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Pendulum Vibrations of the Ripper Charge Mechanism of the Wide-Cut Harrow-Arbor Aggregate

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ABSTRACT: The article provides brief information about the design of the developed mounted wide-grip harrow unit and the results of small angular oscillations of the parallelogram ripper mechanism. It has been established that the angular vibrations of the parallelogram mechanism in the harrow ripper of the harrow unit is described as an oscillatory process with a limited amplitude and to ensure the uniformity of the ripper travel in depth at its equilibrium position, the angle of inclination to the horizon of the longitudinal links of the parallelogram mechanism should be equal to or close to zero.

KEY WORDS: Wide-grip harrowing unit, ripper, parallelogram mechanism, additional grinder angular vibrations, calculated, scheme, method of operational calculus, equilibrium positions.

I. INTRODUCTION

We have developed a hinged wide-coverage harrow-arbor aggregate that, consisting of a central and two side sections connected to it, support wheels and a mechanism for transferring the side sections to the transport position [1].

Rippers in the form of a two-bar harrow are installed on the front beams of all sections by means of parallelogram mechanisms, and harrows are attached to them, in one row of the spike-tooth harrow section (Fig-1).

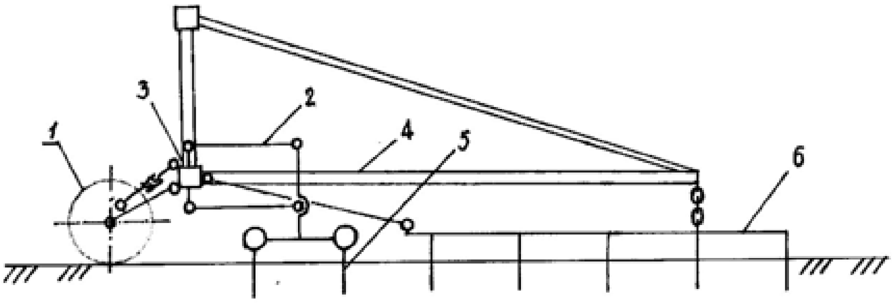

Fig. 1.Diagram of wide-cut harrow agregate.
1-support wheel; 2- parallelogram linkage mechanism; 3-cross beam; 4-frame; 5-ripper; 6- spike-tooth harrow.

This article presents the results of studies of the pendulum vibrations of the ripper of the developed harrow arbor agregate.

Let us consider the pendulum vibrations of a parallelogram mechanism with an additional loader and as a special case without an additional loader, i.e. without a pressure spring. For this, we will draw up a calculated dynamic scheme (Fig. 2) of an equivalent real physical model.

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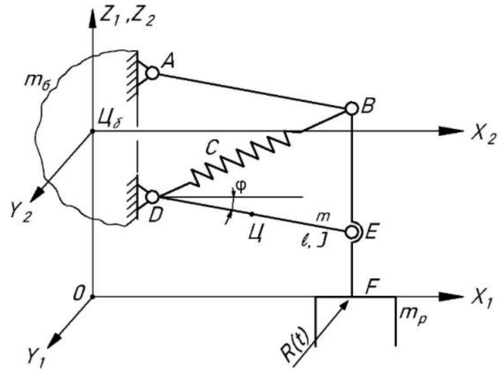

Fig. 2. Calculated dynamic oscillation scheme of the parallelogram ripper mechanism: $C_б$ - center of mass of the hitching bar; C - center of mass of the longitudinal links of the parallelogram mechanism; C- is the stiffness of two parallel springs (additional loaders).

II. SIGNIFICANCE OF THE SYSTEM

The article provides brief information about the design of the developed mounted wide-grip harrow unit and the results of small angular oscillations of the parallelogram ripper mechanism. The study of literature survey is presented in section III, methodology is explained in section IV, section V covers the experimental results of the study, and section VI discusses the future study and conclusion.

