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BASING THE PARAMETERS OF THE WORKING PART OF THE NUT KERNEL SORTING MACHINE

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ABSTRACT

The main parameters of the working part of the walnut kernel sorting machine are based on the article. The walnut kernel sorting machine consists of a kernel drop hopper, sieves and sieves attached frame, spring, vibrator and electric motors. A theoretical study was conducted to determine the optimal geometric and dynamic parameters of these parts. In this case, as the main parameters of the sieve considered as the working part of the machine, the full length of the sieve, the vibration amplitude of the sieve, the dimensions of the circular hole in the sieve, the slope angle of the sieve, the coefficient of friction of the sieve material with the core, the height of the frame, the length of the frame, and the length of the spring were taken. Results of the research were calculated by mathematical modeling and graphs were made.

Key words. Core Drop Hopper, Sieve, Sieve Attached Frame, Spring, Vibrator, Core, Sorting, Full Length Of Sieve, Amplitude Of Vibration Of Sieve, Sizes Of Circular Holes In Sieve, Slope Angle Of Sieve.

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Original Article

INTRODUCTION

At the present time, the demand for the quality indicators of the product in the production and preparation of raw materials in the world market is reflected in the cost of the raw materials of the product. In particular, this situation can be seen from the price differences of different types of hard-shelled nuts and kernels sorted by size. In particular, in walnut kernel prices, the price of sorted kernels is 70-80% higher than the price of unsorted kernels. Today, the world production and processing of walnuts has reached 2.8 million tons [1]. However, due to the insufficient development of equipment for sorting these nut kernels according to quality indicators, sorting is carried out by hand in many countries. Currently developed sorting machines are designed for processing large quantities of kernels on an industrial scale and are not widely used because the cost is too high for small seasonal production farms. Therefore, creating cheap small models of existing sorting devices and improving existing ones is one of the urgent tasks.

Scientific and research work is being carried out in the world aimed at the production of machines for processing walnut kernels, sorting them according to quality indicators, energy saving, sorting accuracy and high productivity. In the Republic of Uzbekistan, an average of 600,000 tons of walnuts are grown annually [2, 4, 5]. Due to the fact that the demand for their processing is increasing day by day, it is urgent to create new efficient, energy-saving generations of sorting devices. Extensive measures are being implemented in the Republic of Uzbekistan to increase the export potential of walnuts and their kernels. For example, in the decision of the President of the Republic of Uzbekistan dated June 1, 2017 PQ-3025 on "Establishment of the association of walnut producers and exporters and organization of its activities", including "... walnut producers "organization of associations of producers and exporters and further increase of walnut production and sales in the domestic and foreign markets" [3, 6] are defined. In order to fulfill the tasks specified in this decision, it is important to develop machines with high sorting accuracy and productivity based on the improvement of existing types of walnut kernel processing and size sorting machines.

Discussions

To analyze the dynamics of the oscillating movement of the machine under study, we use the general scheme of the machine presented in Figure 1. We will analyze the vibration of the car in relation to its center of gravity. Therefore, based on the determination of the center of gravity of each of the parts participating in the vibration, the center of gravity of the machine and the complete moments of inertia relative to this center of gravity are also determined.

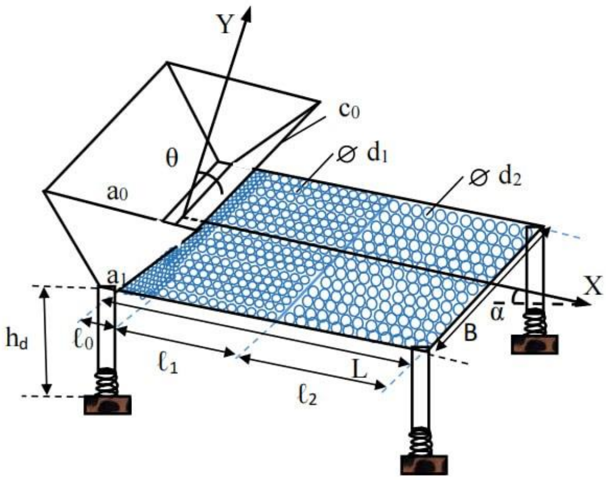

Figure 1: General Scheme of the Machine.

