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Dynamic Analysis of the Motion of a Vibrating Mechanical System

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Abstract: This article focuses on vibrating the working bodies of soil tillage machines, and methods of increasing energy and resource efficiency and reducing traction resistance by vibrating their working bodies are researched.

Keywords: Tillage, vibration, soil density, energy, vibration, power, mass, forced vibration, speed, process, traction resistance, vibration technology, vibrating mechanical system, damping.

Introduction

Today, as a result of the constant increase in the price of energy resources, the issue of saving them has become urgent all over the world. Many ways to solve such problems are being developed, one of these ways is to reduce energy consumption during the operation of agricultural machinery. This process is one of the most energy intensive in agriculture [1].

Currently, one of the most optimal ways to reduce labor and energy consumption is to use resource-saving technologies using improved machines, to increase the productivity of agricultural machines by optimizing the structure and operation parameters of the used equipment [2].

Experts pay special attention to the use of vibration in tillage, which is the most energy-intensive process of modern agricultural production. At the same time, there is currently a tendency to use comprehensive combined tillage tools. In this case, the process of using vibration in soil tillage machines to reduce traction resistance is not fully studied and requires new scientific research [3].

P. I. The tillage unit with a vibrating working body proposed by Butanavichyus consists of a column connected to the frame with a finger joint, a vibrator with a ball attached to the frame, and a spring in tension. The main disadvantage of this unit is that the ball vibrator requires a lot of power to vibrate the working body, the normal pressure increases due to dry friction in the part of the ball and the column in the support connection, and negative forces, i.e. impact forces, are formed [4].

Research method

The unique feature of our offered vibratory chisel-cultivator design is that it consists of two frames, the first frame is connected to the tractor. The second one is hinged to the first frame. This makes them mutually excitable (Fig. 1).

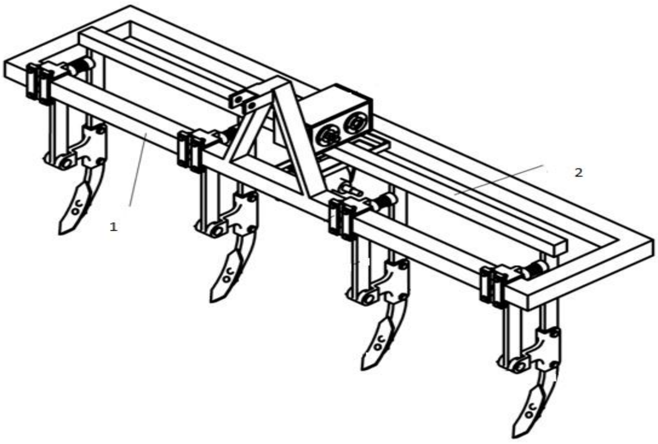


Figure 1. Construction scheme of the vibratory cultivator.

1- frame is fixed; 2- frame drive.

The working bodies and the vibrostimulator are fixedly mounted on the second frame. During operation, it can be assumed that the first frame does not move relative to the tractor, and the second frame is movable relative to them and oscillates.

First, we determine the forces acting on the characteristic links and points of the structure during operation.

They are as follows: $\boldsymbol{F_t}$ - traction force, i.e. driving force, exerted on the aggregate by the tractor; $\boldsymbol{F_b}$ is the periodic excitation force (vibration force) that vibrates the second frame by the vibrostimulator; this force is directed perpendicular to the rack of the working body; elasticity $\boldsymbol{F_e}$ force resulting from the deformation of the spring connecting two frames, i.e. restoring force; resistance force $\boldsymbol{F_q}$ acting on the working body and directed perpendicular to it, this force consists of the sum of several forces; *$\boldsymbol{G}$ is the weight of the cultivator and the internal forces $\boldsymbol{F_r}$ in the links, that is, the reaction forces that appear due to dynamic loads.*

Now let's talk about the forces listed above.

The tractor's maximum pulling force $\boldsymbol{F_t}$ is given in its technical characteristics. This force may vary depending on the condition of the tractor tires and the environment in which they move. The amount of this force to pull the aggregate (cultivator) can be theoretically approximately determined. It is also determined experimentally, and the theoretically found value is determined based on it. The drag force $\boldsymbol{F_q}$ determines the amount of force required to drag the aggregate during operation.

A vibrostimulator installed on the second frame, i.e., a mechanical vibrator, gives vibration to the working bodies. Since the vibrator consists of two unbalanced shafts, the vibration force $\boldsymbol{F_b}$ acting in a specific direction is obtained. This force can be given to the second frame in any direction. To use it effectively, it is advisable to direct it perpendicular to the rack of the working body. The amount of vibrational force is determined by the resistance of the environment. Mechanical and kinematic characteristics of the vibrator are determined depending on the amount of resistance. Due to the influence of the vibrator and the soil, the second frame rotates around the hinge relative to the first frame, and the springs connecting them are stretched or stretched, and the elastic force $\boldsymbol{F_e}$ appears. This power is restorative power.

