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ANALYSIS OF THE STABLE MOVEMENT OF A GEAR SOFTENER IN TERMS OF WORKING DEPTH

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Annotation. *This article provides a theoretical and practical analysis of the stable motion of a toothed рыхлитель (soil loosener) with respect to tillage depth. The influence of variable physical–mechanical properties of soil, surface irregularities, and angular oscillations caused by external forces on the tillage depth is examined. To ensure stable operation of the toothed loosener, differential equations of its angular oscillations are formulated, and based on their solutions, the maximum deviations are determined. According to the research results, it is substantiated that ensuring the stability of tillage depth largely depends on the optimal selection of structural parameters, in particular, the stiffness of pressure springs.*

Keywords*:* *Gear softener, tillage depth, stability, angular oscillations, differential equation, physical and mechanical properties of soil, reaction forces, spring stiffness, forced vibrations, agrotechnical requirements.*

High-quality soil tillage in agriculture is one of the key factors in increasing crop yield. In particular, the efficient operation of toothed looseners used for inter-row cultivation of cotton plays a decisive role in meeting agrotechnical requirements. In this process, maintaining a uniform tillage depth determines the effectiveness of soil aeration, moisture retention, and weed control [1].

Practice shows that during the operation of a gear softener, the forces acting on its working elements vary due to changes in soil density, moisture, and hardness, as well as surface irregularities of the field. This leads to a loss of motion stability of the arperate, particularly causing variations in tillage depth [2]. As a result, the gear softener performs not only translational motion but also angular oscillatory motion.

Therefore, ensuring the stability of tillage depth for a gear softener is an important scientific and practical problem. Solving this problem requires a thorough analysis of its dynamic state, i.e., the forces acting on it and their variation over time. Considering the above factors, it is important to theoretically study the dynamic behavior of the gear softener during operation, particularly its angular oscillations. For this purpose, the system of forces acting

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on the loosener, its equilibrium state, and the process of deviation from this state are analyzed based on mathematical modeling. Below, the formulation and analysis of differential equations describing the angular oscillations of the toothed loosener are presented. Due to the variability in moisture, density, and hardness of the soil between cotton rows, as well as the presence of surface irregularities, the reaction forces $\sum R_b$ and $\sum R_Z$ acting on the gear softener continuously change. As a result, during operation, the gear softener not only moves translationally in the longitudinal-vertical plane but also performs angular oscillations about point O (Fig. 1). These angular oscillations lead to variations in tillage depth, thereby disrupting its stability.

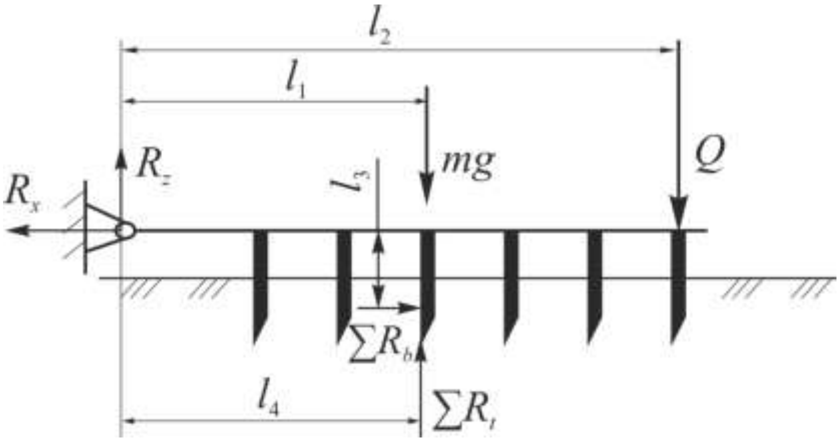

Figure 1. Forces acting on the gear softener during operation.

In order for the gear softener to work the soil at the same depth, the amplitude of its angular vibrations should be as small as possible. To solve this problem, the differential equation of the angular vibrations of the gear softener is formulated and solved [3].

To formulate the differential equation of the angular vibrations of the gear softener, we assume the following constraints:

- the tiller unit equipped with gear softeners moves between the cotton rows at a constant speed;

- the friction forces generated at the hinge connecting the gear softener with the shaft of the cotton cultivator (point O in Fig. 2) are small and do not affect its angular vibrations;

- the linear and angular vibrations of the tiller unit do not affect the angular vibrations of the gear softener;

- the equilibrium position of the gear softener is horizontal and its angle of deviation from this position is small.

