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The Results of Theoretical Studies of the Vibrator Compensating Chamber of the Dispenser of Mineral Fertilizers

[1]Sayakhat Nukeshev, [1]Indira Mamyrbaeva, [2]Kairat Eskhozhin,
[1]Aigul Balabekova and [1]Ziyada Zhaksylykova
[1]Department of Technical Mechanics,
[2]Department of Agricultural Technique and Technology, Kazakh Agro Technical University,
S. Seifullin, Astana, Kazakhstan

Abstract: In agriculture of the Republic of Kazakhstan the wide application of subsurface technologies introduce the main dose of fertilizers is slow due to the lack of necessary equipment. The cultivators tillers KPG 2.2 and the chisel-fertilizers VCO-4, designed for this purpose, sowing machines do not fully comply with agro-requirements on the uniformity and stability of seed placement and closing the working bodies distribution of fertilizers in soil. As a result, these machines have not found widespread use in manufacturing. This directly affects the yield. To solve the problem of uniform distribution of fertilizers inside the soil, the unification of machines for application of mineral and organic fertilizers proposed sowing machine for making the main dose of organic and mineral fertilizers. Obtaining uniform flow of the fertilizer is carried out by applying the compensating chamber with vibrating plate installed under the sowing window. As a result of theoretical studies, we have defined the frequency, amplitude, mode and studied forced oscillations in the case of actions on the middle rod of the periodic disturbing forces without taking into account the resistance. Thus, the proposed sowing device distributes fertilizers evenly, destroys lumps and also excludes arches.

Key words: Fertilizer device, fertilizer, elastic plate, a slide gate, bridge breaking cone, oscillations

INTRODUCTION

Climate change has an impact on food and economic spheres of the state, especially in developing countries that are dependent on the rural economy (Rosenzweig *et al*., 2014; Nelson *et al*., 2014; Elliott *et al*., 2014). In this regard, there is a threat of a food crisis (Gibson, 2016; Margulis, 2014).

It is known that for the last 40-50 years the content of humus in the soils used in the production of grain in Kazakhstan decreased by 20-30% (Kuribasev, 2003; Eleshev, 2005; Baraev, 1975). The main reason I think wind erosion and the low level of fertilizer application. Therefore, in conditions of risky agriculture of Northern Kazakhstan, important in the fight against soil erosion is the application of mineral and organic fertilizers (Khoroshilov, 1966; Gossen, 2004; Chernenok, 2009).

In the practice of agriculture of the Republic of Kazakhstan the introduction of production technology-soil deposit of high dose of fertilizers before sowing and at sowing is slow due to the lack of necessary equipment, although, the viability and environmental safety of this method are not controversial (Suleymenov, 1988; Vakhrameev, 1990). This supports soil fertility has no toxic effect and increases the yield, which is significant in food security of the country (Chernenok, 2009; Kaplan, 2004; Chernovolov, 2000).

The aim of this research is theoretical determination of the natural frequencies, the amplitude of natural modes of vibration of the compensating chamber of the dispenser of mineral fertilizers and without taking into account the resistance of the medium.

MATERIALS AND METHODS

Sowing device comprises a hopper with bridge breaking cone and damper. Have a sowing window of the bunker on the master and driven rollers progressively moving the flexible conveyor with the pins. The driven roller includes a receiving pneumo-fertilizer funnel.

The device operates as follows. Fertilizer by gravity and with the help of bridge breaking cone fall into the seed window and on to progressively moving conveyor. Fertilizers involved by pins are distributed on the surface of the conveyor and transported on the gathering. At the gathering, the driven roller fertilizer from the conveyor are

Corresponding Author: Sayakhat Nukeshev, Department of Technical Mechanics, Kazakh Agro Technical University, S. Seifullin, Astana, Kazakhstan