III. METHODOLOGY

Suppose that the longitudinal links *AB* and *DE* of the parallelogram mechanism (see Fig. 2) are homogeneous, thin, with mass *m* and length *l*, at points *A* and *D*are hingedly attached to a bar of mass m_b. At points *B* and *E*, these links are pivotally connected to the rod *BF*, to the end *F* of which a ripper of mass m_p is rigidly attached. Friction in joints *A*, *B*, *D* and *E*is neglected due to their smallness. Axles $OX_1Z_1$are fixed, $C_бX_2Z_2$ are movable. The movement of the parallelogram mechanism of the ripper relative to the axes $C_бX_2Z_2$ is a rotation around the axis $Ц_бУ_2$. In this regard, we believe that the parallelogram mechanism of the ripper performs pendulum oscillations relative to the center of mass of the coupling bar $C_б$, which, in turn, performs vertical oscillations (due to the unevenness of the soil surface) according to the law.

$$Z_б = Z_{\max} \cos \gamma t, \qquad (1)$$

Where $Z_{\max}$ - amplitude; $\gamma -$ frequency of vertical vibrations of the coupling bar.

The deviations of the longitudinal links of the parallelogram mechanism from the horizontal γ relative to the $Ц_бX_2$ axis are considered negligible.In the general case, the regularity of the dynamic loading of the soil cultivator is of a random nature due to the unevenness of the cultivated surface of the fields and due to the unequal physical and mechanical properties of the soil. However, taking into account the possibility of representing the law of change in external loads, which is of a random nature, in the form of constant and variable (for $n \to \infty$, where *n* is the harmonic number) components, we assume that during operation an external load acts on the cultivator from the side of the cultivated soil, the projection on the vertical of which follows the law

$$R^z(t) = \sum_{n=0,1}^{n_1} R_n^z \cos n \ \omega \ t, \qquad (2)$$

Where R_0^z and R_n^z - constant and variable (with *n* harmonic) components of external loads;

n=1, …,n_1– No. of harmony; $\omega -$ angular velocity of external forces.

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IV. EXPERIMENTAL RESULTS

Drawing up the equation of motion. Taking the angle of deviation of the longitudinal links of the parallelogram mechanism from the horizontal as the generalized coordinate of the system, we describe the kinetic and potential energies and generalized forces.

The kinetic energy of the system, consisting of the sum of the kinetic energies of the coupling bar vibrating in the vertical plane at a speed $\dot{Z}_б$, and the longitudinal links of the parallelogram mechanism and the ripper, performing plane motion, has the form [1-4].

$$T = \frac{1}{2} m_0 \dot{Z}_б^2 - m l \dot{Z}_б \dot{\alpha} \cos\alpha + J\dot{\alpha}^2 + \frac{1}{2} m_p l^2 \dot{\alpha}^2 - m_p l^2 \dot{\alpha} \dot{Z}_б \cos\alpha, \qquad (3)$$

Where $m_0 = m_б + 2m + m_p$, J – moments of inertia of masses of longitudinal links parallelogram mechanism relative to the hinges A, D.

The potential energy of the system is defined as the sum of the potential energies of all moving links of the parallelogram mechanism in the field of gravity.

$$П = -(m + m_p) g\, l\alpha + m_0\, g\, z_б + \frac{1}{2} c (f - l\alpha)^2, \quad (4)$$

Where c is the stiffness of the spring; f – preliminate tension of the spring.

The generalized force, which is a given function of time, is

$$Q^z(t) = -R^z(t) l. \qquad (5)$$

The Lagrange equation for the considered oscillatory system has the form [4]

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\alpha}}\right) - \frac{\partial}{\partial \alpha}(T - П) = -Q^z(t). \quad (6)$$

