point *U* Fixing the doubler wheel brackets to the frame *BE* can be located anywhere on the machine. The cleaning parts (horizontal cleaning shafts with rubber blades) can be mounted on the frame *BE*, are located mainly in the rear part of the machine and are represented in the equivalent diagram as one common working body. Point *E* may be closer to point *U*. These elements of the cleaner as a dynamic system are shown in Figure 2.

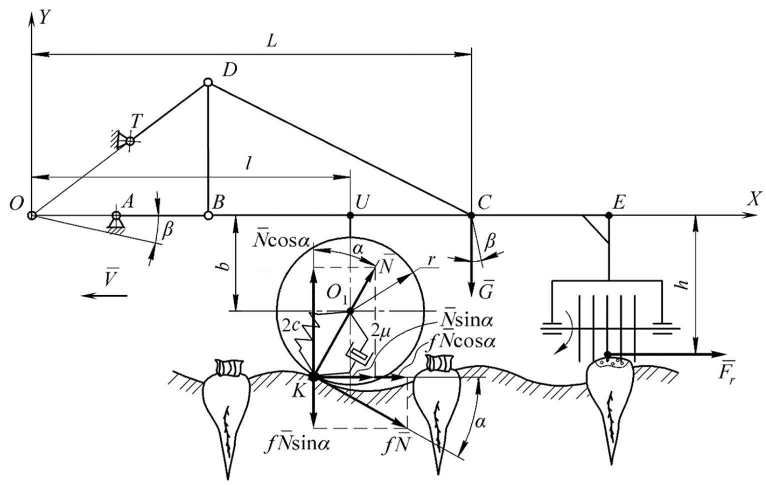

Figure 2. Equivalent scheme of the root crop head cleaner

Note: β – angle of rotation of the cleaner in the longitudinal-vertical plane (generalised coordinate); c – stiffness factor of the pneumatic tyres of the doubler wheels; μ – damping coefficient of the pneumatic tyres of the doubler wheels; N – normal reaction at the point of contact between the doubler wheel and the uneven ground surface; F_r – the force of resistance to the movement of the cleaning working body on the heads of root crops; f – rolling resistance coefficient of the doubler wheel; α – angle of inclination to the horizon of the unevenness of the soil surface passing through the contact point

Source: compiled by the author

All external forces acting on this dynamic system in the process of its movement were identified. If the centre of gravity of the cleaner is *C*, the force of gravity is applied to it $\bar{G}$, which is directed vertically downwards. The cleaning elements in the form of two horizontal shafts form the total force of the working resistance, which is determined by $\bar{F}_r$. In the first approximation, it was assumed that this force is parallel to the frame *BE* and is directed backwards. Furthermore, assuming that the two pneumatic doubler wheels move between the rows of root crops over uneven soil surfaces, the unevenness itself can be approximated by a harmonic function. An equivalent wheel relies on this inequality at the contact point *K* (the equivalent diagram shows that this is a point contact and occurs at any position on the harmonic curve itself). Then at this point of contact, *K* two forces will be applied, one of which is the normal reaction $\bar{N}$ the ground on the doubler wheel, which passes through the centre of the wheel, i.e., the point O_1 and tangent $f\bar{N}$ of soil reaction. Shown reactions $\bar{N}$ and $f\bar{N}$ are mutually perpendicular, and the tangential reaction $f\bar{N}$ is a frictional force equal to the product of the normal reaction $\bar{N}$ and friction coefficient *f*.

Furthermore, since the cleaner's doubler wheels (one equivalent wheel in Fig. 2) are pneumatic, they can be represented as elastic-damping models. Specifically, they should have the appropriate elasticity coefficients *c* and damping coefficients *μ*. Since Figure 2 shows a single equivalent wheel, the coefficients should be double.

The rear hitch of the implement, which has two lower *AB* and central *TD* rods in the operating (floating) position allows free angular rotation of the frame *BE* cleaner, but it occurs around point *O*, a conditional point, located at the intersection of the lines drawn through the specified lower and upper links of the tractor hitch (i.e. lines that are an extension of the specified links).

The next step in constructing an equivalent diagram was to select and designate a coordinate system. Considering that this theory considers the movement of the cleaner in only one plane (longitudinal-vertical), a flat Cartesian coordinate system was chosen. If using point *O* as a coordinate centre, the following coordinate plane will be true: *XOY*. Then, axis *OX* is horizontal and passes through the frame *BE* of the cleaner backwards, and the axis *OY* is directed vertically upwards.

The linear and angular dimensions were then marked on an equivalent diagram. First of all, if, as mentioned above, the cleaner can make free angular turns relative to the reference point *O*, then this point can be the axis of the instantaneous centre of rotation of the central *TD* and lower *AB* The linkage of the rear linkage of the implement. If the angle is presented as *β*, by which the frame *BE* deflects, then it will characterise the oscillatory movements of the cleaner as a dynamic system under consideration relative to the centre *O*. Next, the distance from the point *O* to centre *C* The weight of the cleaner is indicated by *L*. And the corresponding distance from the point *O* to the vertical axis of the doubler wheel (i.e., to the point *U*) is determined as *l*. The radius of the equivalent doubler wheel is denoted by *r*, The height of the centre of the doubler wheel relative to the frame *BE* (thus distance UO_1) to *b*, while the distance from frame *BE* to the ends of the rubber blades of the cleaner (where the total force of working resistance is applied) through *h*. Tangential reaction $f\bar{N}$ deviated from the horizontal by an angle *α*.

Given the accepted designations and dimensions of a normal $\bar{N}$ and tangential $f\bar{N}$ reactions of the soil to the doubler wheel applied at the point *K* can be immediately presented in the form of vertical and horizontal components. The next stage in the development of the theory of the cleaner's motion in the longitudinal-vertical plane was the selection of the dynamics law, which will be used to describe the motion. The most suitable for this purpose is the initial general equation in the form of Lagrange of the second kind of the following form:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\beta}}\right)-\frac{\partial T}{\partial \beta}=Q_{\beta}-\frac{\partial P}{\partial \beta}-\frac{\partial R}{\partial \dot{\beta}}, \tag{1}$$

where *T* – kinetic energy of the system; Q_β – generalised force; *P* – potential energy of the system; *R* – dissipative function; β – generalised coordinate; $\dot{\beta}$ – generalised velocity.

Using the original equation (1), it will be possible to compile a differential equation of motion of the root crop head cleaner in the longitudinal-vertical plane. Since, under the influence of external forces acting on the elements of this dynamic system, it can perform only angular oscillatory movements relative to the point *O*, one generalised coordinate, i.e., the angle of rotation *β*, is true. Thus, the theory of motion of the cleaner in the longitudinal-vertical plane will consist of a single differential equation of its motion in this plane.