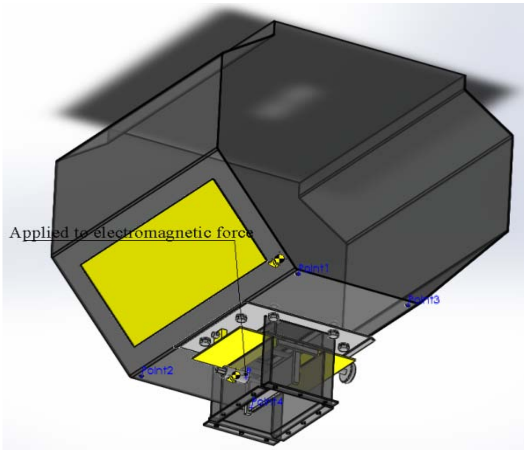

Fig. 1: Compensating chamber of a fertilizer distributing device

moved to the receiver pneumo-fertilizer funnel and distributed to the coulters. Seeding rate is adjusted by changing the frequency of rotation of the leading roller and changing the angle of attack of the pins on the conveyor.

A preliminary exploratory study showed that fertilizer sowing window fall on the forward moving conveyor uneven layer. And this in turn, leads to uneven distribution of organic fertilizers in the soil.

To obtain an evenly distributed layer of fertilizer on the conveyor of the proposed technical solution, consisting in the establishment after sowing window for compensating camera.

Inside the compensating chamber is longitudinally installed elastic plate 1, the center of which one end is fixed to the rod 2 (Fig. 1). The other end of the rod 2 is connected to a vibrator (not shown as a known device) installed outside of the compensating chamber.

In this case the technological process of seeding is carried out as follows. A continuous flow of fertilizer is supplied in the sowing window 3 and then the flow takes place in the compensating chamber where under the influence of the plate 1 is vibrating in the range of audio frequencies, receives a fluidized condition and evenly falls on a steadily moving conveyor. Conducted search experiments showed that the use of vibrating plates in the environment of granulated mineral and organic fertilizers allows breaking the lumps of fertilizer and eliminates bridging.

RESULTS AND DISCUSSION

Previously, we have considered the oscillations of the hopper of the machine for fertilizer. Created a dynamic

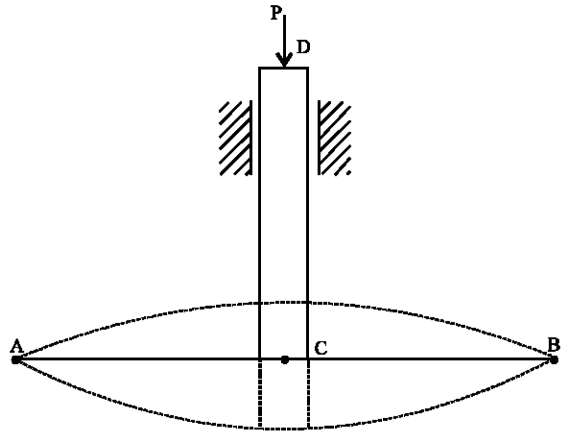

Fig. 2: Scheme of installation of the rods

model of the dispenser by means of matrix mechanics as a system with seven degrees of freedom. The obtained numerical and graphical results of determination of natural frequencies and damped and forced vibrations of a system consisting of a hopper body and a vibrating plate.

In the present research deals with the flexural vibrations of vibrating plate as a one-dimensional continuum system. Model vibrating the plates are in the form of construction, consisting of two rods AB and DC (Fig. 2).

The ends of the thin elastic rod AB is fixed. The DC rod end attached to the middle of the rod AB and can move perpendicular to the rod AB. The rod AB can perform transverse vibrations. It via. the terminal CD acts the exciting force R.

In the beginning, consider the free flexural vibrations of the rod AB assuming that the ends of the rod is rigidly fixed. The case hinged fastening both ends is given in the literature (Biderman, 1972).

Based on the Bernoulli hypothesis and neglecting the inertia forces of the particles of the rod to move along the axis of the differential equation of flexural vibrations of the rod in the case of a beam of constant cross section without taking into account the resistance of the external environment (Vasilenko, 1992) can be written as:

$$\frac{\partial^4 x}{\partial z^4}+\frac{m_0}{EI}\times\frac{\partial^2 x}{\partial t^2}=0 \quad (1)$$

Where:

x(z, t) = The dynamic function of the displacement

EI = The bending stiffness in the plane of oscillation

m_0 = The mass per unit length of the rod

$a=\sqrt{\frac{EI}{m_0}}$ = The speed of propagation of the wave front in the rod