According to this adopted calculation scheme and presented in Figure 2, the differential equation of the angular vibrations of the gear reducer relative to the “O” hinge is as follows [3].

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$$J\frac{d^2\varepsilon}{dt^2}=mgl_1+Ql_2-\left(\sum R_b\right)\left(l_3-l_1\alpha\right)-\left(\sum R_Z\right)l_4, \tag{1}$$

here is the moment of inertia of the J-gear softener relative to the “O” hinge, kg·m^2;
ε-angle of deviation of the gear softener from the equilibrium position, rad;
t -time, s.

We express the vertical reaction force $\left(\sum R_t\right)$ acting on the gear softener through the forces that depend on the stiffness and resistance coefficients of the soil and arise from the variability of its physical and mechanical properties [4], i.e.

$$\sum R_z=\sum R_d+\sum R_v+\sum R_t, \tag{2}$$

here $\sum R_d$-the power of the soil's fertility, N;
$\sum R_v$-force depending on the soil resistance coefficient, N;
$\sum R_t$-force resulting from variations in the physical and mechanical properties of the soil, N.

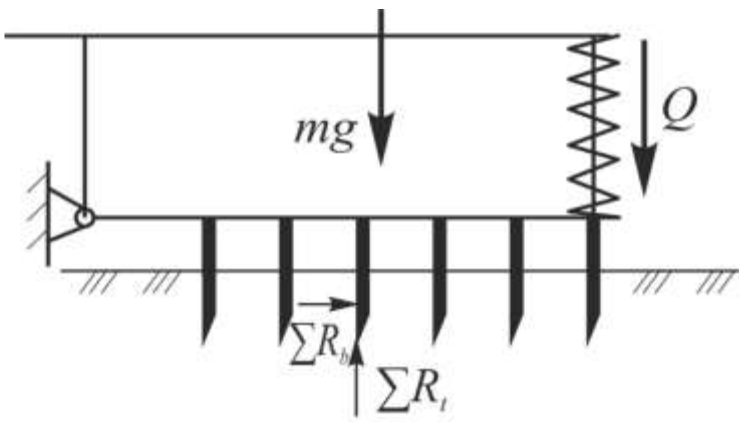
Taking into account expression 2), expression (1) becomes:

$$J\frac{d^2\varepsilon}{dt^2}=mgl_1+Ql_2-\left(R_{tq}\right)\left(l_3+l_1\alpha\right)-\left(\sum R_d+\sum R_v+\sum R_t\right)l_4 \tag{3}$$

Taking into account expressions (1) and (2) and assuming $l_3=l-0{,}5h$ [5] we can reduce expression (3) to the following form.

$$2J\frac{d^2\varepsilon}{dt^2}=mgL+2QL-\left(R_{tq}\right)\left(2l-h+L\alpha\right)-\left(\sum R_d+\sum R_v+\sum R_t\right)L$$

(4)



a)

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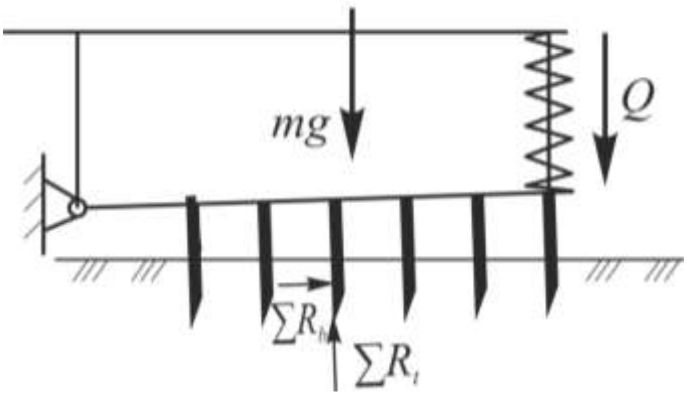

b)

a) gear softener in equilibrium; *b)* gear softener in an unbalanced state

Figure 2. Scheme for studying angular vibrations of a gear softener

Gear softener in equilibrium (Figure 2, a)

$$\sum R_d = n_t h_{o'} C_t , \quad (5)$$

$$\sum R_v = 0 ; \quad (6)$$

$$\sum R_t = 0 ;$$

and

$$Q = Q_o , \quad (7)$$

where C_t-the friction force of the soil on one tooth of the gear softener, N/(m·gear);

Q_o-the tension force of the gear softener pressure springs in its equilibrium position, N.