The next step is to determine the conditions of the doubler wheel movement when considering the angular oscillatory movements of the cleaner. In general, each of the two tracing wheels of the cleaner moves in the row spacing of root crops, where, at the time of harvesting, the soil has sufficient density and hardness. However, due to the weight of the entire cleaner, its doubler wheels slightly deform the surface layers of the soil and move over uneven surfaces that can be identified by a harmonic function, such as a cosine function. Then the current ordinate *y* of this harmonic function can be an equation of the following form:

$$y=y_0\left(1-\cos\frac{2\pi\cdot x}{l_1}\right)=y_0\left(1-\cos\frac{2\pi\cdot V\cdot t}{l_1}\right), \tag{2}$$

where y_o – amplitude of inequality fluctuations; l_1 – period of fluctuations in soil surface roughness; *x* – current coordinate of the bump.

In this case, the current coordinate x unevenness of the soil surface is defined as:

$$x=V\cdot t, \tag{3}$$

where *V* – translational speed; *t* – current time. Subsequently, all the components included in the general initial equation (1) were determined for the considered dynamic system.

Following the equivalent scheme (Fig. 2) kinetic energy T of this dynamic system should be determined under the condition of rotational movement of the cleaner around the point O. Therefore, given that the angle of rotation of the cleaner is marked β, the expression for kinetic energy T was presented as follows:

$$T = \frac{I_{oz} \cdot \dot{\beta}^2}{2}, \tag{4}$$

where I_{oz} – moment of inertia of the cleaner relative to the axis passing through the point O, perpendicular to the longitudinal-vertical plane.

The potential energy P of the dynamic system under consideration is determined by the work of elastic forces of the doubler wheels, which in this case are pneumatic. As is well known, this elasticity force work is a function of the deflections of the pneumatic wheel tyres. These deflections can be analytically determined as follows. Considering the smallness of the angle α These deflections can be represented as the difference between the amplitudes of vertical vibrations y_1 the centres of the doubler wheels, i.e., the point O_1 and the longitudinal profile of the soil surface itself, which is determined by the current ordinate y. And the determined difference $(y_1 - y)$ identifies deflections of the pneumatic wheel.

Considering the above expression for determining the potential energy P of the oscillatory motions of this dynamic system was as follows:

$$P = \frac{2 \cdot c \cdot (y_1 - y)^2}{2} = c \cdot (y_1 - y)^2. \tag{5}$$

Next, the components of expression (5) were identified. As such, amplitude y_1 of vertical oscillations of the centre of the doubler wheel (point O_1) related to the rotation angle β as follows:

$$y_1 = l \cdot \beta, \tag{6}$$

where l – constructional dimensions are shown in the equivalent diagram.

Then, considering (6), expression (5) takes the following form:

$$P = c \cdot (l \cdot \beta - y)^2 = c \cdot (l^2 \cdot \beta^2 - 2 \cdot l \cdot \beta \cdot y + y^2). \tag{7}$$

To determine the dissipative function R the following expression was used:

$$R = \frac{2\mu \cdot (\dot{y}_1 - \dot{y})^2}{2} = \mu \cdot (\dot{y}_1 - \dot{y})^2. \tag{8}$$

Given that the first derivative of expression (6) is equal to $\dot{y}_1 = l \cdot \dot{\beta}$, then the dissipative function R will have the final form:

$$R = \mu \cdot (l^2 \cdot \dot{\beta}^2 - 2 \cdot l \cdot \dot{\beta} \cdot \dot{y}_1 + \dot{y}^2). \tag{9}$$

Thus, all the values on the left side of equation (1) and most of the values on the right side have been determined. Based on this, it was possible to transform them according to the requirements of the general equation (1). So partial derivatives, as well as derivatives from the results obtained in time t, that are related to kinetic energy T, were as follows:

$$\frac{\partial T}{\partial \beta} = 0, \tag{10}$$

and

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\beta}}\right) = I_{oz} \cdot \ddot{\beta}. \tag{11}$$

Next, similar transformations were performed for the potential energy P by the requirements of expression (1). The partial derivative of the function (7) by the parameter β (i.e., the generalised coordinate) was as follows:

$$\frac{\partial P}{\partial \beta} = 2c \cdot l^2 \cdot \beta - 2c \cdot l \cdot \beta \cdot y. \tag{12}$$

Following the conditions stipulated by expression (1), the partial derivative of the dissipative function R by parameter $\dot{\beta}$ (i.e., the generalised speed) will be determined as follows:

$$\frac{\partial R}{\partial \dot{\beta}} = 2\mu \cdot l^2 \cdot \dot{\beta} - 2\mu \cdot l \cdot \dot{y}. \tag{13}$$

The next step was to find an expression that would define the generalised force Q_β at the generalised coordinate β. For this purpose, an expression was first drawn up, according to which the basic operation was defined δA_β of all active forces on the possible displacement of a given dynamic system. In this case, these possible displacements will be on the possible angular displacement $\delta\beta$. First, a basic operation δA_β of active forces on the possible movement $\delta\beta$. Using the equivalent scheme (Fig. 2), it is possible to calculate:

$$\begin{aligned}\delta A_\beta = G \cdot L \cdot \delta\beta - N \cdot \cos\alpha \cdot (l - r \cdot \sin\alpha) \cdot \delta\beta - N \cdot \\ \cdot \sin\alpha \cdot (b + r \cdot \cos\alpha) \cdot \delta\beta - f \cdot N \cdot \cos\alpha (b + r \cdot \cos\alpha) \cdot \delta\beta + f \cdot N \cdot \\ \cdot \sin\alpha \cdot (l - r \cdot \sin\alpha) \cdot \delta\beta - F_r \cdot h \cdot \delta\beta.\end{aligned} \tag{14}$$

Since:

$$\delta A_\beta = Q_\beta \cdot \delta\beta, \tag{15}$$

where Q_β – generalised force by angular coordinate β, then based on (14), the expression of the generalised force Q_β is as follows:

$$\begin{aligned}Q_\beta = \frac{\delta A_\beta}{\delta\beta} = G \cdot L - N \cdot \cos\alpha \cdot (l - r \cdot \sin\alpha) - N \cdot \sin\alpha \cdot \\ \cdot (b + r \cdot \cos\alpha) - f \cdot N \cdot \cos\alpha (b + r \cdot \cos\alpha) + \\ + f \cdot N \cdot \sin\alpha \cdot (l - r \cdot \sin\alpha) - F_r \cdot h,\end{aligned} \tag{16}$$

where L, r, b, h and α – the corresponding linear and angular dimensions, which are indicated on the equivalent diagram (Fig. 2). Expression (16) shows that the generalised force Q_β is the algebraic sum of the moments of all active forces acting on a given dynamic system. Assuming that the angle α is small enough, the following is true: $\cos\alpha \approx 1$, and $\sin\alpha \approx \alpha$. The expression (16) may be presented in a simplified form:

$$\begin{aligned}Q_\beta = G \cdot L - N \cdot (l - r \cdot \alpha) - N \cdot \alpha \cdot (b + r) - f \cdot N \cdot (b + r) + \\ + f \cdot N \cdot \alpha (l - r \cdot \alpha) - F_r \cdot h.\end{aligned} \tag{17}$$

After several transformations, the following was obtained:

$$Q_\beta = G \cdot L - N \cdot [-f \cdot r \cdot \alpha^2 - (b - f \cdot l) \cdot \\ \cdot \alpha - f \cdot b - f \cdot r - l] - F_r \cdot h. \quad (18)$$