Using the parameters given in Figure 1, the center of gravity of all parts of the machine was calculated in relation to the OXY coordinate axes as follows [7-9].

Mass and center of gravity of the sieve for removing fine debris

$$m_0 = \left(b_1 l_0 B - n_0 \pi r_0^2 b_1\right)\rho = \left(l_0 B - n_0 \pi r_0^2\right) b_1 \rho \qquad (kg)$$

$$x_{l_0} = \frac{l_0}{2}\cos\alpha;\ y_{l_0} = \left(L - \frac{l_0}{2}\right)\sin\alpha \qquad (m) \qquad (1)$$

The full length of the sieve system

$$L = l_0 + l_1 + l_2 \qquad (m) \qquad (2)$$

Appropriate masses and center of gravity of sieves

$$m_1 = (b_1 l_1 B - n_1 \pi r_1^2 b_1)\rho = (l_1 B - n_1 \pi r_1^2) b_1 \rho \qquad (m)$$

$$x_{l_1} = \left(l_0 + \frac{l_1}{2}\right)\cos\alpha;\ y_{l_1} = \left(L - \left(l_0 + \frac{l_1}{2}\right)\right)\sin\alpha$$

$$m_2 = (l_2 B - n_2 \pi r_2^2) b_1 \rho$$

$$x_{l_2} = \left(l_0 + l_1 + \frac{l_2}{2}\right)\cos\alpha = \left(L - \frac{l_2}{2}\right)\cos\alpha;\ y_{l_2} = \frac{l_2}{2}\sin\alpha \qquad (3)$$

Sieve support and frame mass center of gravity

$$m_{tr} = 2b_2 d\rho[B + L(1 + 2\mathrm{Sin}\,\alpha)]$$

$$x_p = \frac{L}{2};\ y_p = 0 \qquad (4)$$

Total mass and center of gravity of the vibrating part of the machine

$$m = m_\delta + m_0 + m_1 + m_2 + m_{tr} \qquad (kg)$$

$$x_C = \frac{x_\delta m_\delta + x_0 m_0 + x_1 m_1 + x_2 m_2 + x_p m_{tr}}{m} \qquad (m) \qquad (5)$$

$$y_C = \frac{y_\delta m_\delta + y_0 m_0 + y_1 m_1 + y_2 m_2 + y_p m_{tr}}{m} \qquad (m)$$

Based on the formulas given above, the calculated dimensions of each part of the machine prepared for the experiment are presented in Table 1.

Table 1:Calculated size value of walnut sorting machine parameters

T/R	The Name of the Main Parameters	Size Value	Unity
1	Fine sieve length	0.2	m
2	1/8 the length of the sieve	0.5	m
3	1/4 the length of the sieve	0.5	m
4	The full length of the sieve	1.2	m
5	Sieve elevation angle	0.34907	rad
6	Iron density	7800	kg/m³
7	Bunker metal thickness	0.002	m
8	Sieve width	0.6	m
9	The number of holes in the smallest cleaning sieve	2857.1	piece
10	1/8 is the number of holes in the sieve	2083.3	piece
11	1/8 is the diameter of the hole in the sieve	0.01	m
12	The cleaning part, the diameter of the hole in the sieve	0.005	m
13	1/4 is the diameter of the hole in the sieve	0.015	m
14	1/4 is the number of holes in the sieve	980.39	piece
15	Profile metal thickness	0.002	m
16	Sieve full mass	19.163	kg

T/R	The Name of the Main Parameters	Size Value	Unity
17	Bunker mass	6.2793	kg

Analysis

We consider that the oscillating motion of the machine is an infinitesimally small deflection relative to the point O1. Based on the above-mentioned considerations, a simplified calculation scheme of the machine can be described as follows (see Figure 2).