Consistent with their fluctuations the solution of the motion Eq. 1 represented in the form:

$$x(z,t) = u(z)\cos(pt+\varphi) \tag{2}$$

Where:

$u(z)$ = The amplitude function
p = The angular frequency
φ = The initial phase

Substituting Eq. 2 into Eq. 1, we have:

$$\frac{d^4u}{dz^4}-\alpha^4u = 0 \tag{3}$$

Where:

$$\alpha^4 = \frac{p^2m_0}{EI} = \frac{p^2}{a^2}$$

The roots of the characteristic equation corresponding to Eq. 3 is equal to $\pm\alpha$ and $\pm\alpha\sqrt{-1}$. In accordance with this solution of the homogeneous Eq. 3 are expressed via trigonometric and exponential functions of argument (αz). However, considerable convenience is the use of introduced by Krylov combinations of these functions. Labeling Krylov functions with symbols K_1-K_4 can represent the solution of Eq. 3 in the form:

$$u(z) = C_1K_1(\alpha z)+C_2K_2(\alpha z)+C_3K_3(\alpha z)+C_4K_4(\alpha z) \tag{4}$$

where, $C_1, ..., C_4$ is the constant of integration:

$$\begin{aligned} K_1(\alpha z) &= \frac{1}{2}(\operatorname{ch}\alpha z+\cos\alpha z) \\ K_2(\alpha z) &= \frac{1}{2}(\operatorname{sh}\alpha z+\sin\alpha z) \\ K_3(\alpha z) &= \frac{1}{2}(\operatorname{ch}\alpha z-\cos\alpha z) \\ K_4(\alpha z) &= \frac{1}{2}(\operatorname{sh}\alpha z-\sin\alpha z) \end{aligned} \tag{5}$$

Successive derivatives of the Krylov functions are linked by dependencies:

$$\begin{aligned} \frac{d}{dz}K_1(\alpha z) &= \alpha K_4(\alpha z) \\ \frac{d}{dz}K_2(\alpha z) &= \alpha K_1(\alpha z) \\ \frac{d}{dz}K_3(\alpha z) &= \alpha K_2(\alpha z) \\ \frac{d}{dz}K_4(\alpha z) &= \alpha K_3(\alpha z) \end{aligned} \tag{6}$$

Fig. 3: Rod with clamped ends

Then, the derivative of the function (Eq. 4) can be written in the form:

$$\begin{aligned} \frac{du}{dz} = {} & C_1\alpha K_4(\alpha z)+C_2\alpha K_1(\alpha z)+ \\ & C_3\alpha K_2(\alpha z)+C_4\alpha K_3(\alpha z) \end{aligned} \tag{7}$$

At each end of the beam there are two boundary conditions that depend on the method of binding Fig. 3. In the case of fixing the ends of the rod the boundary conditions for the left end can be written as:

$$u(0) = 0;\ \left.\frac{du}{dz}\right|_{z=0} = 0 \tag{8}$$

and on the right end of the bar:

$$u(1) = 0;\ \left.\frac{du}{dz}\right|_{z=1} = 0 \tag{9}$$

We apply the boundary conditions (Eq. 8) subject to Eq. 5 and 6. We get:

$$\begin{aligned} & u(0) = C_1\times C1 = 0 \rightarrow C_1 = 0 \\ & \left.\frac{du}{dz}\right|_{z=0} = C_2\alpha = 0 \rightarrow C_2 = 0 \end{aligned}$$

In this case, the solution to Eq. 4 of the differential Eq. 3 simplifies to:

$$u(z) = C_3K_3(\alpha z)+C_4K_4(\alpha z) \tag{10}$$

We now apply the boundary conditions (Eq. 9):

$$\left.\begin{aligned} & u(1) = C_3K_3(\alpha l)+C_4K_4(\alpha l) = 0 \\ & \left.\frac{du}{dz}\right|_{z=1} = C_3aK_2(\alpha l)+C_4\alpha K_3(\alpha l) = 0 \end{aligned}\right\}$$

From this:

$$C_3 = -C_4\times\frac{K_4(\alpha l)}{K_3(\alpha l)} \tag{11}$$

then:

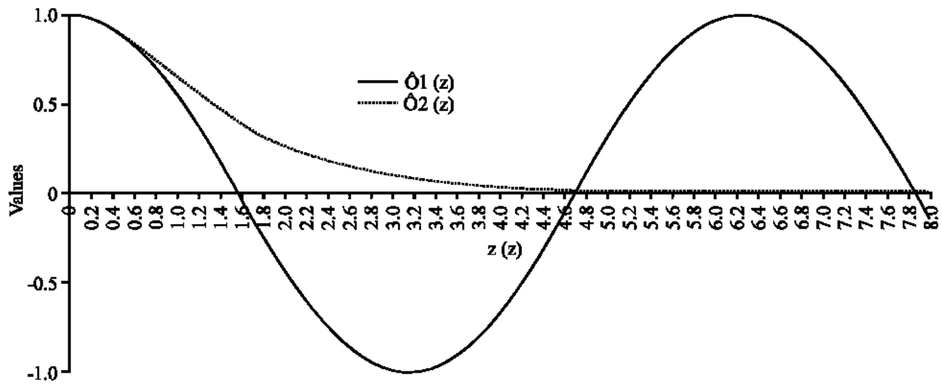

Fig. 4: Graphical solution of the frequency (Eq. 13)

$$\left.\frac{du}{dz}\right|_{z=l} = C_4\alpha\left(-\frac{K_4(\alpha l)}{K_3(\alpha l)}\times K_2(\alpha l)+K_3(\alpha l)\right)=0$$

Given that $C_4\alpha\neq 0$, we have equation of frequencies, expressed through Krylov functions in the form:

$$K_3^2(\alpha l)-K_2(\alpha l)\times K_4(\alpha l)=0 \quad (12)$$

where we have introduced the notation $\lambda = \alpha l$ or $\alpha = \lambda/l$. Substituting here the expression of the function $K_2(\alpha l)$, $K_3(\alpha l)$, $K_4(\alpha l)$ according to Eq. 5, we obtain equation of frequencies, expressed via. trigonometric and hyperbolic functions:

$$\left(\frac{1}{2}(\text{ch}\lambda-\cos\lambda)\right)^2-\left(\frac{1}{2}(\text{sh}\lambda+\sin\lambda)\right)\times\left(\frac{1}{2}(\text{sh}\lambda-\sin\lambda)\right)=0$$

Using expressions of Krylov functions (Eq. 5), we get:

$$\cos\lambda = \frac{1}{\text{ch}\lambda} \quad (13)$$

A graphical solution of this equation is shown in Fig. 4. The point of intersection of the two lines give the values of λ. It follows from figure that the roots of Eq. 13 is infinite. A zero value of ë corresponds to translational or rotational motion of the beam as a rigid body. The roots of Eq. 13: $\lambda_1 = 4.694$; $\lambda_2 = 7$ and if k>2, the solution can be represented as:

$$\lambda_k = \frac{2k-1}{2}\pi, \quad k = 1,2,3,\ldots \quad (14)$$

The oscillation frequency is determined by Eq. 15:

$$p_k = \frac{\lambda_k^2}{l^2}\sqrt{EI/m_0} = \frac{\lambda_k^2}{l^2}a \quad (15)$$

The amplitude functions are determined by Eq. 16, Fig. 5:

$$u_k(z) = K_4(\lambda_k)\times K_3\left(\lambda_k\frac{z}{l}\right)-K_3(\lambda_k)\times K_1\left(\lambda_k\frac{z}{l}\right),\ k = 1,2,3,\ldots \quad (16)$$

Normal frequencies and forms of natural oscillations of a rod with fixed ends defined above. Applies the results obtained to solve the problem of forced oscillations of a rod with clamped ends, when the middle rod is applied perturbing periodic force (Fig. 6):

$$P(t) = P_0\sin\omega t$$

forced oscillations expanding in eigenfunctions of the system and expressing the solution as an infinite series. The advantage of this method is its generality.

