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Dynamic models of cultivator spring tine performance

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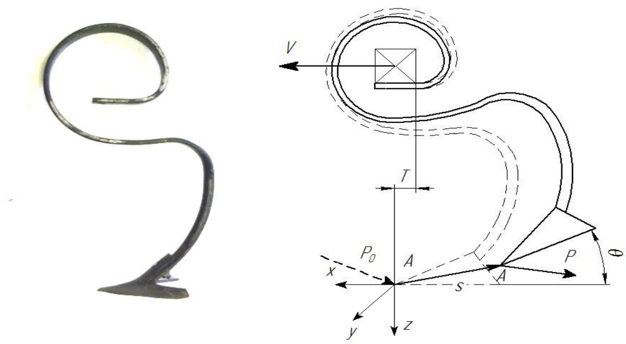
Abstract. The article presents results of spring-tine cultivator blade movement modeling. The basic equation of terraelasticity is taken as the initial one. The new model takes into account the spring tine configuration and the vibration mode. It is noted that the model becomes multifrequent. The field reserach confirmed multifrequency and mixed nature of the spring tine vibration spectrum. It is found that the frequency spectrum with technologically relevant vibration accelerations can be more than 1000 Hz. The authors make a conclusion that it is necessary to use high frequencies to improve the quality of spring-tine cultivator blade performance.

1 Introduction

Fastening of the blades to the frame of modern cultivators is usually made with spring tines (Fig. 1), which usually get P load deformation of the working parts with spatial elastic displacements S of the blade edge and rotation by an angle θ [3]. This distorts the depth of blade motion and the cutting geometry, as well as decrease the quality of performance.



a) b)

Fig. 1 Spring tine (a) and its design scheme (b):
A – blade edge; *T* – reference of the blade edge to the vertical; *X*, *Y*, *Z* – coordinate axes; P_0 and *P* – the initial and current loads respectively; *V* – portable speed of the cultivator frame.

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However, the springs have certain advantage compensating for these shortcomings. Long-term study of the spring tines performance demonstrated that, due to deformations they receive significant vibrations improving technological process: reducing stickiness and energy consumption, and improving the quality of soil cultivation [7-8]. Optimization of the spring tines oscillating processes is of great importance for soil cultivation so development of spring tine performance dynamic models is of great priority.

2 Research objective

The general theory of terra-elasticity [4-6] considers movement of the working part in soil to be an interaction of two subsystems: the technological process of working part and soil interaction and the subsystem of working part elastic fixing. The general model of terraelasticity [5] has a number of advantages: it is compact, universal, has a fundamental nature; being expressed in the form usual for analytical mechanics. However, there are few specifics in it, it does not take into account that the soil is a continuous medium and the elastic fixing is a mechanical system with distributed parameters. Models of such subsystems should be built on a microlevel [1], [9].

Complexity and labor intensivity of terraelasticity modeling at the micro level requires analytical models to be more convenient for engineering practice. Currently such models are those at the macro level.

3 The oretical framework

For a macro level, a system with distributed parameters must be reduced to an equivalent system with lumped parameters [1-2]. We use a typical method of decomposing the forms of rod elastic displacements according to the modes of natural oscillations [10]. Let us take into account that the elastic displacements of the attachment points depend on their position, determined by the vector ζ and the time t:

$$\bar{s} = \bar{s}\,(\zeta, t). \qquad (1)$$

For small elastic displacements, the function (ζ, t) can be expanded in terms of the natural oscillation modes $\xi = \xi\,(\zeta)$:

$$s = \sum_{i=1}^{m} q^*{}_i\,(t)\,\xi_i(\zeta), \qquad (2)$$

where m is the number of the system natural frequencies; $q^*{}_i(t)$ are the so-called normal coordinates characterizing the intensity of the oscillations eigenmodes $\xi_i\,(\zeta)$.

Kinetic energy of the system is

$$T = \frac{1}{2}\int_n \left(\frac{\partial s}{\partial t}\right)^2 dm. \qquad (3)$$

If we substitute here the values from (2) and take into account the eigenfunctions orthogonality, we obtain that the kinetic energy is the sum of the kinetic energies corresponding to each of the normal coordinates:

$$T = \frac{1}{2}\sum_{i=1}^{n} \mu_i \dot{q}_i^{*2} , \qquad (4)$$

where μ_i is the reduced mass of the system for the *i* form of natural oscillations:

$$\mu_i = \int_n \xi_i(\zeta) dm$$

Taking into account the eigenforms orthogonality we see that potential energy of the system, will also be SS of the normal coordinates:

$$П = \frac{1}{2}\sum_{i=1}^{n} c_{ik} q_{ik}^{*2} , \qquad (5)$$

where c_{ik} are coefficients having the rigidity dimension.