Assuming that the doubler wheel is close enough to the frame, and therefore $b \approx r$, the final generalised power of the Q_β will be as follows:

$$Q_\beta = G \cdot L - N \cdot [-f \cdot r \cdot \alpha^2 - (b - f \cdot l) \cdot \\ \cdot \alpha - 2f \cdot r - l] - F_r \cdot h. \quad (19)$$

Next, it was necessary to determine the angle α, which is included in expressions (14)-(19). The angle α is related to the function y, i.e., with the function of vertical ordinates of a cosine function that reflects the longitudinal profile of the soil surface roughness given by expression (2). After differentiating expression (2) by the coordinate x the following expression is obtained, the value of which is equal to tan α:

$$\frac{dy}{dx} = \frac{2\pi \cdot h_0}{l_1} \cdot \sin\frac{2\pi \cdot V \cdot t}{l_1} = \tan\alpha. \quad (20)$$

From expression (20), the value of the desired angle α is found:

$$\alpha = \arctan\left(\frac{2\pi \cdot h_0}{l_1} \cdot \sin\frac{2\pi \cdot V \cdot t}{l_1}\right). \quad (21)$$

Finally, based on the use of the obtained expressions (10)-(13) and (21), their substitution in the original equation (1) and the necessary general transformations, the following differential equation of movement of the root crop head cleaner in the longitudinal-vertical plane was obtained:

$$a_2 \cdot \ddot{\beta} + a_1 \cdot \dot{\beta} + a_o \cdot \beta = \\ = b_1 \cdot \dot{y} + b_o \cdot y + C, \quad (22)$$

where $a_2 = I_{oz}$; $a_1 = 2\mu \cdot l^2$; $a_o = 2c \cdot l^2$; $b_1 = 2\mu \cdot l$; $b_o = 2c \cdot l$; $C = G \cdot L + N \cdot [f \cdot r \cdot a^2 - (b + f \cdot l) \cdot \alpha + 2r \cdot f - l] - F_r \cdot h$.

Equation (22) is a second-degree linear differential equation and is a mathematical model of the movement of a root crop head cleaner in the longitudinal-vertical plane mounted behind a wheeled aggregating tractor. The output parameter of this dynamic system is the angle , i.e., the angle of rotation of the cleaner in the longitudinal-vertical plane.

The next step was to solve the differential equation (22). For this purpose, the most suitable method is the transformation of the differential equation (22) into an algebraic equation. In this case, the method of operational calculus, in particular the Laplace transform method, was applied. According to this method, an operator of the following form was introduced (Bulgakov, 2023):

$$s = \frac{d}{dt}. \quad (23)$$

Using the operator (23) and transforming the differential equation (22), an algebraic equation is obtained, which has the following form:

$$(a_2 \cdot s^2 + a_1 \cdot s + a_o) \cdot \beta(s) = (b_1 \cdot s + b_o) \cdot y(s) + C \cdot 1(s). \quad (24)$$

Analysing expression (24), its components reflect the input external disturbances $y(s)$, in the form of oscillations of the ordinates y profile of soil surface irregularities, which are transferred to the output parameter $\beta(s)$, and also reflect discrete changes in the values of the parameters included in the coefficient C on the same output parameter $\beta(s)$.

This circumstance allowed us to create two transfer functions $W_1(s)$ and $W_2(s)$, which reflect the nature of the oscillatory movements of the cleaner in the longitudinal-vertical plane. The values of these transfer functions are respectively equal:

$$W_1(s) = \frac{b_1 \cdot s + b_o}{a_2 \cdot s^2 + a_1 \cdot s + a_o}, \quad (25)$$

and

$$W_2(s) = \frac{C}{a_2 \cdot s^2 + a_1 \cdot s + a_o}. \quad (26)$$

It should be noted that, due to the linearity of the algebraic equation (24), any parameter determined by the coefficient C can be chosen in the numerator of the transfer function (26) using the principle of superposition. The next stage was the numerical solution of the obtained mathematical model of the oscillatory movements of the cleaner according to the developed theory.

To carry out numerical modelling, first of all, it was determined what the values of the parameters and quantities included in equations (21) and (22) could be. It concerns the force $\bar{N}$, the effect of which is transmitted to the wheel from the ground and drag forces $\bar{F}_r$ movement of the cleaning working body along the head of the root crop during the technological process of cleaning it from the remains of tops.

In general, the unknown normal reaction of the soil $\bar{N}$ can be determined by directly measuring it in the field. In this study, its value was determined by the calculation method at the static equilibrium of the root crop head cleaner. For this purpose, equation (19) was considered since $Q_\beta = 0$.

Resistance force $\bar{F}_r$ of the movement of the cleaning operating element along the head of the root crop was determined under the condition of its equality in size to the combing force Q of tops remains from the head of the root crop (Pogorely & Tatyanko, 2004). For the analytical expression of this combing force Q, the following steps were followed. For example, it is well known that the taproot stems of many root crops (e.g., sugar beet), located close to the head, have a cross-sectional shape close to a triangular shape. At the same time, it has a similarly triangular depression on one side. A conditional graphical model of the cross-section of a sugar beet tops petiole was developed (Fig. 3). The geometric dimensions of the section are shown on the graphical model itself. They were used to calculate the cross-sectional area of the petiole stalk.

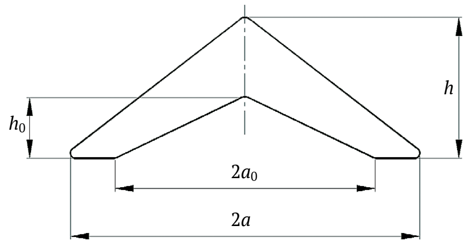

Figure 3. Graphical model of the cross-section of the petiole of a root crop
Source: compiled by the author

Further, it was assumed that the process of combing off the residual tops is carried out directly on the root crop head itself, in the place of conditional fixation of the petiole due to deformation of the mechanical shear (combing) of the petiole. Then the process of combing the tops from the head of the root crop will be possible if the following ratio is met (Pogorely & Tatyanko, 2004):

$$\frac{Q}{n \cdot F} \geq [t], \tag{27}$$

where Q – the force required to comb the tops off, N; $[t]$ – permissible tangential shear stresses for the stalk, $N{\cdot}mm^{-2}$; F – cross-sectional area of one petiole, mm^2; n – number of cuttings to be combed at the same time.