We consider this considered mechanical system as ideal, having a coupling relationship. Let the amplitude and frequency of the torque M0 supplied to the system be p. The considered conservative mechanical system is affected by the generalized harmonic external force Qg, in addition to the damping generalized resistance force Qq. Let's formulate the 2nd type Lagrange equation for the machine:

$$\frac{d}{dt}\frac{\partial T}{\partial x} + \frac{\partial W}{\partial x} = Q_g + Q_q \qquad (N) \qquad (6)$$

The generalized resistive force can be written as:

$$Q_q = -b\dot{y}_c \quad (7)$$

When determining the external force acting on the system, using the method of possible displacement, we divide the work δA by the work δr by the external force in moving point A to point A1:

$$Q_g = \frac{\delta A}{\delta r}; \qquad (N)$$

Note that the displacement of point A is very small $\partial r = y_C \delta\varphi$ we can write in the form The external force rotates the point A around the point δφ through an angle dφ, Uwhere:

$$Q_g = \frac{M\delta\varphi}{y_C\delta\varphi} = \frac{M}{y_C} = F_0 Sin(\omega_1 t + \gamma_0) \qquad (N) \qquad (8)$$

The frequency of action of this force can be expressed by the linear speed of point A:

$$\omega_1 = \frac{\upsilon_A}{y_C} = \frac{\dot{x}}{y_C} \qquad (rad / sek) \qquad (9)$$

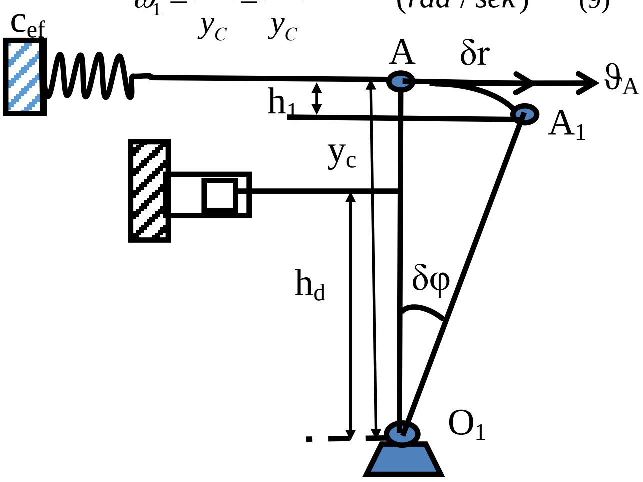

Figure 2: Scheme for Calculating the Vibration Motion of the Machine

The kinetic energy of the system is as follows:

$$T = \frac{1}{2} I \omega_1^2 = \frac{1}{2} I \left(\frac{\dot{x}}{y_C} \right)^2 \qquad (J) \qquad (10)$$

The potential energy is equal to the sum of changes in the potential energies of the equivalent spring and machine mass:

$$W = W(C) + W(P) \qquad (J)$$

Potential energy of the spring:

$$W(C) = -A(C) = \frac{1}{2} C \lambda^2 = \frac{1}{2} C (\lambda_{CT} + x)^2 \qquad (J) \ (11)$$