Let us calculate the corresponding quantities

$$\frac{\partial T}{\partial \dot{\alpha}} = 2J\dot{\alpha} - m l \dot{z}_б \cos\alpha + m_p l^2 \dot{\alpha} - m_p l \dot{z}_б \cos\alpha,$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\alpha}}\right) = 2J\ddot{\alpha} - m l \ddot{z}_б \cos\alpha + m l \dot{z}_б \dot{\alpha} \sin\alpha + m_p l^2 \ddot{\alpha} - m_p l \ddot{z}_б \cos\alpha + m_p l \dot{z}_б \dot{\alpha} \sin\alpha;$$

$$\frac{\partial T}{\partial \alpha} = m l \dot{z}_б \dot{\alpha} \sin\alpha + m_p l \dot{z}_б \dot{\alpha} \sin\alpha,$$

$$\frac{\partial П}{\partial \alpha} = -(m + m_p) g\, l + lc\,(f - l\alpha) l.$$

After substituting their values in (6) and obvious simplifications, as well as taking into account expressions (1) and (2), the differential equations of motion of the system can be represented in the form

$$\ddot{\alpha} + K^2 \alpha = A - B \cos\upsilon t - \frac{l}{M} \sum_{n=1}^{n_1} R_n^z \cos n\omega t, \qquad (7)$$

Where $K^2 = \dfrac{cl^2}{M}$ - the square of the natural frequency of the system

$$A = \frac{1}{M}[(m + m_p) g\, l + c\, f\, l - R_0^z\, l],$$

$$M = 2J + m_p l^2, \qquad B = \frac{m + m_p}{M} l\, z_{max} \upsilon^2.$$

Since in an equilibrium position, i.e. at $\alpha = 0$,

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$$(m+m_p)gl+cfl-R_0^z l=0$$

equation (7) takes the form

$$\ddot{\alpha}+K^2\alpha=-B\cos\upsilon t-\frac{l}{M}\sum_{n=1}^{n_1} R_n^z\cos n\omega t. \tag{8}$$

Thus, the pendulum oscillations of the parallelogram mechanism of a ripper with a supplementary loader are described by a linear inhomogeneous differential equation of the second order.

The solution of the equation of small angular vibrations of the mechanism of the linkage of the soil cultivator with the additional loader. Let us find the solution of equation (8) satisfying the initial conditions

$$\alpha(0)=\alpha_0, \qquad \dot{\alpha}(0)=\dot{\alpha}_0=0,$$

Where α_0 - initial deviation of the longitudinal links of the parallelogram ripper mechanism from the horizontal.

To solve equation (8), we use the method of operational calculus using the Laplace transform [1-4].

Introducing the notation

$$\frac{1}{P}=1, \qquad X(C)=\alpha(t), \qquad \frac{P}{P^2+\gamma^2}=\cos\gamma t,$$

$$\frac{P}{P^2+(n\omega)^2}=\cos n.\omega t$$

and using the theorem on the image of derivatives, we write

$$\ddot{\alpha}(t)=P^2X(P)-P\alpha(0),$$

Where $\alpha(0)=\alpha_0$ at $t=0$

Then according to (8), representing the equations will take the form

$$P^2X(P)-P\alpha_0+K^2X(P)=-B\frac{P}{\rho^2+\gamma^2}-\frac{l}{M}\sum_{n=1}^{n_1} R_n^z\frac{1}{P^2+(n\omega)^2},$$

Hence the image of the required function

$$X(P)=\frac{P}{P^2+K^2}\alpha_0-\frac{1}{P^2+K^2}\cdot\frac{P}{P^2+\gamma^2}B-\frac{l}{M}\sum_{n=1}^{n_1} R_n^z\frac{2}{P^2+K^2}\cdot\frac{P}{P^2+(n\omega)^2}. \tag{9}$$

To find the original of the depicting equation (9), we use the tabular data and the method of expanding the fraction into simple fractions [1-4].

If we use tabular data, then we write

$$\frac{P}{P^2+K^2}=\cos Kt.$$

To find the originals of the second and third terms of the representing equation (9), we use the method of expanding the fraction into simple fractions, then

$$\frac{P}{(P^2+K^2)(P^2+\gamma^2)}=\frac{aP+b}{P^2+K^2}+\frac{c\rho+d}{\rho^2+K^2}, \tag{10}$$

Where *a*, *в*,*c*and *d* – constant coefficients to be determined.