When the gear softener deviates from the equilibrium position by an angle ε downwards (Fig. 2, b)

$$\sum R_d = n_t \left(h_{o'} + 0,5L\varepsilon \right) C_t ; \quad (8)$$

$$\sum R_v = 0,5 n_t b_t L \frac{d\varepsilon}{dt} ; \quad (9)$$

$$\sum R_t = -n_t \Delta R_{t1}(t); \quad (10)$$

$$Q = Q_o - C_n L\alpha , \quad (11)$$

where: b_t- The coefficient of resistance of the soil gear softener to one gear, Ns/($m \cdot$ gear);

ΔR_{t1}-R_{t1} variable constituent of power, N;

C_n- gear softener pressure spring total stiffness, N/m.

We substitute the values $\sum R_d$, $\sum R_v$, $\sum R_t$, and Q in expressions (7)-(8) into (4) and obtain the following result

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$$J\frac{d^2\varepsilon}{dt^2}=mgL+2(Q_0-C_nL\alpha)L-(R_{tq})(2l\ -h+L\varepsilon)-$$
$$-\left[n_t\left(h_{o'}+0,5L\varepsilon\right)C_t+0,5n_tb_tL\frac{d\varepsilon}{dt}-n_t\Delta R_{t1}\left(t\right)\right]L. \qquad (12)$$

We can rewrite this expression as follows:

$$2J\frac{d^2\varepsilon}{dt^2}+\frac{1}{2}n_tb_tL^2\frac{d\varepsilon}{dt}+\left(2C_nL^2+R_{tq}L+\frac{1}{2}n_tC_tL^2\right)\varepsilon=$$
$$=mgL-n_th_{o'}LC_t-R_{tq}\left(2l-n\right)+2Q_0L+n_t\Delta R_{t1}\left(t\right)L \qquad (13)$$

or

$$J\frac{d^2\varepsilon}{dt^2}+\frac{1}{4}n_tb_tL^2\frac{d\varepsilon}{dt}+\left(qC_nL\ +\frac{1}{2}R_{tq}L+\frac{1}{4}n_tC_tL\ \right)L\varepsilon=$$
$$=\frac{1}{2}\left(mgL-n_th_{o'}LC_t-R_{tq}\left(2l-h\right)+2Q_0L+n_t\Delta R_{t1}\left(t\right)L\right). \qquad (14)$$

Since the force R_{tq}, i.e. the resistance of the gear softener to traction, is variable, expression (14) represents parametric vibrations [3]. However, since the damping capacity of the soil is large, the parametric vibrations of the gear softener are not observed in this process and it performs forced oscillations relative to the hinge "*O*" under the influence of the force $n\Delta R_{t1}(t)$ [3,4]. Based on the above, we assume that the force R_{tq} is constant and equal to its average value, i.e. $R_{tq}=R_{tq}^{o'}$. Taking this into account, equation (14) takes the following form

$$J\frac{d^2\varepsilon}{dt^2}+\frac{1}{4}n_tb_tL^2\frac{d\varepsilon}{dt}+\left(C_nL\ +\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_tC_tL\ \right)L\varepsilon=$$
$$=\frac{1}{2}\left(mgL-n_th_{o'}LC_t-R_{tq}^{o'}\left(2l-h\right)+2Q_0L+n_t\Delta R_{t1}\left(t\right)L\right). \qquad (15)$$

when the gear softener is in static equilibrium

$$mgL-n_th_{o'}LC_t-2R_{tq}^{o'}\left(2l-h\right)+2Q_0L=0. \qquad (16)$$

Taking this into account, equation (15) takes the following form:

$$J\frac{d^2\varepsilon}{dt^2}+\frac{1}{4}n_tb_tL^2\frac{d\varepsilon}{dt}+\left(C_nL\ +\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_tC_tL\ \right)L\varepsilon=n_t\Delta R_{t1}\left(t\right)L. \qquad (17)$$

To solve this equation, we assume that the force $\Delta R_{t1}(t)$ varies according to the following harmonic law [4]

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$$\Delta R_t(t)=\sum_{n=1}^{n_i}\Delta R_t^n \cos n\omega, \qquad (18)$$

here ΔR_t^n - the amplitude of the harmonics of the alternating force acting on one tooth, N;

n-1, 2, 3, … n_i number of harmonics;

ω – $\Delta R_t^n(t)$ power frequency, s^{-1}.

