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Dynamics of the movement of the ripping tool for surface tillage

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Abstract. The study aims to analyze the stability of the movement of the ripper tool for surface tillage. The basic principles and methods of classical mechanics, mathematical analysis, and statistics were used in this study. The dynamics of the movement of a ripper tool for surface tillage are considered, depending on the forces acting on it and its design parameters. A computational dynamic model is developed, and an equation describing the angular oscillations of the longitudinal links of the parallelogram ripper mechanism is obtained. It is established that the uniformity of the depth of the cultivator depends on the amplitude and frequency changes of the components of the disturbing force, physical and mechanical properties of the soil, the moment of inertia of the Ripper, the length of the longitudinal links parallelogram mechanism, the forces of the pre-tension pressure springs, and its stiffness. It is established that the required uniformity of the depth of tillage can be achieved by selecting the pre-tension force of the pressure spring and its rigidity. Theoretical and experimental studies have established that the required uniformity of the processing depth with minimal energy consumption of the soil is provided with a pre-tension force of the pressure spring of the ripper attachment mechanism of 350 n, spring stiffness of 40 n / cm, and a speed of 2.0 m/s.

1 Introduction

In Uzbekistan, large-scale measures are being taken to develop resource-saving equipment and technologies with high efficiency [1-23], which reduce labor and energy costs, save resources in the cultivation of agricultural crops and ensure high-quality harvesting with the least losses [3, 10, 11, 19, 20].

Farmers and farmers are mostly engaged in the cultivation of vegetables, fruits and melons, and small plots. To increase the efficiency of their economy, they receive 2-3 harvests of these crops on the same plot during the year. Therefore, they till the soil in different periods of the year (spring, summer, autumn). This means that machines and tools for such farms should be highly maneuverable, easy to operate, light, and designed to work in different conditions.

In the conditions of cotton monoculture, wide-reach, large-sized machines, and tools with a powerful tracked tractor were developed for working in large farms and on large areas.

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It is obvious that in the conditions of farms and peasant farms, tracked tractors are economically unprofitable due to their high cost, low utilization rate during the year, and even more so in small areas. This is also evidenced by the rich foreign experience where wheeled tractors are used everywhere in farms in all agricultural operations.

However, there are no-tillage machines for aggregating with wheeled tractors in production, and there have been practically no developments on them. Taking this into account, the Republic is currently actively conducting research and development projects aimed at creating a new generation of wheeled tractors of class 0.6-1.4 for farmers and peasant farms and a train of machines and tools for them.

One of the most important in the agrotechnical complex for preparing the soil for sowing crops is the surface treatment (harrowing) of fields. It is carried out to preserve moisture, destroy blocks, destroy sprouting weeds, level the field surface somewhat, and prevent the introduction of salts into the upper layers of the soil [31-32].

This work aims to develop a technological scheme and substantiate the main parameters of the tool for surface tillage for wheeled tractors of class 0.6-1.4.

2 Methods

The basic principles and methods of classical mechanics, mathematical analysis, and statistics were used in this study

Taking into account the above, we have developed a scheme and manufactured a tool for surface tillage. In the tool, two rippers are installed on one row. The ripper consists of two transverse bars (pipes) with teeth. It is installed on the hitch bar using two parallelogram mechanisms with pressure springs [35].

During the tool's operation, due to the variability of the physical and mechanical properties of the soil, the uneven micro-relief of the force fields R_x, R_z Figure 1, acting on the ripper, continuously change. As a result, the balance of the ripper is constantly disturbed, and it makes angular fluctuations that lead to a change in the depth of tillage [36].

The calculated dynamic model of the ripper is shown in Figure 1. It consists of a frame 1, a parallelogram hitching mechanism 2, and a ripper 3 [35, 37, 38].