Based on the obtained equality.(10) can be written

$$(a\rho+b)(\rho^2+\gamma^2)+(c\rho+d)(\rho^2+K^2)=\rho.$$

Equating in both sides of this identity the coefficients at the same degrees of *P*, we obtain a system of linear algebraic equations for the unknown coefficients *a*, *b*, *c* and *d*:

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$$P^3 : a + c = 0;$$
$$P^2 : в + d = 0;$$
$$P : a\gamma^2 + ck^2 = 1;$$
$$P^0 : b\gamma^2 + dk^2 = 0.$$

Having solved this system, we find

$$c = -a; \qquad d = -b; \qquad a = \frac{1}{\gamma^2 - \kappa^2}; \qquad b = 0; \qquad d = 0.$$

We will finally record

$$\frac{P}{(P^2+K^2)(P^2+\gamma^2)} = \frac{1}{\gamma^2 - K^2}\cdot\frac{P}{P^2+K^2} - \frac{1}{\gamma^2-K^2}\cdot\frac{P}{P^2+\gamma^2} =$$
$$= \frac{1}{\gamma^2 - K^2}(\cos Kt - \cos\gamma t).$$

Similarly to the one obtained, you can record

$$\frac{P}{(P^2+K^2)(P^2(n\omega)^2)} = \frac{1}{(n\omega)^2 - K^2}(\cos kt - \cos n\omega t)$$

Taking into account the results of the performed transformations, passing to the original, we finally get

$$\alpha(t) = \alpha_0 \cos Kt - \frac{B}{\gamma^2 - K^2}(\cos Kt - \cos\gamma t) -$$
$$- \frac{l}{M}\sum_{n=1}^{n_1} R_n^z \frac{1}{(n\omega)^2 - K^2}(\cos Kt - \cos n\omega t). \tag{11}$$

Analyzing the result obtained, it can be noted that the pendulum oscillations of the ripper hinge mechanism are described as an oscillatory process with a limited amplitude. To study the amplitude, we transform (11) in the form

$$\alpha(t) = D\cos Kt + E\cos \upsilon t + F\cos n\omega t, \tag{12}$$

where

$$D = \alpha - E - F, \qquad E = \frac{B}{\gamma^2 - K^2}; \qquad F = \frac{l}{M}\sum_{n=1}^{n_1} R_n^z \frac{1}{(n\omega)^2 - K^2}.$$

The maximum value of the amplitude $\alpha(t)$ is equal based on module

$$|D| + |E| + |F| = C. \tag{13}$$

With this in mind, the value $\alpha(t)$ fluctuates within

$$|D| + |E| + |F| \geq \alpha(t) \geq -|D| - |E| - |F|. \tag{14}$$

Differentiating expression (12) twice, we have

$$\ddot{\alpha}(t) = -K^2 D\cos Kt + \gamma^2 E\cos\gamma t - (n\omega)^2 F\cos n\omega t, \tag{15}$$

hence the maximum value of $\ddot{\alpha}$ is

$$|D|K^2 + |E|\gamma^2 + |F|(n\omega)^2 = N. \tag{16}$$

We pass to the consideration of the pendulum oscillations of the parallelogram mechanism of the ripper without an additional loader, i.e. if there is no spring, then the equation of motion of the oscillatory system (8) at c=0 takes the form

$$\ddot{\alpha}(t) = -B\cos\gamma t - \frac{l}{M}\sum_{n=1}^{n_1} R_n^z \cos n\omega t \tag{17}$$

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Integrating equation (17), we obtain

$$\ddot{\alpha}(t) = -\frac{B}{\gamma}\sin \gamma t - \frac{l}{M}\sum_{n=1}^{n_1} R_n^z \frac{1}{n\omega}\sin n\omega t + D_1. \tag{18}$$

Remembering the initial angles in the form $\alpha(0) = 0 \; npu \;\; t = 0,$, we get

$$D_1 = 0$$

After integrating (18), we record

$$\alpha(t) = \frac{B}{\gamma^2}\cos \gamma t + \frac{l}{M}\sum_{n=1}^{n_1} R_n^z \frac{1}{n\omega}\cos n\omega t + D_2 \tag{19}$$