The deformation of a spring in a static state can be called λ, and its elongation due to impact can be called δx:

$$\lambda = \lambda_{CT} + S_A = \lambda_{CT} + \delta x \ (12)$$

Change in potential energy of the machine due to displacement in the vertical direction:

$$W(P) = -A(P) = -mgh_1 \ (13)$$

In that case:

$$h_1 = y_C (1 - \cos \delta\varphi) (14)$$

δφ we consider the angle to be infinitesimally small and extend it to the 2nd term in the Taylor series:

$$\cos \delta\varphi = 1 - \frac{1}{2} \delta\varphi^2 + ... \approx 1 - \frac{1}{2} \delta\varphi^2 \ (15)$$

$$W(P) = -\frac{1}{2} mgy_C \delta\varphi^2 \qquad (16)$$

$$\delta\varphi = \frac{x}{y_C}$$

When the machine is in equilibrium with respect to vibration, the following equilibrium condition is fulfilled:

$$\left(\frac{\partial W}{\partial x} \right)_{x=0} = 0 \qquad (17)$$

we determine the amount of deformation in the state of static equilibrium:

$$C(\lambda_{CT} + x) - \frac{mgx}{y_C} = 0$$

$$x = 0 \text{ da } C\lambda_{CT} = 0$$

$$\lambda_{CT} = 0 \qquad (18)$$

Putting the above expressions into the 2nd type Lagrange equation, we get the differential equation of the horizontal vibration of the machine under consideration with respect to the point O_1:

$$\frac{I}{y_C^2}\ddot{x} + C_t x = -b\dot{x} + F_0 Sin(\omega t + x_1) \quad (19)$$

When determining the resistance coefficient b in the equation (18), we consider it as a damper resistance to ensure smooth vibration of the machine. Taking into account that this resistance is placed at a distance hd, it can be taken as proportional to the linear velocity of this point O1 relative to the center. Based on these points, we find the relationship between the damping coefficient and the resistance coefficient based on the following equations:

$$R_q = \beta \upsilon_k$$

$$\upsilon_k = h_d \delta\dot{\varphi} = h_d \frac{\dot{x}}{y_C}$$

$$R_q = \beta \frac{h_d}{y_C}\dot{x}$$

$$b\dot{x} = \beta \frac{h_d}{y_C}\dot{x};$$

Damping coefficient:

$$b = \frac{h_d}{y_c}\beta$$

$$\beta = \frac{y_C}{h_d} b \quad (20)$$

(19) to make the equation normal:

$$\ddot{x} + \frac{C_t y_C^2}{I_t} x + \frac{b y_C^2}{I_t}\dot{x} = \frac{F_0 y_C^2}{I_t} Sin(\omega t + \gamma_0)$$

$$\ddot{x} + 2n\dot{x} + k^2 x = h_0 Sin(\omega t + \gamma_0) \qquad (21)$$

$$2n = \frac{by_C^2}{I_t} \text{ - binary artificial coefficient;} \qquad (22)$$

$$k^2 = \frac{C_t y_c^2}{I_t} \text{ - the square of the oscillation cyclic frequency;}$$

$$h_0 = \frac{F_0 y_C^2}{I_t} \text{ - acceleration coefficient.}$$

(22) we determine the solution of the equation by the sum of the general solution of the homogeneous equation and the particular solutions of the inhomogeneous equation:

$$x = x_1 + x_2 \qquad (23)$$

(23)is a homogeneous equation of:

$$\ddot{x} + 2n\dot{x} + k^2 x = 0 \qquad (24)$$

Characteristic equation

$$S^2 + 2n_1 S' + k^2 = 0$$

$$S_{1,2} = -n \pm \sqrt{k^2 - n^2} \quad (25)$$

1) n<k – small vibrations;

2) n>k – big swings;

3) n=k – critical resistance vibration.