Using expression (27), the force Q, The number of tops necessary for combing the petioles from the root crop head was determined as follows:

$$Q = n \cdot F \cdot [t]. \tag{28}$$

Next, the necessary values included in expression (28) were determined. Thus, using the graphical model of the petiole (Fig. 3), the area of its cross-section was equal to:

$$F = \frac{1}{2} \cdot 2a \cdot h - \frac{1}{2} \cdot 2a_o \cdot h_o, \tag{29}$$

or

$$F = a \cdot h - a_o \cdot h_o. \tag{30}$$

Using expressions (28) and (30), the final force found is $\bar{Q}$ combing the petiole of the tops. It is equal to:

$$Q = (a \cdot h - a_o \cdot h_o) \cdot n \cdot [t]. \tag{31}$$

Considering the aforesaid, a specific numerical value of the force was determined as Q, which is required to comb one stalk of sugar beet tops. The values that can be used in the calculations were suggested by L. Pogorely and N. Tatyanko (2004). Specifically, the following linear dimensions of the cuttings were used: $a = 5$ mm; $a_o = 2$ mm; $h = 5$ mm; $h_o = 2$ mm. Permissible tangential stress values $[t]$ for this plant material were also used according to L. Pogorely & N. Tatyanko (2004) and were deemed equal: $[t] = 1.14 \cdot 10^6$ Pa.

Applying the expression (31) and the specified dimensions of the force Q of combing for 1 petiole stalk was:

$$Q = (5 \cdot 5 - 2 \cdot 2) \cdot 10^{-6} \cdot 1.14 \cdot 10^6 = 23.9, \text{N}. \tag{32}$$

Considering that the combing of green and strong petioles in this cleaner occurs on both sides and that each head of the root crop will contain on average 4 petioles that need to be separated (combed), then, based on the value obtained in (32), it was assumed that $F_r = Q \approx 100$ N.

The prototype of the root crop head cleaner was equipped with tracing wheels with pneumatic tyres 7.50R16, which are widely used in various agricultural machines. For them, the values of the stiffness coefficients c and damping μ were selected from the reference literature.

Radius r of a doubler wheel with a pneumatic tyre of this brand in an undeformed state was also selected from the reference literature. The force of gravity G of the cleaner was determined by weighing the improved design of the cleaner, considering some of the adhering soil. The moment of inertia I_{oz} of the root crop head cleaner was determined analytically. Parameters of the soil surface unevenness profile (h_o, l_1 and f), as well as the design parameters of this device (l, L, b and h), were set and changed during numerical calculations on a personal computer (PC). The speed of translational movement V of the cleaner was adopted according to its field tests.

The PTC Mathcad 15 software was used to perform numerical calculations. Using the obtained analytical dependencies, as well as design and kinematic parameters, the amplitude-frequency, and phase-frequency characteristics of the oscillatory movements of the cleaner, which was modelled as an oscillatory dynamic system, were calculated on a PC and plotted in the form of graphical dependencies.

RESULTS AND DISCUSSION

The nature of the curves reflected in the constructed dependencies precisely indicates the response of the specified dynamic system to external perturbations provided to it and changes in specific structural, kinematic, and operational parameters of the cleaner when it performs the technological cleaning process.

Numerical calculations of the obtained mathematical model are impossible without the use of constant values included in the final expression. The values of structural and force parameters (with an indication of their ranges of variation) used in the calculations are given in Table 1.

The graphical dependences obtained in the calculations make it possible to evaluate the behaviour of the cleaner as an oscillatory system under the action of a disturbing influence at the appropriate frequency, i.e. when the ordinates y change, due to the unevenness of the soil surface and the elastic-damping properties of the doubler wheels (stiffness coefficients c and damping μ), as well as some design parameters of the cleaner (e.g. length l and L).

It should be emphasised at once that the external disturbance, determined by the fluctuations of the ordinates y, given by expression (2), provides the frequency of oscillations of the main range, which varies from 0 to 24 s^{-1}. This is determined by the fact that the main tillage in the fields where root crops are grown forms zones of greater and lesser compaction, which alternate periodically. This is caused by the formation of soil layers that are laid down during ploughing. And although the high-frequency components of the bumps are levelled out later, their low-frequency components remain. Therefore, the representation of soil surface irregularities identified by harmonic functions is quite natural. Based on repeated measurements, it was found that in many cases, soil surface irregularities provide fluctuations in the longitudinal profile ordinates that reach 12 m^{-1}. If we assume that the translational movement of the cleaner will be carried out with a translational speed not exceeding 2.0 ms^{-1}, then the maximum frequency of oscillations of soil surface irregularities, based on which, in this theory, an external disturbance is formed, will be in the range of up to 24 s^{-1}. At the same time, if the behaviour of the dynamic system under consideration is not fully determined by the specified basic frequency range, it will be increased.

Table 1. Structural and kinematic parameters of the root crop head cleaner of improved design

Parameter	Determinant	Unit of measurement	Value
Distance from the suspension axis to the centre of mass	L	m	3.0...4.0
Distance from the suspension axis to the vertical axis of the doubler wheel	l	m	2.8...3.8
Distance from the ends of the rubber blades to the frame	h	m	0.685
Stiffness coefficient of pneumatic tyres for doubler wheels	c	$kN{\cdot}m^{-1}$	115...515
Damping coefficient of pneumatic tyres for doubler wheels	μ	$N{\cdot}s{\cdot}m^{-1}$	350...1350
Half the height of uneven ground	h_o	m	0.01...0.03
Step unevenness of the soil surface	l_1	m	0.70
Gravitation force	G	N	9100
Moment of inertia relative to the axis of rotation	I_{oz}	$N{\cdot}m{\cdot}s^2$	3000...3500
Rolling resistance coefficient of the doubler wheel	f	–	0.12...0.16
Resistance force for moving the cleaning implement on root crop heads	F_r	N	100
Normal reaction at the point of contact between the doubler wheel and uneven ground surface	N	N	4100
The radius of the doubler wheel in the undeformed state	r	m	0.370
Distance from frame to centre of doubler wheel	b	m	0.420
Translational speed	V	$m{\cdot}s^{-1}$	1.5...2.5

Source: compiled by the author

The amplitude-frequency characteristics of the dynamic system's working out of the oscillations of the soil surface unevenness profile at different values of the stiffness coefficient c of the pneumatic tyres of the doubler wheels were considered. As follows from the curves shown in Figure 4, the amplitude-frequency characteristics tend to increase. At the same time, the increase in the ratio c, for example, from 115 $kN{\cdot}m^{-1}$ to 315 $kN{\cdot}m^{-1}$, and then to 515 $kN{\cdot}m^{-1}$ makes this increase slower. In all cases, this increase is a negative development. Using pneumatic tyres for doubler wheels with the lowest values c, in our case $c = 115$ $kN{\cdot}m^{-1}$, the increase in the amplitude-frequency response occurs at a disturbance frequency from 0 to 28 s^{-1}. At a frequency of 28 s^{-1}, the peak value of this increase occurs, after which there is a sharp decrease in the amplitude-frequency response. It is known from the theory of automatic control that this is the so-called resonant peak, which corresponds to 1.54 $rad{\cdot}m^{-1}$, which is characterised by changes in the height of soil surface irregularities not exceeding 0.03 m. However, to match the resonance peak, the cleaner has to be rotated by an angle β no more than 3°. Then, when the frequency of oscillations of the disturbing effect on this dynamic system is increased by more than 28 s^{-1} and to values of 40 s^{-1} doubler wheels with the following coefficient values c, have a sharply decreasing amplitude-frequency response that is close to its desired value.