According to the method of principal coordinates displacement of any point of the beam can be represented as:

$$x(z,t) = \sum_{k=1}^{\infty} q_k(t)\times u_k(z) \quad (17)$$

Where:

$u_k(z)$ = The deflection during normal oscillation (amplitude function)

qk(t) = The time function (main coordinate) determined from equation:

$$\ddot{q}_k + p_k^2 q_k = Q_k(t)/\mathfrak{M}_k \quad (18)$$

Where:

p_k-k = The natural frequency defined by expression (Eq. 14)

Q_k = The generalized force, equal to the sum of the products of the disturbing forces on moving their applications are k-normal volume fluctuations

$\mathfrak{M}_k = \int_0^l u_k^2 m_0 dz$ = The generalized mass

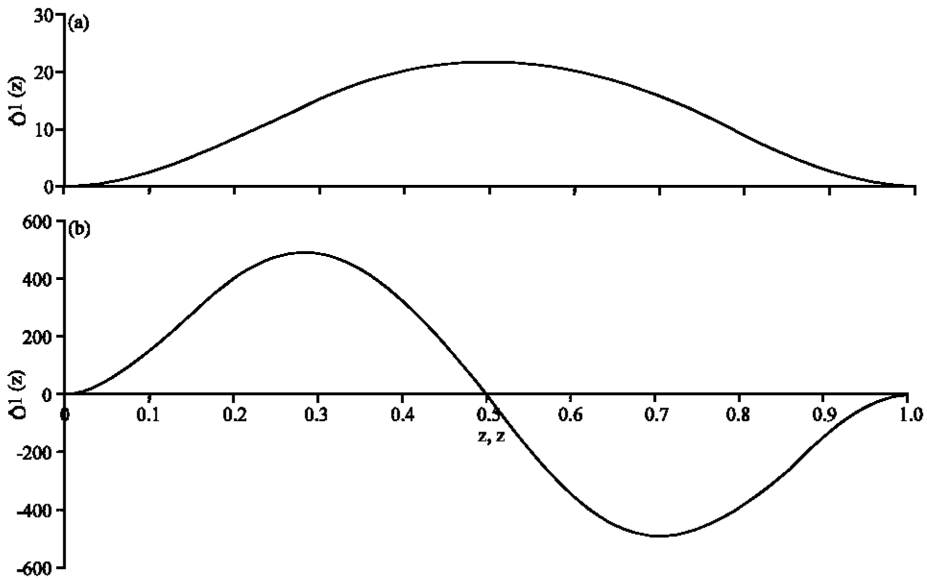

Fig. 5: a, b) Amplitude function oscillations of a rod with rigidly clamped ends according to Eq. 16 in a dimensionless form for $\lambda_1 = 4.694$ and $\lambda_2 = 7.854$

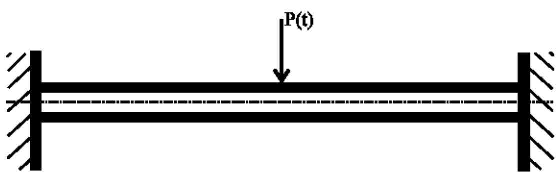

Fig. 6: Rod with clamped ends with driving force applied in the middle

For beams of constant cross-section on elastic supports hard line when k-oscillation, is a line corresponding to the expression (Eq. 16) (Table 1). Thus, the generalized mass is:

$$\mathfrak{M}_k = \int_0^1 \left(\begin{matrix} K_4(\lambda_k)\times K_3\left(\lambda_k \frac{z}{1}\right) - \\ K_3(\lambda_k)\times K_1\left(\lambda_k \frac{z}{1}\right) \end{matrix} \right)^2 m_0 dz = \frac{m_0 l}{4} B_k \quad (19)$$

where the constant value is marked with:

$$B_k = \left[\begin{matrix} \left(\begin{matrix} sh\lambda_k - sin\lambda_k + \\ cos\lambda_k \end{matrix} \right)^2 \left(\frac{1}{4\lambda_k} sh(2\lambda_k) + \frac{1}{2} \right) - \\ \left(\begin{matrix} (sh\,\lambda_k - sin\,\lambda_k)^2 - \\ (ch\,\lambda_k - cos\,\lambda_k)^2 \end{matrix} \right) \left(\frac{2}{\lambda_k} ch\lambda_k \left(\begin{matrix} cos\,\lambda_k + \\ sin\lambda_k \end{matrix} \right) - 2 \right) + \\ \left(\begin{matrix} sh\lambda_k - sin\lambda_k + \\ ch\lambda_k - cos\lambda_k \end{matrix} \right)^2 \left(\frac{1}{4\lambda_k} sin(2\lambda_k) + \frac{1}{2} \right) \end{matrix} \right] \quad (20)$$