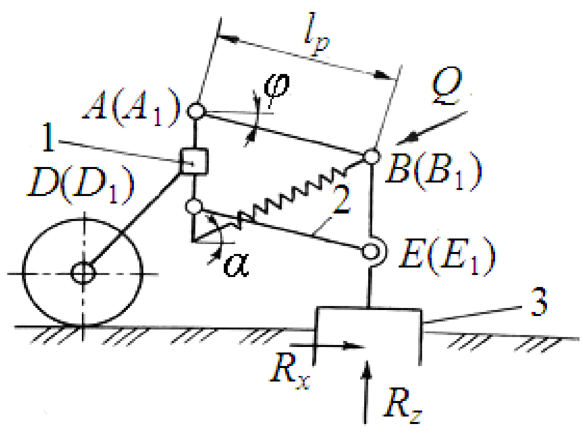

Fig. 1. Calculated dynamic model of the ripper

Assume that the longitudinal links AB (A_1 B_1) and DE (D_1E_1) (links A_1 B_1 and D_1E_1 are not shown in the figure) of the parallelogram mechanism are uniform and thin, at points A (A_1)

and D (D_1) are pivotally attached to the frame, and at points B (B_1) and E (E_1) – with a ripper.

Figure 1 presents: ℓ_n is length longitudinal parts of the hitch mechanism; R_x, R_z are horizontal and vertical components of force resistance of the soil; φ is the angle of deflection of the longitudinal links from the equilibrium position; Q is the pressure force of a pressure spring; α_0 is the angle of the force vector Q_0 to the horizon.

We assume that: the movement of the Assembly in a straight line and at a constant speed; the impact of the frame on the cultivator slightly and theoretical calculations can be ignored; the deviation of the working body from the equilibrium position occurs in the range of small angles and increments the second and more degrees can be neglected; the equilibrium position of the longitudinal parts of the hitch mechanism horizontal.

3 Results and Discussion

We also assume that the vertical reaction R_z soil cultivator is the sum of the elastic forces F_y, linearly dependent on its vertical movement, the damping forces (viscosity) R_C, linearly dependent on the speed of vertical movement of the Ripper, and disturbing forces $\Delta R_z(t)$ arising from the variability of physico-mechanical properties of soil, i.e.

$$R_z = F_y + R_c + \Delta R_z(t) \tag{1}$$

The friction in the joints A (A_1), B (B_1), D (D_1), and E(E_1) is neglected due to their smallness.

For the generalized coordinate, we take the angle φ of the deviation of the longitudinal links of the parallelogram mechanism from the horizontal.

Applying the differential equation of rotation of a rigid body around a fixed axis and taking into account the results of [38, 39], we obtain:

$$\begin{gathered} J\frac{d^2\varphi}{dt^2} = -R_x\ell_r\sin\varphi + Q_r\ell_r\sin\varphi + \left[m_r + 0{,}5\left(m_v + m_n\right)\right]x \\ xg\ell_p\cos\varphi + Q_v\ell_p\cos\varphi - R_z\ell_p\cos\varphi \end{gathered} \tag{2}$$

or taking into account (1)

$$\begin{gathered} J\frac{d^2\varphi}{dt^2} = -R_x\ell_p\sin\varphi + Q_r\ell_p\sin\varphi + \left[m_r + 0{,}5\left(m_v + m_n\right)\right]x \\ xg\ell_p\cos\varphi + Q_v\ell_p\cos\varphi - \left[F_u + R_s + \Delta R_z(t)\right]\ell_p\cos\varphi \end{gathered} \tag{3}$$

Since the angle φ is quite small, then, gently $\sin\varphi \approx \varphi$ and $\cos\varphi \approx 1$, we get

$$\begin{gathered} J\frac{d^2\varphi}{dt^2} = \left(Q_r - R_x\right)\ell_p\varphi + \left[m_r + 0{,}5\left(m_v + m_n\right)\right] + g\ell_p + \\ + Q_v\ell_p - \left[F_u + R_s + \Delta R_z(t)\right]\ell_p\cos\varphi \end{gathered} \tag{4}$$

where J is the moment of inertia of the ripper relative to the suspension axis; m_r is the mass of the ripper; m_n, m_v, is the mass of the lower and upper longitudinal links of the

hitch mechanism; Q_r, Q_v are the horizontal and vertical components of the spring pressure force.
To the static equilibrium position

$$F_y = \Delta_{ST} K_{zr} Z_p \quad (5)$$

$$R_s = 0 \quad (6)$$

$$Q = Q_0\,; \;\; Q_r = Q_0 \cos\alpha_0; \;\; Q_v = Q_0 \sin\alpha_0 \quad (7)$$