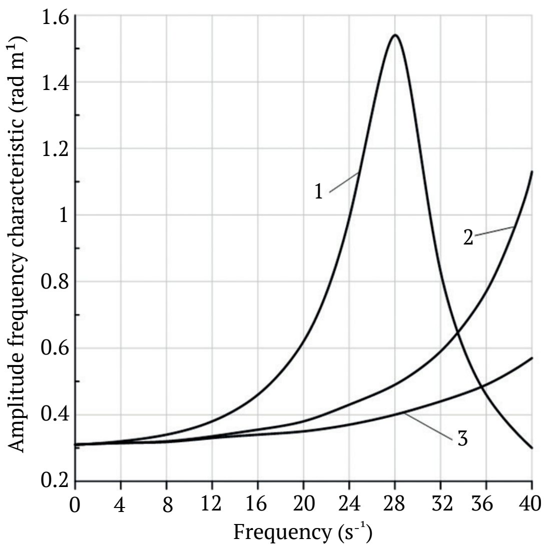

Figure 4. Dependence of the amplitude-frequency response of the cleaner's processing of oscillations of the soil surface unevenness profile on different values of the stiffness coefficient c pneumatic tyres of the doubler wheels

Note: 1 – 115 $kN{\cdot}m^{-1}$; 2 – 315 $kN{\cdot}m^{-1}$; 3 – 515 $kN{\cdot}m^{-1}$

Source: compiled by the author

Analysis of the amplitude-frequency characteristics of the dynamic system's working out of the soil surface irregularities profile at different values of the stiffness coefficient *c* at higher values, namely 315 kN·m^{-1} and 515 kN·m^{-1} shows that there are no longer any peak values, but only a corresponding slow increase, which makes it possible to recommend them as preferred. In addition, based on the graphs in Figure 4, even with a disturbance frequency of 24 s^{-1} an increase of coefficient *c* from 115 kN·m^{-1} to 515 kN·m^{-1} provides the required reduction of the amplitude-frequency characteristics, starting from 1.00 rad·m^{-1} (curve 1), then 0.42 rad·m^{-1} (curve 2) and 0.38 rad·m^{-1} (curve 3) respectively, a 42% reduction in the first case and a 10% reduction in the second case.

The phase-frequency characteristics were also plotted, which indicate a delay in the reaction of the dynamic system under consideration to the action of a disturbing influence (Fig. 5). As shown by the above phase-frequency characteristics at different values of the stiffness coefficient *c* of the pneumatic doubler wheels of the cleaner, an increase in this indicator leads to an improvement in the actual phase-frequency characteristics. At the same time, it is at the disturbance frequencies at the level of up to 12 s^{-1} that various coefficient values *c* do not significantly affect the delayed reaction time of the cleaner.

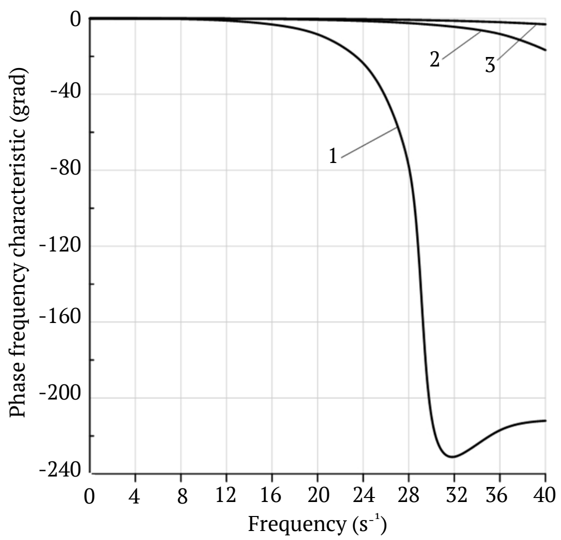

Figure 5. Dependence of the phase-frequency response of the cleaner's processing of oscillations of the soil surface unevenness profile on different values of the stiffness coefficient *c* pneumatic tyres of the doubler wheels

Note: 1 – 115 kN·m^{-1}; 2 – 315 kN·m^{-1}; 3 – 515 kN·m^{-1}

Source: compiled by the author

As can be seen from the changes in the phase-frequency characteristics shown in Figure 5, with an increase in the frequency of the disturbing effect from 12 s^{-1} to 32 s$_{-1}$ at *c*, which equals 315 kN·m^{-1} (curve 2) and *c*, which equals 515 kN·m^{-1} (curve 3) The values of these indicators in both cases are close to zero. From the theory of automatic control, it is known that this indicates that the reaction of the cleaner, as a dynamic system under consideration, is too late. This is the case that should not be considered desirable. At the value of the coefficient *c*, which equals 115 kN·m^{-1} at frequency y 32 s^{-1} the phase-frequency response increases, up to the value of 230o (4 rad), that, on the contrary, there is a possibility to consider this phenomenon positive.

Thus, the phase-frequency characteristics that characterize the delay in the response of the root crop head cleaner to the disturbing effect as shown by the calculations, are almost independent of the coefficient *c* of the stiffness of the tyres of its doubler wheels. As a result, if we assess the impact of the *c* cleare's functionality as a dynamic system under consideration, based on the obtained amplitude-frequency and phase-frequency characteristics, the optimal value *c* is an indicator whose value equals 315 kN·m^{-1}, provided that the fluctuations of the ordinates of the longitudinal profile of soil surface irregularities do not exceed the mark in 12 s^{-1}. To implement this numerical value of the coefficient *c*. The stiffness of the pneumatic tyres of its doubler wheels requires the calculation of specific indicators that will ensure it, provided that the doubler wheels of the previously mentioned brand are used. Coefficient *c* for a doubler wheel is almost entirely determined by air pressure ρ_a in its pneumatic tyre.

To perform this calculation, the normal strain h_{pt} of the pneumatic tyre was first determined following the known formula:

$$h_{pt} = \frac{G_k}{2\pi \cdot \rho_t \cdot \sqrt{r_0 \cdot r_c}}, \tag{33}$$

where G_k – normal load of doubler wheel, N; ρ_t – pneumatic tyre air pressure, Pa; r_o – wheel radius in the unloaded state, m; r_c – radius of curvature of the pneumatic tyre tread, m.