Dissipation energy for systems with low dissipation is

$$Д = \frac{1}{2}\sum_{i=1}^{n} \xi_{ik} \dot{q}_{ik}^{*} , \qquad (6)$$

where ξ_{ik} are dissipation coefficients with respect to the coordinates.

The work of external forces f distributed over the surface *S* is

$$U = \int_S f(S) s(П, t) dS \cdot \qquad (7)$$

By assuming the normal coordinates as extended, and substituting expressions for kinetic and potential energy (4) and (5) and dissipation energy (6) into Lagrange equations of the second kind

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_k^*}\right) - \frac{\partial T}{\partial q_k^*} + \frac{\partial U}{\partial q_k^*} = 0, \qquad k = 1, 2, \ldots n \qquad (8)$$

we obtain a terraelasticity macromodel by normal coordinates q^*:

$$[\mathbf{A}]\ddot{q}^* + [\mathbf{B}]\dot{q}^* + [\mathbf{C}]q^* = \mathbf{F}(t), \qquad (9)$$

where q^* is the column vector of extended normal coordinates; $\mathbf{F}(t)$ is the vector of the external influences; [**A**] is the diagonal matrix of the inertia coefficients; [**B**] – symmetric matrix of dissipation coefficients; [**C**] is a symmetric matrix of stiffness coefficients.

All matrixes of the model have the size m * m, corresponding to the number of forms of natural frequencies of oscillations of the system. Since the number of eigenfrequencies of a distributed system is large, the matrices [**A**], [**B**], [**C**] can in principle be of infinite order. As a result, the dynamic model of an elastic system with distributed parameters is replaced by an infinite system of second-order differential equations.

The number of equations (9) can be limited depending on the purpose of the problem. To analyze the vibration effect, about 20 natural frequencies should be taken into account.

However, such a terraelasticity model by the normal coordinates of the vibration modes has a drawback: it operates with amplitudes of its own forms which are difficult to identify and measure.

Model (9) can be improved. We also express the model terraelasticity by blade edge *q* extended coordinates, which are more convenient for measurements.

According to the rules of mechanics, the extended coordinates q are related to the normal coordinates q^* by the similarity transformation

$$\overline{q} = [\mathbf{V}]\ \overline{q}^*, \tag{10}$$

where [**V**] is the similarity matrix of dimension nxm; m is the number of the tine vibration modes considered.

The elements of the similarity matrix are expressed in terms of the values of the proper forms of displacements and oscillations on the blade edge A ($z = A$):

$$\upsilon_{ki} = \xi_{ki}(A), \quad i = 1, \dots m, \quad k = 1, \dots n \tag{11}$$

Then the extended coordinates of the blade are expressed in the series by the forms of natural oscillations ξ_{1k} (ℓ):

$$q_k = \sum_{i=0}^{m} q_k^* \xi_{ik}(A) = q_{k0} + \xi_{k1} q_1^*(t) + \xi_{k2}\, q_2^*(t) + \dots + \xi_{km}\, q_m^*(t), \tag{12}$$

$$k = 1, \dots n$$

Here the subscript 0 is a constant component symbol; $\xi_{k0} = 1$.

According to D'Alembert principle, the equilibrium condition of the forces acting on the working member is: inertial $\mathbf{F}_{in}$, dissipative $\mathbf{F}_{dis}$, elastic $\mathbf{F}_y$ and external $\mathbf{F}$:

$$\mathbf{F}_{in} = -[\mathbf{A}]\ \ddot{\overline{q}}, \quad \mathbf{F}_{dis} = [\mathbf{B}]\ \dot{\overline{q}}, \quad \mathbf{F}_y = [\mathbf{C}]\ \overline{q}, \tag{14}$$

where [**A**], [**B**], [**C**] are square symmetric matrices of coefficients of inertia, dissipation, rigidity of dimension 6 x 6, respectively.

Taking into account relations (14), the equilibrium condition (13) unfolds into a system of differential equations

$$[\mathbf{A}]\ \ddot{\overline{q}} + [\mathbf{B}]\dot{\overline{q}} + [\mathbf{C}]\overline{q} = \mathbf{F}. \tag{15}$$

It externally keeps the structure of equations (9), but unlike them, the matrices [**A**], [**B**] and [**C**] will have a dimension of 6x6.

The deterministic load F is assumed to be a vector function of the system state vector $\mathfrak{I}$ [3]. We get:

$$[\mathbf{A}]\ \ddot{\overline{q}} + [\mathbf{B}]\ \dot{\overline{q}} + [\mathbf{C}]\overline{q} = \mathbf{F}(\mathfrak{I}) \tag{16}$$

The state vector has a design component $\mathfrak{I}_o$ that does not depend on elastic displacements (rigid fixing) and the component U determined by elastic displacements that distort the design value.