When the working bodies move, it is affected by the resistance force $\boldsymbol{F_q}$ from the environment. This force is complex and depends on the coefficient of friction, the force of gravity, the resistance force depending on the shape of the working body and the speed of the unit. The force of resistance is opposite to the direction of motion.

The gravity of the cultivator. It depends on two frames, a vibrator and a mass of working bodies.

During the operation of our unit, reaction forces $\boldsymbol{F_r}$ occur in the hinges connecting the two frames due to dynamic loads and shocks. These reaction forces have a negative effect on the reliable operation of the hinged connections. Therefore, it is necessary to reduce the amount of reaction forces as much as possible.

First, let's put the active forces acting on it in the constructive scheme of the cultivator in their direction (Fig. 2).

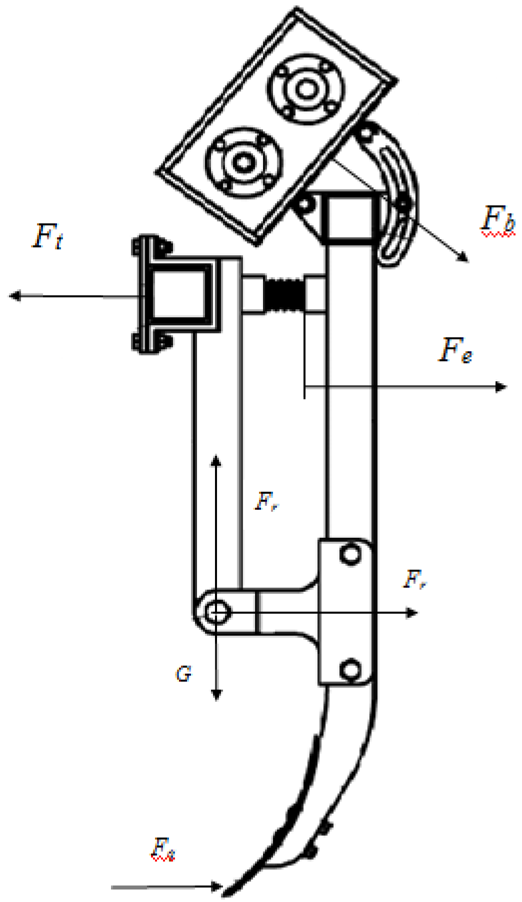


Figure 2. Forces Acting on the Cultivator.

It can be seen from the construction scheme of the vibratory cultivator that the working bodies and the second frame on which the vibrator is installed are connected to the first frame through hinges and springs, so they can vibrate each other, that is, they form a vibrating mechanical system.

In order to study the vibration behavior of the working bodies in the soil, we first study their free vibrations when the vibrator is not started (Fig. 3).

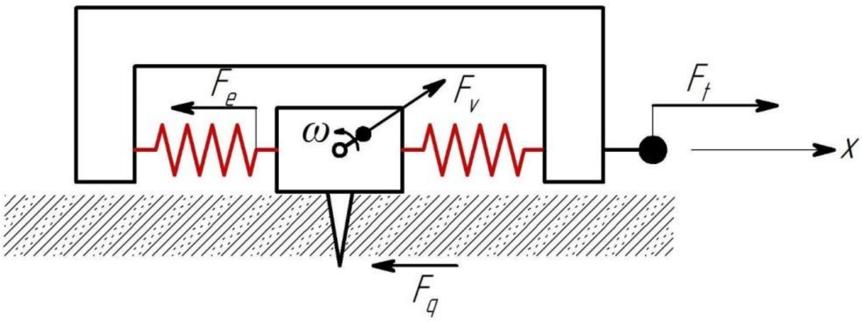
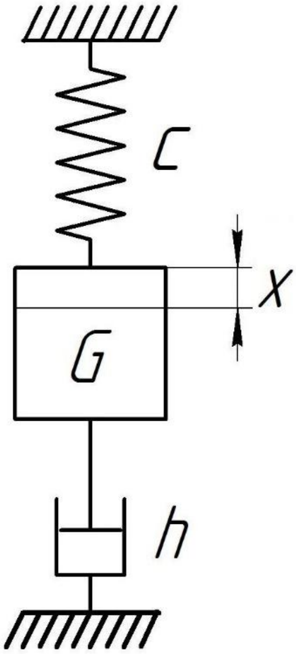


Figure 3. Calculation scheme for an oscillating mechanical system.

Dissipative forces, or in other words damping forces, must be taken into account in order to study the vibrations of the cultivator working body in the soil in a situation close to reality. In our case, the constituents of this force include friction forces (between the soil and the working body), resistance forces, and soil deformation forces. The damping force proportional to speed, which causes energy to be dissipated or absorbed, is called viscous damping. It is mathematically very convenient for us to replace the resistance force that exists in the working process of the cultivator with the viscous damping force. In this case, viscous damping is defined from the condition that it absorbs energy equal to the energy dissipated by the actual resistive forces during the cycle.