Using Hooke's linear law for elastic deformations, the elasticity coefficient was found to be c for the pneumatic tyre of the doubler wheel using the following expression:

$$c = \frac{G_k}{h_{pt}} = 2\pi \cdot \rho_t \cdot \sqrt{r_0 \cdot r_c}. \tag{34}$$

Finally, the air pressure ρ_t was determined in a pneumatic tyre, depending on the elasticity coefficient *c* and design parameters of the doubler wheel:

$$\rho_t = \frac{c}{2\pi \cdot \sqrt{r_0 \cdot r_c}}. \tag{35}$$

Since for the root crop head cleaner under consideration, the main parameter has the value $r_o \approx rc = r = 0.37$ m, of doubler wheels, the calculation showed that the air pressure inside its pneumatic tyres ρ_a should be approximately 135 kPa. It is quite possible to achieve this parameter for the specified pneumatic tyre. Also, the corresponding calculations were carried out on a PC and the amplitude-frequency and phase-frequency characteristics were obtained, showing their influence on the oscillations of the rear-mounted cleaner depending on another indicator of the pneumatic tyres of its doubler wheels, i.e., the damping coefficient μ. Thus, the amplitude-frequency and phase-frequency characteristics shown in Figure 6 and Figure 7 show that an increase in the external disturbance, i.e.,

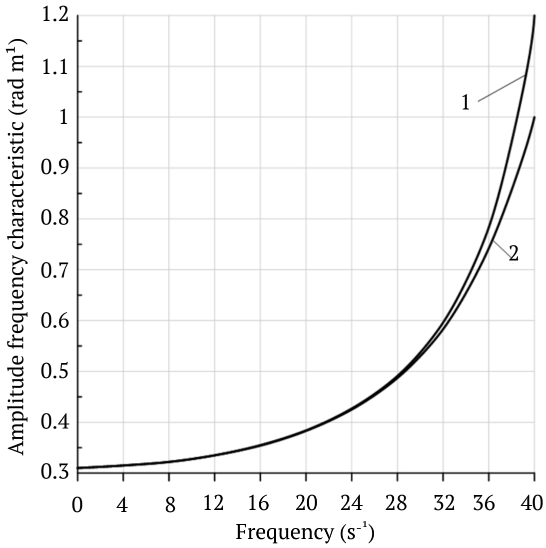

Figure 6. Dependence of the amplitude-frequency characteristics of the cleaner's response to vibrations of the soil surface unevenness profile on different values of the damping coefficient μ

Note: 1 – 350 N·s·m^{-1}; 2 – 1350 N·s·m^{-1}

Source: compiled by the author

fluctuations in the ordinates y longitudinal profile of soil surface irregularities, depending on the values of the coefficient μ have an upward trend.

However, the form of the obtained dependences shown in Figure 6 indicates that the nature of the change in these amplitude-frequency characteristics has little dependence on the value of the damping coefficient μ of of the pneumatic tyres of the cleaner's doubler wheels. At oscillation frequencies of the disturbing influence, which are in the range from 0 to 24 s^{-1}, this dependence is negligible, despite a significant increase in the coefficient μ.

Although outside this frequency range (higher than 24 s^{-1}) there is a difference between the obtained amplitude-frequency characteristics, but it is also insignificant. As follows from the curves shown in Figure 6, at the frequency of the disturbing effect from 30 s^{-1} to 40 s^{-1} of lower coefficient values μ, the higher value of the amplitude-frequency response corresponds to (curve 1). That is the lower the scattering (dissipative) potential of the purifier as a dynamic system, the more sensitive it is to the action of external disturbing influences and vice versa.

In the selected frequency range of the external disturbance from 0 to 24 s^{-1} the difference between the obtained phase-frequency characteristics shown in Figure 7 (curves 1 and 2) is insignificant since it does not exceed 1.5°. In general, the analysis of these indicators shows that the higher the value of the coefficient μ damping of a given dynamic system, which is shown by curve 2, the later this dynamic system reacts to the action of an external disturbance. If the frequency of the latter is only at the level of 40 s^{-1}. The difference in this delay is approximately 17° (meaning 0.3 rad) or 0.0075 s by time.

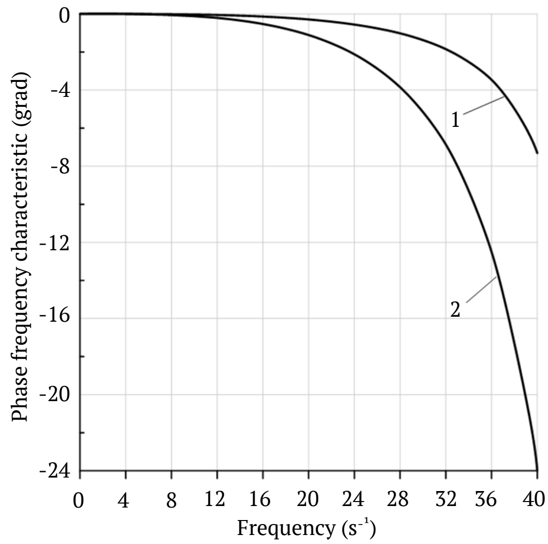

Figure 7. Dependence of the phase-frequency characteristics of the cleaner's response to vibrations of the soil surface unevenness profile on different values of the damping coefficient μ

Note: 1 – 350 N·s·m^{-1}; 2 – 1350 N·s·m^{-1}

Source: compiled by the author

Based on the assessment of the impact of the damping characteristics of the wiper's doubler wheels, there are two important cases:

- the invariance of the response of the cleaner to the disturbing effect when the coefficient is changed was established μ in the range 350 N·s·m^{-1} to 1350 N·s·m^{-1};
- the qualitative regularity of the growth of the phase-frequency characteristic of a given dynamic system is observed, provided that the value of its damping coefficient μ increases.

Next, amplitude and frequency characteristics of the cleaner, as a dynamic system, working out the disturbing effect associated with fluctuations in the profile of soil surface irregularities at different values l of the length were considered, i.e., the distance from the suspension axis of the cleaner to the vertical axis of its doubler wheel. As can be seen from the curves in Figure 8, the amplitude-frequency characteristics tend to increase at given lengths l. However, with a shorter length l, which equals 2.8 m the growth is the most intensive.

A significant improvement in the amplitude-frequency characteristics of the disturbance working out by the cleaner can be achieved by increasing the length of the l. That is, an increase in length l from 2.8 to 3.8 m ensures a decrease in the amplitude-frequency characteristics of the cleaner as a dynamic system. Analysis of the change in the amplitude-frequency characteristics of the cleaner's processing of oscillations of the soil surface roughness profile from different values of the length l, shown in Figure 9, shows that at oscillations in frequencies from 0 to 24 s^{-1} the difference in these characteristics can reach 0.22 rad·m^{-1}. At the same time, since the height of soil surface irregularities varies from 0.01 m to 0.03 m, this difference is from 0.13 to 0.39°.

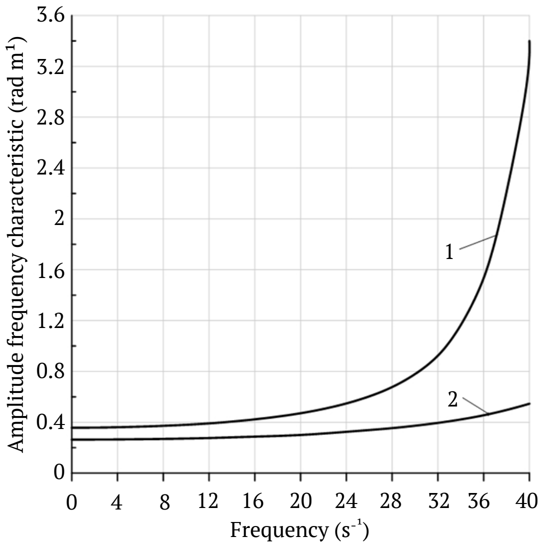

Figure 8. Dependence of the amplitude-frequency characteristics of the cleaner's response to vibrations of the soil surface unevenness profile on different values of the length l