Then $\mathfrak{I} = \{\mathfrak{I}_o + \mathbf{U}\}$; where $\mathbf{U} = \{\overline{q}_1, \dot{\overline{q}}_1\}$ (17)

Since the low-frequency oscillations and high-frequency vibrations play different roles when the cultivator blade is operating, it is advisable to represent the summands of the series (12) with three summands:

$$q_k = q_{k0} + \xi_{k1}\, q_1^*(t) + \sum_{i=2}^{m} q_i^* \xi_{ki}(0), \qquad k = 1, \dots n. \tag{18}$$

The first summand characterizes the average (static) elastic displacements of the blade – they are the largest ones. The second summand determines the lowest low-frequency oscillations of the blades relative to the average, having a maximum on the blade edge, $\xi_{11} = 1$, $k = 1, \dots m$.

The summands, beginning with the third one, characterize vibrations – high-frequency oscillations with small amplitudes; for the uniformity of the stroke they are insignificant, but they make the vibroeffect of the elastic fixing.

After this separation of the load into low-frequency and high-frequency and rearrangement, equation (16) takes the form:

$$[\mathbf{A}]\ddot{\bar{q}}_1+[\mathbf{B}]\dot{\bar{q}}_1+[\mathbf{C}]\bar{q}_1 = \mathbf{F_1}\{\mathfrak{I}_1\}+\mathbf{F_2}\{\bar{q}^*_{2-m}\}, \tag{19}$$

where $\mathfrak{I}_1 = \{\mathfrak{I}_o + \mathbf{U_1}\}$.; $\mathbf{U_1}=\{\bar{q}_1, \dot{\bar{q}}_1\}$, $\mathbf{F_1}\{\mathfrak{I}_1\}$ is low-frequency component of the resistance forces, $\mathbf{F_2}\{\bar{q}^*_{2-m}\}=\mathbf{F_2}\{\sum_{i=2}^{m}\bar{q}^*_i\xi_i(A)\}$ is high-frequency component of the resistance forces caused by vibrations.

The vibrations are caused by the influence of the inhomogeneities of the soil on the stand, as centered random noise $f(t)$; $E\{f(t)\} = 0$. By reflecting the excitation of high-frequency vibrations of white noise by the functional $[f(t)]$, we obtain from (19) a macro-level model of terraelasticity by extended coordinates:

$$[\mathbf{A}]\ddot{\bar{q}}_1+[\mathbf{B}]\dot{\bar{q}}_1+[\mathbf{C}]\bar{q}_1= \mathbf{F_1}\{\mathfrak{I}_1\}+\mathbf{F_2}\{\bar{q}^*_{2-m}[\mathbf{f}(\mathrm{t})]\}. \tag{20}$$

This 6-dimensional physical model divides the spectrum of the oscillations into low-frequency and high-frequency parts. In contrast to the structure of the general terraelasticity model, an additional component $\mathbf{F}_2$ $\{[f(t)]$ appears in it, reflecting the physics of high-frequency vibrations. In fact, the spring tine plays performs as a resonator converting random noise into high-frequency vibrations.

Thus, the advanced mathematical model of terraelasticity describes all the features of the spring tines performance in analytical way: multifrequency and mixed low-frequency oscillations and vibrations.

These features have been verified a number of experiments [1]. The S-shaped spring tine of the Kongskilde type (see fig. 1), the most common for cultivators, has been taken as an object of research.

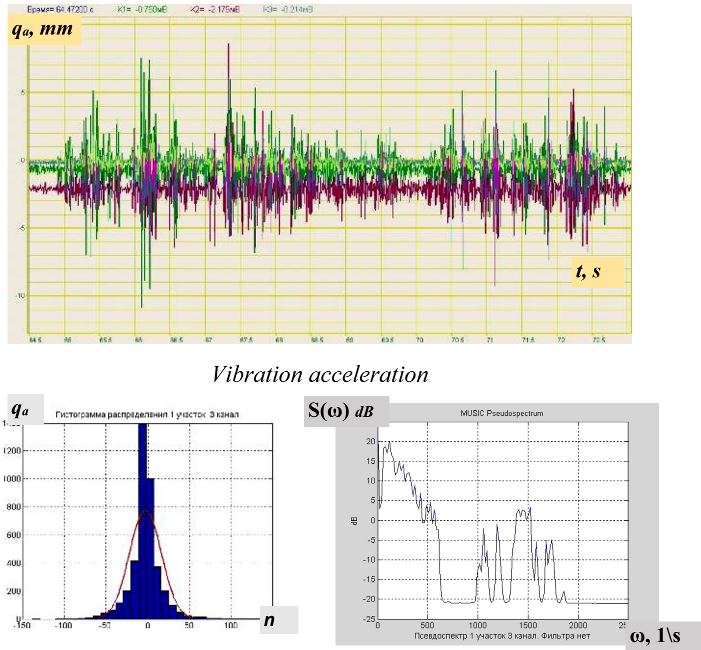
The purpose of experiments was to test the multifrequency, bias and to estimate the spectrum boundaries of the significant vibration frequencies for the spring tine. Vibrations with an intensity of 0.5 g were considered to be significant for vibration acceleration.