Table 1: Results of calculation

Parametr	$\lambda_1 = 4.694$	$\lambda_2 = 7.854$	$\lambda_3 = 10.995$	$\lambda_4 = 14.137$
B_k	8997.400	4.265	2.138	1.102

Table 2: Generalized coordinate

Parameter	$\lambda_1 = 4.694$	$\lambda_2 = 7.854$	$\lambda_3 = 10.995$	$\lambda_4 = 14.137$
$4P_0/m_0l\times B_k(p_k^2 - \omega^2)$	-0.000084	-0.18	-0.368	-0.737

We present the results of calculations. The generalized force:

$$Q_k = P(t) u_k\left(\tfrac{1}{2}\right) = P_0 \left(\begin{matrix} K_4(\lambda_k)\times K_3\left(\frac{\lambda_k}{2}\right) - \\ K_3(\lambda_k)\times K_4\left(\frac{\lambda_k}{2}\right) \end{matrix} \right) \sin \omega t \quad (21)$$

Equation that determines the functions takes the $q_k(t)$ form:

$$\ddot{q}_k + p_k^2 q_k = \frac{4P_0}{m_0 l B_k} \sin \omega t \quad (22)$$

The solution of this equation has the form:

$$q_k(t) = \frac{4P_0}{m_0 l \times B_k (p_k^2 - \omega^2)} \sin \omega t \quad (23)$$

Below shows the calculated amplitude of forced oscillations for some generalized coordinates when $P_0 = 50$ N, $\omega = 50$ Hz, $m_0 l = 0.1068$ kg (Table 2). The displacement at any cross section of the rod under forced vibration is determined by Eq. 24:

$$x(z,t)=\frac{4P_0}{m_0 l}\sin\omega t\sum_{k=1}^{\infty}\frac{K_4(\lambda_k)\times K_3\left(\lambda_k\frac{z}{l}\right)-K_3(\lambda_k)\times K_1\left(\lambda_k\frac{z}{l}\right)}{B_k\left(p_k^2-\omega^2\right)} \quad (24)$$

The deflection become infinite when the frequency ω of the perturbation coincides with one of natural frequencies of the beam.

Now consider a problem of flexural vibrations of a rod with a rigid fastening of both ends in the presence of viscous friction when the coefficients of viscous friction is proportional to the mass or stiffness elements of the system. The differential equation of motion of a rod subject to external and internal friction is written as:

$$EI\left(\frac{\partial^4 x}{\partial z^4}+a_2\frac{\partial^5 x}{\partial z^4\partial t}\right)+m_0\frac{\partial^2 x}{\partial t^2}+a_1\frac{\partial x}{\partial t}=0 \quad (25)$$

Where:

α_1 = The factor of external friction

α_2 = The coefficient taking into account internal friction

Equation 25 has a solution of the form Eq. 17:

$$x(z,t)=q_k(t)u_k(z)$$

where, $u_k(z)$-k-form of natural vibrations of the beam without friction (Eq. 16). Substituting the expression (Eq. 17) into Eq. 25 and taking into account that the function $u_k(Z)$ satisfies the differential Eq. 3, we obtain the differential equation that determines the generalized coordinate $q_k(t)$:

$$\ddot{q}_k+\left(\alpha_1+p_k^2\alpha_2\right)\dot{q}_k+p_k^2 q_k=0,\ k=1,2,3,\ldots \quad (26)$$