Note: 1 – 2.8 m; 2 – 3.8 m

Source: compiled by the author

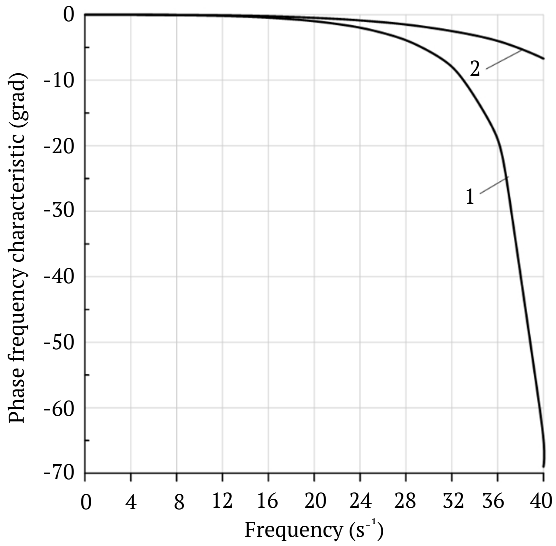

Figure 9. Dependence of the phase-frequency characteristics of the cleaner's response to vibrations of the soil surface unevenness profile on different length values l

Note: 1 – 2.8 m; 2 – 3.8 m

Source: compiled by the author

The graphical dependences of the phase-frequency characteristics of the cleaner's processing of oscillations of the soil surface roughness profile at different values of the length l were constructed, which show that increasing the length of the l leads to a decrease in the phase-frequency response. This is undesirable for the cleaner as an oscillating system. This shows that at the oscillation frequency of the disturbing influence, which is equal to 40 s^{-1} The decrease is 1 rad. Considering the time over which this decrease occurs, it is only 0.025 s, which indicates the high sensitivity of the cleaner to this disturbance. Considering the frequency changes from 0 to 24 s^{-1}, length l changes do not cause corresponding changes in phase-frequency characteristics.

It can be argued that the oscillation of the cleaner is not affected by changes in length l. The installation location of the doubler wheels on the cleaner frame should be considered close to the cleaning working bodies, which will certainly create conditions for accurate copying of root crop heads from the remains of tops on the root. Amplitude and frequency characteristics of the cleaner, as a dynamic system, working out the disturbing effect associated with fluctuations in the profile of soil surface irregularities at different length values L, thus distance from point O of the cleaner suspension to the centre of mass of the cleaner, i.e., the point C. These changes in the amplitude-frequency characteristics are shown in Figure 10.

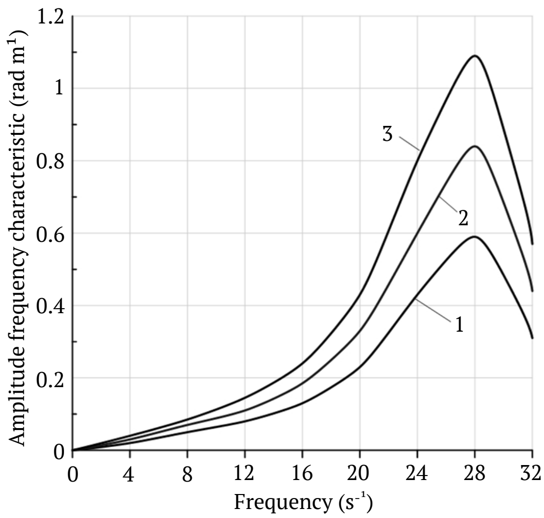

Figure 10. Dependence of the amplitude-frequency characteristics of the cleaner's processing of oscillations of the soil surface unevenness profile from different length values L

Note: 1 – 2.5 m; 2 – 3.0 m; 3 – 3.5 m

Source: compiled by the author

As the view of the above amplitude-frequency characteristics shows, it is necessary to strive to ensure that the centre of mass of the cleaner (point C, Fig. 2) was as close as possible to the pole of the instantaneous centre of rotation of the rear linkage of the implement (point *O*). In this case, the amplitude-frequency characteristics of the cleaner in terms of its working out of vibrations of the ordinates of the longitudinal profile of soil surface irregularities are reduced.

It should be noted that the maximum values of these amplitude-frequency characteristics occur at the frequency of the disturbance, which is 28 s^{-1}. Two significant cases follow from this. First, the frequency value is outside the accepted range of 0 to 24 s^{-1}. Secondly, the resonance peaks occurring at this frequency (however, as in Fig. 4) are far from those that may be undesirable for the design of a root crop head cleaner from the remains of tops on the root. Moreover, the lower the frequency of oscillations of the disturbing influence, the closer the obtained amplitude-frequency characteristics are to the zero mark, which is certainly desirable. A lower frequency corresponds to a longer period of oscillations of the ordinates of the profile of soil surface irregularities. This indicates the need to improve the quality of basic tillage in the process of growing both root crops and any other crops, which are known to form a harmonious appearance of surface irregularities.

Dependence of the phase-frequency characteristics of the cleaner's processing of oscillations of the soil surface unevenness profile on different values of the length *L*, which was established by computer calculations, showed the complete invariance of changes in this constructive index concerning the phase shift. Therefore, these graphical dependencies are not presented here. Analysing the results of the study and comparing them with previously published works, it should be noted that the closest earlier cited studies of M. Khelemendyk (2011); V. Bulgakov *et al.* (2017; 2023), while aimed to obtain qualitative indicators of oscillatory movements of the cleaner attached to the aggregating tractor, do not cover studies of all possible options for the influence of the design, kinematic and operational characteristics of the cleaners themselves on the stability of their movements in the longitudinal-vertical plane. In addition, the study by V. Bulgakov *et al.* (2017) does not perform a qualitative analysis of the oscillatory movement of the cleaner itself, but only provides a detailed mathematical formalisation of the cleaner's movement over uneven soil surfaces.

In addition, although the influence of the elastic-damping properties of the caster wheels on the stability of the wiper as a dynamic system, in particular, the elasticity coefficient *c* and the damping coefficient μ is certainly important, considering several other design parameters of the wiper as a whole can cause even better conditions for stable movement. In addition, increasing the stability of the cleaner frame in the longitudinal-vertical plane by changing only the elastic-damping properties of the doubler wheels and finding a place for their placement on the mainframe can cause controversial situations related not only to the accuracy of doubler soil surface irregularities in the row spacing but also directly in the row itself. For example, for a certain set of parameters, their influence, i.e., even greater sensitivity to external disturbances, can be so intense that other indicators that ensure the accuracy of copying each head of the root crop will deteriorate. In this case, the most suitable conditions are compromise solutions that will allow the prevailing factors to be the main ones. These factors take place only in the case of movement of root crop head cleaners from the remains of tops on the root when both the stability of movement and the accuracy of copying each root crop head are extremely important.