$$\Delta R_z(t) = 0 \quad (8);$$

$$\varphi = 0 \quad (9)$$

where Δ_{ST} is the vertical movement of the ripper under the action of its weight force and the pre-tension force of the spring; K_{zr} is the number of ripper teeth; Z_p is the coefficient of soil hardness related to one tooth; Q_0 is spring pre-tensioning force; α_0 is the angle of inclination of the force vector to the horizon.
When the ripper deviates from the equilibrium position by an angle φ,

$$F_y = \left(\Delta_{ST} + \ell_p \varphi\right) K_{zr} Z_p \quad (10)$$

$$R_s = \ell_p \varphi\, K_{zr} v_p \quad (11)$$

$$Q_r = \left[Q_0 + Z\ell_p\left(1-\cos\varphi\right)\right]\cos\alpha \approx Q_0 \cos\alpha \quad (12)$$

$$Q_v = \left[Q_0 + Z\ell_p \sin\varphi\right]\sin\alpha \approx \left(Q_0 - Z\ell_p\varphi\right)\sin\beta \quad (13)$$

where v_p is the coefficient of resistance (viscosity) of the soil, reduced to one tooth; Z is the stiffness of the pressure spring. Substituting these values F_y, R_c, Q_r and Q_v and Q_v in (4) and assuming that we have $\alpha \approx \alpha_0$

$$J\frac{d^2\varphi}{dt^2} = \left(Q_0 \cos\alpha_0 - R_x\right)\ell_p + \left[m_r + 0{,}5\left(m_v + m_n\right)\right] g\ell_p + \left(Q_0 - Z\ell_p\varphi\right)\ell_p \sin\alpha_0 - \left[\,\left(\Delta_{st} + \ell_p\varphi\right) K_{zr} Z_p + K_{zr} v_p \ell_p \varphi + \Delta R_z(t)\,\right]\ell_p \quad (14)$$

In the static equilibrium position

$$\left[m_r + 0{,}5\left(m_v + m_n\right)\right] g\ell_p + Q_c \ell_p \sin\alpha_0 - - \Delta_{st} + \ell_p R_{zr} Z_p\, \ell_p = 0 \quad (15)$$

Taking this into account, the differential equation of the angular vibrations of the ripper is written as:

$$J\frac{d^2\varphi}{dt^2} = v_p K_{zr} \cdot \ell_p^2 \frac{d\varphi}{dt} + \left(R_x - Q_0 \cos\alpha_0 + + Z_p \ell_p \sin\alpha_0 + Z_p K_{zr} \ell_p\right)\ell_p \varphi = -\Delta R_z(t)\ell_p \quad (16)$$

Due to the variability of R_x, equation (16) is a linear inhomogeneous second-order differential equation with a variable coefficient.

From the course of the theory of oscillations, it is known that in the system described by equation (16), theoretically, various parametric resonances are possible. However, numerous experimental studies conducted in laboratory and field conditions [10, 11] suggest that due to the large damping capacity of the soil, parametric fluctuations of the ripper are not observed. It performs forced oscillations under the action of the force $R_z(t)$. Therefore, in further studies, we will consider the force Rx to be a constant value equal to its average value.

Following the above, we consider the forced vibrations of the ripper.

Assuming that the disturbing force acting on the ripper changes according to the harmonic law, i.e.

$$\Delta R_z(t)=\sum_{n=1}^{n_1}\Delta R_p\cos(n\omega t-\beta_p) \tag{17}$$

where ΔR_p is the amplitude of the corresponding harmonic; n=1,2…n_1 is the number of harmonics (n_1 is number of the last, achieve together), ω is the circular frequency change of the disturbing forces.