Research result and discussion

Let's study the free vibrations of the oscillating mechanical system in the state of impact of the soil on the second frame of the cultivator using the classical model (Fig. 4).



4-расм.

The damping equation of the cumulative mass motion in the dynamic model is as follows:

$$\frac{G}{g}\ddot{x} = G - (G + cx) - h\dot{x} \qquad (1)$$

1. Here h is the viscous damping coefficient or damping constant, which is the force per unit velocity. A minus sign in front of it means that the damping force is always in the opposite direction to the speed.
2. G is the gravity of the aggregated mass, c is the stiffness of the joint.

dividing the left and right sides of the equation by G/g and

$$p^2 = \frac{cg}{G};\ 2n = \frac{hg}{G}, \qquad (2)$$

introducing notations, we obtain the equation of free vibrations in the presence of viscous damping

$$\ddot{x} + 2n\dot{x} + p^2 = 0 \qquad (3)$$

This equation is a second-order linear differential equation with constant coefficients. Its general solution is given in many literatures.

$$x = e^{-nt}\left(C_1 \cos p_g\, t + C_2 \sin p_g\, t\right), \qquad (4)$$

where C_1 and C_2 are constants determined from initial conditions.

e^{-nt} the multiplier decreases over time. Due to this, the vibration also fades. The rotational frequency and corresponding period of damped oscillations in the presence of viscous resistance are determined from the following expressions

$$p_g = \sqrt{p^2 + n^2},\ \tau_g = \frac{2\pi}{p_g} = \frac{2\pi}{p}\,\frac{1}{\sqrt{1-(\frac{n^2}{p^2})}}.$$

$\lambda = \frac{n}{p}$ – is the damping coefficient, which is important not only in our case, but also in the case of vibration protection.

To find the constants C_1 and C_2 in equation (4), we assume that at the beginning of the movement, that is, at $t=0$, the body (the second frame) has moved from the equilibrium position to a distance X_0 and has a speed of $(\dot{X_0})$. Using these conditions, they are determined and put into equation (4), as a result, equation (4) becomes

$$X = e^{-nt}\left(X_0 \cos p_g\, t + \frac{\dot{X}_0 + nX_0}{p_g} \sin p_g\, t\right). \qquad (5)$$

Or equivalently

$$X = Ae^{-nt}(\cos p_g\, t - \alpha_g) \qquad (6)$$

here $A = \sqrt{{C_1}^2 + {C_2}^2} = \sqrt{{a_0}^2 + ({x_0}^2 + nx_0)^2/p_g^2}$

$$\alpha_g = arctg\frac{C_2}{C_1} = arctg\left(\frac{\dot{X}_0 + nX_0}{pX_0}\right).$$

Our above considerations are valid for the case where $n<p$. If $n>p$, the solution of our equation (3) will be in this form.

$$X = c_1 e^{r_1 t} + c_2 e^{r_2 t} \qquad (7)$$

$t = 0$ да $x = x_0,\ \dot{x} = \dot{x_0}$ If we find the constants using the initial condition, equation (7) becomes this form

$$X = \frac{x_0 - \dot{r}_0 x_0}{r_1 - r_2} e^{r_1 t}\, \frac{r_1 x_0 - \dot{X}_0}{r_1 - r_2} e^{r_1 t} \qquad (8)$$

It can be seen that the solution is not periodic and does not represent oscillatory motion. Bump resistance is so great that after the body is removed from the equilibrium position, it cannot oscillate and slowly moves towards the center of equilibrium. We know that the mechanical system in this state is overdamped, and its motion is called non-periodic.

A special case exists between the underdamped and overdamped cases. In this case, $n=p$, and the oscillating system begins to lose its oscillating character from this moment. A critical value of the viscous damping coefficient is included for this case

$$h_{kp} = 2n\frac{G}{g} = -2p\frac{G}{g} = 2\sqrt{\frac{CG}{g}}.$$

$n = p$ the general solution for the case is as follows

$$X = e^{-pt}[X_0 + (\dot{X}_0 + nX_0)t]. \quad (9)$$

We assume that the value of n, which depends on the damping constant h, is always positive. Because it represents the power of resistance. Due to this force, energy is dissipated, the vibration amplitude decreases and the motion fades. A positive n (damping oscillation) refers to stable motion.

Summary.

The studies mentioned above show that effective results are achieved by applying forceful vibration to the hooves. This, in turn, improves the quality of tillage and reduces the traction resistance of the aggregate, as well as ensures the fulfillment of the specified agrotechnical requirements. Based on the above considerations, the following conclusions can be reached: If we assume that the first frame is stationary, the second frame moves in a rotational motion relative to it. All working bodies vibrate in parallel planes. The specific vibration frequency of the system depends on the moment of inertia of the second frame relative to the hinge and the coefficients of uniformity of the springs.

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