The search for the accuracy of copying the rows of sugar beet root crops during the harvesting of tops, and, accordingly, the quality indicators of this technological process were considered by I. Storozhuk & V. Pankiv, (2015) and E. Berezhenko (2020). However, theoretical studies in this particular formulation are not considered here and the amplitude-frequency and phase-frequency characteristics were not provided. Some results of solving a similar problem were considered in the works of V. Nadykto *et al.* (2020), but they consider agricultural machines hitched to the rear of the aggregating tractor or frontally mounted ploughing and chopping machines. Moreover, all these works consider the stability of the movement of the aggregated agricultural machines in the longitudinal-vertical plane. However, the specifics of the functioning of tractor-mounted machines and the studies in the known works differ significantly from the operation of the cleaner, where the conditions of stable movement under the influence of external disturbances, such as uneven soil surface in the rows of crops, must be combined with external disturbances in the row itself.

Fundamental studies of the stability of agricultural machines mounted on tractor-mounted aggregates were presented by H. Beloev *et al.* (2021). However, the study considered various effects of external disturbances on the forced oscillations of ploughs in the longitudinal-vertical plane, which, although generally important, are not required to be sensitive to the copying of soil surface irregularities, as is the case when considering the oscillations of cleaners. The main focus is on the vertical oscillations of the front and rear axles of the power vehicle itself, i.e., the wheeled tractor. In this study, the amplitude-frequency and phase-frequency characteristics of the tractor are also plotted following the oscillations of the unevenness of the soil surface. However, the energy vehicle is essentially a symmetric dynamic system, and the vibrations of its front and rear axles are interconnected. Consideration of the oscillatory movements of the rear-mounted implement as a dynamic system that represents a "physical pendulum" is not presented here.

Thus, using the constructed new mathematical model, which is the theory of oscillatory movements in the process of plane-parallel oscillatory motion in the longitudinal-vertical plane, it was possible to determine the design, kinematic and operational parameters of the rear-mounted peeler that ensures the stability of its movement using a computer. This, in turn, will contribute to a significant improvement in the quality of the technological process performed by the root crop head cleaner to remove the remains of tops on the roots. A comparative analysis of the numerical values of the obtained parameters with the results of similar studies by other authors indicates the reliability of the developed theory.

CONCLUSIONS

The basic principles of the theory of oscillatory motion of the root crop head cleaner in the longitudinal-vertical plane, mounted behind a wheeled aggregating tractor, have been developed, which allow numerical modelling on a PC to determine its dynamic properties depending on the action of the disturbing influence caused by changes in the ordinates of soil surface irregularities at the appropriate frequency.

The influence of the stiffness of pneumatic doubler wheels, characterised by the coefficient c on the functioning of the cleaner as a dynamic system under consideration, based on the obtained amplitude-frequency and phase-frequency characteristics, and it is established that the optimal value of this indicator is 315 $kN \cdot m^{-1}$, provided that the oscillations of the ordinates y of the longitudinal profile of soil surface irregularities will not exceed 12 s^{-1}. The implementation of this condition can be ensured by the formation of air pressure in the pneumatic tyres of the wiper's doubler wheels, which should be 135 kPa.

Based on the evaluation of the influence of the damping characteristics of the pneumatic doubler wheels of the cleaner, it was found that when the coefficient μ changes from 350 $N \cdot s \cdot m^{-1}$ to 1350 $N \cdot s \cdot m^{-1}$ an invariance of the delay in the response of the cleaner to the disturbance in the frequency range from 0 to 24 s^{-1} is present. At the same time, there is a regularity of growth of the phase-frequency characteristic of this dynamic system under the condition of increasing the value of its damping coefficient μ.

The influence of such a design parameter as the distance from the suspension axis of the rear-mounted cleaner to the vertical axis of its doubler wheel in the considered frequency range from 0 to 24 s^{-1} of oscillations of the ordinates of the longitudinal profile of soil surface irregularities is insignificant. This follows from the nature of changes in the obtained amplitude-frequency and phase-frequency characteristics. However, to ensure more accurate copying of root crop heads, which will generally improve the quality of this technological process, the place of installation of the doubler wheels on the cleaner frame should be considered their close location to the cleaning working bodies. To improve the dynamics of oscillatory movements of the rear-mounted cleaner in the longitudinal-vertical plane, its centre of mass should be located as close as possible to the pole of the instantaneous centre of rotation of the rear hitch of the aggregating tractor.

The subject of the next theoretical study should be the construction of a mathematical model in which the irregularities of the soil surface are approximated by a non-harmonic curve, and the external disturbance is formed by a random function. The next stage of the research should include comprehensive experimental studies to determine the effect of angular oscillatory movements of the rear-mounted cleaner on the quality of separation of the residual tops from the heads of root crops on the root. In conducting these field studies, a wide range of changes in the design, kinematic and operational parameters of the cleaner should be used when performing this technological process. It is also necessary to find an opportunity to determine the qualitative indicators of cleaning by a variable input parameter, namely different frequencies of oscillations of the disturbing effect created by different types of soil surface irregularities.

ACKNOWLEDGEMENTS

None.

CONFLICT OF INTEREST

None.

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Розробка теорії руху очисника головок коренеплодів у повздовжньо-вертикальній площині, навішеного позаду колісного трактора

Анотація. Актуальність розробки аналітичних методів для вивчення стійкості руху очисників коренеплідних культур на тракторах виникає з необхідності покращення ефективності і якості очищення головок коренеплодів, оскільки від цього залежить якість кінцевого продукту та його товарні властивості. Метою даного дослідження є підвищення стійкості руху очисника, шляхом побудови математичної моделі його коливального руху у повздовжньо-вертикальній площині, та її числового розв'язування для визначення впливу параметрів у відповідь на зовнішній збурювальний вплив. Для виконання поставленої мети була розроблена нова теорія руху очисника головок коренеплодів у повздовжньо-вертикальній площині, навішеного позаду колісного трактора, під дією збурювального впливу нерівностей повздовжнього профілю поверхні ґрунту на амплітудні та фазові частотні характеристики його кутових коливань. Шляхом проведених на персональному комп'ютері числових розрахунків знайдені умови, за якими підвищення стійкості руху очисника у зазначеній площині буде в разі, коли коефіцієнт c жорсткості пневматичних шин копіювальних коліс буде мати значення 315 kN·m^{-1}. Такий результат забезпечується тоді, коли тиск у шинах пневматичних коліс буде 135 kPa. Що стосується демпфувальних характеристик шин пневматичних копіювальних коліс, які визначаються коефіцієнтом μ, то встановлено, що при його зміні від 350 N·s·m^{-1} до 1350 N·s·m^{-1} має місце інваріантність запізнення реакції очисника на збурювальний вплив при зміні його частот від 0 до 24 s^{-1}. Також встановлено, що вплив геометричних розмірів очисника, є несуттєвим у діапазоні частот коливань ординат повздовжнього профілю нерівностей поверхні ґрунту від 0 до 24 s^{-1}. Це випливає з характерів змін отриманих амплітудно-частотних та фазово-частотних характеристик. Методика побудови математичної моделі плоскопаралельного коливального руху очисника може бути використаною при аналогічних аналітичних дослідженнях інших задньонавішених на колісний агрегатуючий трактор сільськогосподарських машин

Ключові слова: нерівності ґрунту; еквівалентна схема; коливання; диференціальні рівняння; амплітудно-частотні; фазово-частотні характеристики