Substituting the value in equation ΔR_p(t) (16), we get:

$$J\frac{d^2\varphi}{dt^2}+v_pK_{zr}\ell_p^2\frac{d\varphi}{dt}+\left(R_x-Q_0\cos\alpha_0+\right.$$
$$\left.+Z_pK_{zr}\ell_p\right)\ell_p\varphi=-\left(\sum_{n=1}^{n_1}\Delta R_p\cos(n\omega t-\beta_p-\delta_p)\right)\ell_p \tag{18}$$

Solving (18), we obtain the following equation, which determines the forced oscillations of the ripper

$$\varphi(t)=\frac{\ell_n}{J}\sum_{n=1}^{n_1}\frac{\Delta R_p\cos(n\omega t-\beta_p-\delta_p)}{\sqrt{\left[\frac{\left(R_x-Q_0\cos\alpha_0+Z\ell_p+Z_p\ell_pK_{zr}\right)\ell_p}{J}-(n\omega^2)^2\right]+\frac{4v_p^2K_{zr}^2\ell_p^4}{J^2}(n\omega)^2}} \tag{19}$$

Where

$$\beta_p=a\,rctg\frac{v_pK_{zr}\ell_p^2\omega}{\left(R_x-Q_0\cos\alpha_0+Z\ell_p\sin\alpha_0+Z_pK_{zr}\ell_p\right)\ell_p-J(n\omega)^2}-\text{phase shift}$$

n=1,2,…n (20)

From the analysis of the expression (19), it follows that the uniformity of the depth of the cultivator depends on the amplitude and frequency changes of the components of the disturbing force, physical and mechanical properties of the soil, the moment of inertia of the Ripper, the length of the longitudinal links parallelogram mechanism, the forces of the pre-tension pressure springs, and its stiffness.

According to the formula (19), the dependence of the change in the amplitude *A* of the forced vibrations of the ripper on the stiffness and tension force of the spring (reloader) is constructed (Fig.2) l_p= 0.40 m; n=1; =300 N; α_0=45°; K_{zr}= 20 pcs; ω=1 S-1; v_p=20·N / cm; J=$m_p \ell_n^2$; m_p= 24 kg; Z_n=4000 N / m; R_x=800 N [10].

As can be seen from Fig. 2, an increase in the stiffness of the pressure spring from 25 to 70 N / cm leads to a decrease in the amplitude of the angular vibrations of the ripper, and consequently, an increase in

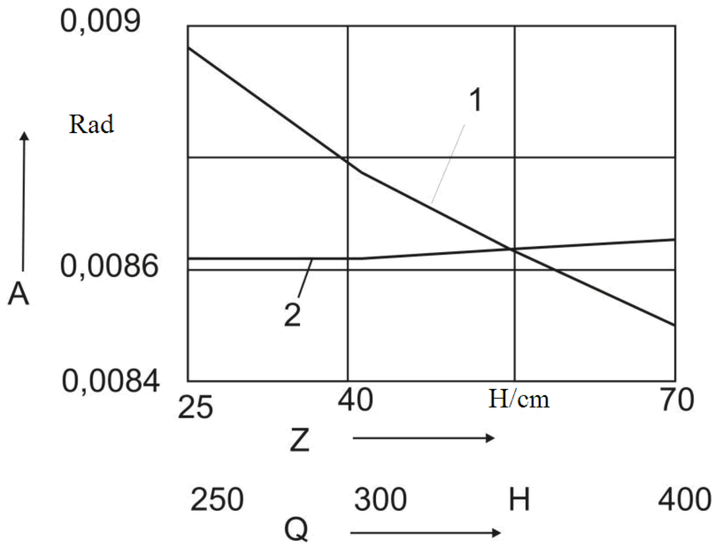

Fig. 2. Change in the amplitude *A* of the angle of deviation φ (t) of the longitudinal links of the parallelogram mechanism from the horizontal, depending on the stiffness *Z* and the tension force *Q* of the spring (reloader): 1-A=f (Z); 2-A=f (Q)uniformity of the depth of tillage. A change in the spring tension force from 250 to 400 N does not significantly affect these indicators.

4. Conclusions

1. A computational dynamic model is developed, and an equation describing the angular oscillations of the longitudinal links of the parallelogram ripper mechanism is obtained.
2. It is established that the required uniformity of the depth of tillage can be achieved by selecting the pre-tension force of the pressure spring and its rigidity.
3. Theoretical and experimental studies have established that the required uniformity of the processing depth with minimal energy consumption of the soil is provided with a pre-tension force of the pressure spring of the ripper attachment mechanism of 350 N, spring stiffness of 40 N / cm, and a speed of 2.0 m/s.

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