Taking into account (18), (17) takes the following form

$$J\frac{d^2\varepsilon}{dt^2}+\frac{1}{4}n_t b_t L^2\frac{d\varepsilon}{dt}+\left(C_n L+\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_t C_t L\right)L\varepsilon=n_t\left(\sum_{n=1}^{n_i}\Delta R_t^n\cos n\omega t\right)L$$

(20)

We can reduce this expression to its normal form. To do this, we divide both sides by J and introduce the notations $\frac{1}{4J}n_t b_t L^2=2H$ and $\frac{1}{J}\left(C_n L+\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_t c_t L\right)L=K^2$ as a result, we obtain:

$$\frac{d^2\varepsilon}{dt^2}+2H\frac{d\varepsilon}{dt}+K^2\varepsilon=\frac{n_t}{J}\left(\sum_{n=1}^{n_i}\Delta R_t^n\cos n\omega t\right)L\cdot \qquad (21)$$

The solution of this equation, which represents the forced oscillations of a gear softener, has the following form [6]

$$\varepsilon(t)=\frac{n_t\left(\sum_{n=1}^{n_i}\Delta R_t^n\cos(n\omega t+\delta)\right)L}{J\sqrt{\left[K^2-(n\omega)^2\right]^2+4H^2(n\omega)^2}}, \qquad (22)$$

where

$$\varepsilon=arctg\frac{n_t b_t L^2(n\omega)}{4\left[\left(C_n L+\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_t C_t L\right)L-J(n\omega)^2\right]}. \qquad (23)$$

Taking into account the above definitions, expression (22) takes the following form

$$\varepsilon(t)=\frac{n_t\left(\sum_{n=1}^{n_i}\Delta R_t^n\cos(n\omega t+\delta)\right)L}{\sqrt{\left[\left(C_n L+\frac{1}{2}R_{tq}^{o'}+\frac{1}{4}n_t C_t L\right)L-J(n\omega)^2\right]^2+\left(\frac{n_t b_t L^2}{4}\right)^2(n\omega)^2}}. \qquad (24)$$

According to this expression, the maximum angle of deviation of the gear reducer from the equilibrium position is

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$$\varepsilon_{\max} = \frac{n_t \left(\sum_{n=1}^{n_i} \Delta R_t^n \right) L}{\sqrt{\left[\left(C_n L + \frac{1}{2} R_{tq}^{o'} + \frac{1}{4} n_t C_t L \right) L - J (n\omega)^2 \right]^2 + \left(\frac{n_t b_t L^2}{4} \right)^2 (n\omega)^2}}. \qquad (25)$$

Based on this value of $\varepsilon_{\max}$, we determine the maximum deviation of the working depth of the gear reducer from the specified value $\Delta h_{\max}$.

$$\Delta h_{\max} = \pm \frac{1}{2} L \sin \varepsilon_{\max} = \pm \frac{1}{2} L \left\{ \sin \frac{180}{\pi} \cdot n_t \left(\sum_{n=1}^{n_i} \Delta R_t^n \right) L \times \right.$$

$$\left. \times \left[\left(C_n L + \frac{1}{2} R_{tq}^{o'} + \frac{1}{4} n_t C_t L \right) L - J (n\omega)^2 \right]^2 + \left(\frac{n_t b_t L^2}{4} \right)^2 (n\omega)^2 \right\}^{-\frac{1}{2}} . \qquad (26)$$

According to current agrotechnical requirements, for uniform processing of cotton rows at the required level, the value of $\Delta h_{\max}$ should be less than ±1 cm [4]. Analysis of expression (25) shows that this is achieved mainly by correctly selecting the stiffness of the toothed softener pressure spring.

The research results indicate that the stable motion of the toothed loosener in terms of tillage depth is mainly determined by the variability of the soil's physical and mechanical properties, as well as by the external forces acting on it. Due to variations in soil moisture, density, hardness, and field surface irregularities, the toothed loosener performs not only translational motion but also angular oscillatory motion during operation. These oscillations lead to variations in the tillage depth, thereby reducing its stability.

Based on the conducted theoretical analysis, the angular oscillations of the toothed loosener were described by a differential equation, and the regularities of its forced oscillatory motion were established. The calculations showed that it is possible to determine the maximum deviation angle from the equilibrium position and the corresponding maximum deviation of the tillage depth.

Furthermore, the results showed that due to the high damping properties of the soil, parametric oscillations do not manifest significantly in practice, and the system mainly operates in a forced oscillation regime. To ensure that the tillage depth meets agrotechnical requirements and remains uniform, its maximum deviation must remain within a specified limit.

In general, it was determined that one of the key factors in ensuring the stable operation of the toothed loosener is the proper selection of the stiffness of the pressure springs. This makes it possible to reduce the amplitude of angular oscillations and ensure the stability of the tillage depth.

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