Writing the initial conditions in the form $\alpha(0) = \alpha_2 \;\; at \;\; t = 0$, we obtain

$$D_2 = \alpha_0 - \frac{l}{M}\sum_{n=1}^{n_1} \frac{R_n^z}{(n\omega)^2} - \frac{B}{\gamma^2}.$$

Substituting the values of the constant integration D_2 in (19), we finally write down the equation for describing the oscillatory motion of the ripper linkage mechanism without an additional loader about the equilibrium position with an arbitrary choice of the origin (center of mass) of the hitch bar in the form

$$\alpha(t) = \alpha_0 - \frac{2B}{\gamma^2}\sin^2\frac{\gamma t}{2} - \frac{2l}{M}\sum_{n=1}^{n_1} \frac{R_n^z}{(n\omega)^2}\sin^2\frac{n\omega}{2}t. \tag{20}$$

It can be noted that the movement of the ripper mechanism without an additional loader is described in the form of an oscillatory process relative to the initial deviation of the longitudinal links of the parallelogram mechanism.

In order to analyze the results obtained, on the basis of formulas (13) and (16), graphs of changes in the maximum amplitudes $\alpha(t) \;\; u \;\; \ddot{\alpha}(t)$ were constructed (Fig. 3).

As can be seen from Fig. 3, with an increase in the length of the longitudinal links and a decrease in the rigidity of the additional loader, the amplitudes $\alpha(t) \;\; u \;\; \ddot{\alpha}(t)$ decrease, therefore, the stability of the ripper stroke in depth improves.

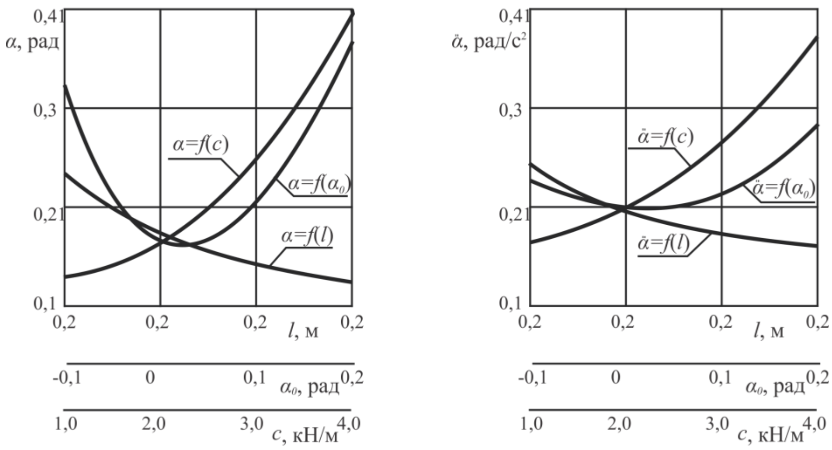

Fig. 3.Dependence of the amplitudes $\alpha(t) \;\; u \;\; \ddot{\alpha}(t)$ on the initial deflection angle of the longitudinal links of the parallelogram mechanism, their length and stiffness of the additional loader.

Depending on α_0 amplitudes $\alpha(t) \;\; and \;\; \ddot{\alpha}(t)$, they change according to the parabola law, while the smallest values of the amplitude are obtained at $\alpha_0 = 0$.

VI. CONCLUSION AND FUTURE WORK

Therefore, when the ripper is in equilibrium, the angle of inclination to the horizontal of the longitudinal links should be equal to or close to zero.

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REFERENCES

[1]. Lavrentyev M.A., Shabat B.V. Methods of the theory of functions of a complex variable. M. Nauka. chief editor of physics and mathematics lit. 1987. pp. 688.
[2]. Jerevneev V.F., Kalnitsky L.A., Sapogov N.A. Special course in mathematics for universities.M. Higher School.1970. pp. 416.
[3]. Bateman G., Erdein A. Tables of integral transformations.I. 2 V. M. Nauka, editor-in-chief of physics and mathematics literature.1968. Vol. 1. Fourier, Laplace, Melinna transforms. 343.
[4]. Shakhno K.U. Elements of the theory of functions of a complex variable and operational calculus.- Minsk: Higher School. - 1970 .-pp. 416.