Let us represent $2n_k = \alpha_1+p_k^2\alpha_2$. The values of the coefficient n_k take on the basis of experimental data. For example, when a harmonic perturbation of frequency, given that no losses on internal friction for most materials or structural hysteresis of the frequency do not depend, I suppose n_k inversely proportional to the frequency of disturbance. The differential Eq. 26 can be written in the form:

$$\ddot{q}_k+2n_k\dot{q}_k+p_k^2 q_k=0,\ k=1,2,3,\ldots \quad (27)$$

The solution of the differential Eq. 27 subject to initial conditions t = 0, $q_k(0) = q_{k0}$; $\dot{q}_k(0)=\dot{q}_{k0}$:

$$q_k=A_k e^{-nt}\sin\left(p_k^* t+f_0\right) \quad (28)$$

Where:

$$p_k^*=\sqrt{p_k^*-n^2}$$

$$A_k=\sqrt{q_{k0}^2+\frac{\left(\dot{q}_{k0}+nq_{k0}\right)^2}{p_k^2-n^2}}$$

$$\varphi_0=\mathrm{arctg}\left(\frac{q_{k0}\sqrt{p_k^2-n^2}}{\dot{q}_{k0}+m_{qk0}}\right)$$

The differential equation of forced oscillations under the action of the periodic perturbation forces with damping in principal coordinates can be written as:

$$\ddot{q}_k+2n_k\dot{q}_k+p_k^2 q_k=Q_k/\mathfrak{M}_k$$

where, the generalized force and generalized mass are determined according to the expression (Eq. 19-21):

$$\ddot{q}_k+2n_k\dot{q}_k+p_k^2 q_k=\frac{4P_0}{m_0 lB_k}\sin\omega t \quad (29)$$

when $n_k<p_k$ the general solution of Eq. 29 takes the form:

$$q_k=e^{-n_k t}\left(\begin{matrix}C_1\cos\left(p_k^* t\right)+\\ C_2\sin\left(p_k^* t\right)\end{matrix}\right)+\frac{h_k}{\sqrt{\left(p_k^2-\omega^2\right)^2+4n_k^2\omega^2}}\sin(\omega t-\varepsilon) \quad (30)$$

Where:

h_k = $4P_0/m_0 lB_k$

ε = $\mathrm{arctg}\ 2n_k\omega/p_k^2-\omega^2$

C_1 and C_2 = The constants of integration

Equation 30 shows that the body performs a complex oscillatory motion and it consists of two harmonic oscillations. The first term $e^{-n_k t}\left(C_1\cos\left(p_k^* t\right)+C_2\sin\left(p_k^* t\right)\right)$ expresses the self-oscillations of the rod and the second member $\frac{h_k}{\sqrt{\left(p_k^2-\omega^2\right)^2+4n_k^2\omega^2}}\sin(\omega t-\varepsilon)$ determines the forced oscillations. Natural oscillations are damped due to the factor $e^{-n_k t}$ and after some time they disappear. Therefore, almost (since, the presence of resistance is inevitable), we can assume that after a certain period of time, called the period of establishment, the body will make only forced oscillations. Below shows the calculated amplitude of forced oscillations for some generalized coordinates when P_0 = 50 N, ω = 50 Hz, $m_0 l$ = 0.1068 kg (Table 3).

Table 3: $n_k<p_k$ the general solution of equation

Parameters	λ_1 = 4.694, n_1 = 0.9	λ_2 = 7.854, n_2 = 0.8	λ_3 = 10.995, n_3 = 0.6	λ_4 = 14.137, n_2 = 0.4
$\frac{h_k}{\sqrt{\left(p_k^2-\omega^2\right)^2+4n_k^2\omega^2}}$	-0.000084	-0.178	-0.368	-0.739

Thus, the use of mineral fertilizers can significantly increase the yield. However, the lack of necessary equipment leads to an irrational distribution of fertilizers and consequently to an increase in financial costs. The proposed method makes it possible to determine the amplitude oscillations and evenly distribute the fertilizers.

CONCLUSION

We solved the problem of the flexural vibrations of a rod with a rigid fixing of both ends as a one-dimensional model of vibrating plate of the hopper of the fertilizer machine. Use the method of principal coordinates gave the opportunity to define their own frequency, amplitude, mode shapes and without taking into account the resistance of the medium and to study forced oscillations in the case of actions on the middle rod of the periodic disturbing forces. The natural frequencies of the rod approximately coincided with the results of the model calculation of the bunker that had been done before.

RECOMMENDATION

The result can be used for further analysis of the influence of the characteristics of vibrations on uneven and fragile sowing of mineral fertilizers.

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