To measure the vibration, three piezoceramic vibration acceleration sensors (accelerometers) of the KV-10 type with a small mass and dimensions were mounted under the blade in three coordinate directions.

Signals from the sensors were transmitted through filters with a bandwidth of 0-5000 Hz to the spectrum analyzer A17-T8 of ZETLAB, where they were digitized with a sampling frequency of 25 KHz, processed and sent to a computer.

The spectrum of the spring tine eigenfrequencies turned out to be multifrequent for any distributed system; it began at 10 Hz, and had several frequencies in the range up to 75 Hz (10.3, 17.9, 34.2, 73.8 Hz).

Field studies of the dynamics of elastic spring racks showed greater intensity of dynamic processes. The intensity of vibration acceleration reached 5g, which makes them a significant technological factor. On fig.2 gives as an example the vibration acceleration of the spring tine as well as amplitudes and spectrograms distribution characteristics with an average mode of tine performance: a depth of 12 cm, a speed of 7.2 km / h.



Vibration acceleration

Histogram *Spectrogram*

Fig. 2. Example of spring tine vibration characteristics.

Mixed oscillations of the spring racks are clearly visible during vibration acceleration: low-frequency oscillations are visible, where high-frequency vibrations and white noise are superimposed. Mixed signals have a significant debytion from the normal distribution, that is reflected by the histograms. Spectral analysis of the spectral power density S(ω) for mixed signals masks high frequencies [11], that leads researchers to a delusion about the limited range of oscillations at a maximum of 30 Hz. Therefore, to determine the spectral composition of the signal, pseudo spectra are used with the MUSIC (Multiple Signal Classification) method [5], which makes it possible to estimate the frequencies of the harmonic components of the signal fairly accurately by arrangement of the pseudospectrum curve peaks.

The MUSIC pseudo spectrum detected at least 10 harmonics of vibrations in the 12-600 Hz band for the spring tine. It is a sensation that intense vibrations appear in the 1000- 1800 Hz band.

4 Practical relevance

The high frequency components of the spring tines vibrations make an important reserve for improving the quality of the cultivator spring working parts. To intensify the vibratory processes of the cultivator spring tines, it is necessary to learn how to control them.

5 Conclusion

Advanced terraelasticity models by extended coordinates with separation of frequency bands reflect all the features of spring tine cultivators performance and allow to manage and optimize their vibrational processes.

References

1. V. I. Ignatenko, Bulletin of the DSTU, **9**, 142-150 (2009)
2. V. I. Ignatenko, V. P. Zharov, Scient. ideas of the Cauc., **5**, 215-219 (2006)
3. I. V. Ignatenko, *Energy aspects of the interaction of an elastically fixed working part with soil in agricultural mechanics*: Monograph (DSTU, Rostov-on-Don, 2002)
4. I. V. Ignatenko, Bulletin of the DSTU, **8**, 3(38), 268 (2008)
5. I. V. Ignatenko, Journ. of the Scien. ideas of the Cauc., **14**, 90 (2002)
6. *Terraelasticity. Application for the development of elastic working parts of cultivators*: Monograph (DSTU, Rostov-on-Don, 2011)
7. S. V. Levitsky *Studies of the vibration effect of elastic suspension of the working parts of a high-speed blade cultivator in order to reduce the traction resistance* (Rostov-on-Don, 1981)
8. N. K. Mazitov, R. A. Sakhapov, N. K. Galyautdinov, M. N. Mazitov, Tract. and agricult. mach., **7**, 30 (2007)
9. S.G. Mudarisov, Tract. and agricult. mach. **7**, 13 (2005)
10. Ya. G. Panovko, *Osnovy prikladnoy teorii kolebaniy i udara* (Politekhnika, Leningrad, 1990)
11. I. K. Shakirov, M.N. Mazitov and others, Tract. and agricult. mach., **10**, 29 (2002)
12. M.J. Butson, D.H. Rackham, J. arg. Engg res., **26**, 5, 419 (1981)
13. K. Moller, A. Gasella, Mach. emotori agric, **12** (1959)
14. S. Nambu, S. Hata, T. Endo, Agr. Nach., Japan, **45**, 3, 311 (1983)
15. S. Stroe-Oprea, M. Munteanu, Stud. Cercet. de mec. aplic., **7**, 3, 235 (1973)