That's why the machine works with small vibrations $n<k$, in that case

$$S_{1,2} = -n \pm i\sqrt{k^2 - n^2}; \; i = \sqrt{-1}$$

$$\begin{cases} x = e^{-nt}\left(C_1 e^{ik_1 t} + C_2 e^{-ik_1 t}\right) = e^{-nt}\left(C_1 Cosk_1 t + C_2 Sink_1 t\right) \\ \dot{x} = -ne^{-nt}\left(C_1 Cosk_1 t + C_2 Sink_1 t\right) + e^{-nt}\left(-C_1 k_1 Sink_1 t + C_2 k_1 Cosk_1 t\right) \end{cases}$$

$$k_1 = \sqrt{k^2 - n^2}$$

We find the integration coefficients using the initial conditions:

$$t = 0; \; x = x_0, \; \dot{x} = x_0$$

$$\begin{cases} x_0 = C_1; \quad C_1 = x_0 \\ \dot{x}_0 = -nC_1 + C_2 k_1; \quad C_2 = \dfrac{\dot{x}_0 + n\,x_0}{k_1} \end{cases}$$

$$x = e^{-nt}\left(x_0 Cosk_1 t + \frac{\dot{x}_0 + nx_0}{k_1} Sink_1 t \right)$$

We determine the vibration amplitude of the machine and the initial vibration phase γ0:

$$C_1 = A_1 Sin\gamma_0$$
$$C_2 = A_1 Cos\gamma_0$$

$$A_1 = \sqrt{C_1^2 + C_2^2} = \sqrt{x_0^2 + \frac{\dot{x}_0 + nx_0}{k_1}} \quad (26)$$

$$tg\gamma_0 = \frac{C_1}{C_2} = \frac{k_1 x_0}{\dot{x}_0 + nx_0} \quad (27)$$

$$\gamma_0 = arctg\left(\frac{k_1 x_0}{\dot{x}_0 + nx_0} \right) \quad (28)$$

(27)-(28) using formulas we write x_1 as follows:

$$x_1 = A_1 e^{-nt} Sin(k_1 t + \gamma_0) \quad (29)$$

The oscillation period of the machine:

$$T_1 = \frac{2\pi}{k_1} = \frac{2\pi}{\sqrt{k^2 - n^2}} \quad (30)$$

Fading decrement:

$$\delta = e^{nT_1} \quad (31)$$

Logarithmic decrement:

$$\Lambda = \ln\delta = nT_1 \quad (32)$$

$$n = \frac{\Lambda}{T_1} = \frac{\Lambda}{2\pi}\sqrt{k^2 - n^2} \Rightarrow n = \frac{\Lambda_k}{\sqrt{4\pi^2 + \Lambda^2}} \quad (33)$$

$$b = \frac{2nI_t}{y_C^2} \quad (34)$$

(18) twe look for the non-homogeneous solution of the equation x_2 in the following form:

$$x_2 = ASin(\omega_2 t + \delta - \varepsilon) \quad (35)$$

"When the system works in dynamic equilibrium, there are only forced oscillations. Let the cyclic frequency of this vibration be ω2, and the initial phase δ, the solution of the equation can be found as follows"[10; 136 – 148 –b, 11, 12]:

$$\theta = \omega_2 t + \delta$$

$$x_2 = ASin(\theta - \varepsilon); \quad \dot{x}_2 = A\omega_2 Cos(\theta - \varepsilon); \quad \ddot{x}_2 = -A\omega_2^2 Sin(\theta - \varepsilon) \tag{36}$$

$$A\omega_2^2 Sin(\theta - \varepsilon) + 2nA\omega_2 Cos(\theta - \varepsilon) + k^2 ASin(\theta - \varepsilon) = h_0 Sin(\omega_2 t + \delta) = h_0 Sin\theta$$

$$A\omega_2^2 Sin\theta Cos\varepsilon + A\omega_2^2 Cos\theta Sin\varepsilon + 2nA\omega_2 Cos\theta Cos\varepsilon + 2nA\omega_2 Sin\theta Sin\varepsilon +$$
$$+ k^2 ASin\theta Cos\varepsilon - k^2 ACos\theta Sin\varepsilon = h_0 Sin\theta$$

$$\begin{cases} -A\omega_2^2 Cos\varepsilon + 2nA\omega_2 Sin\varepsilon + k^2 ACos\varepsilon = h_0 \\ A\omega_2^2 Sin\varepsilon + 2nA\omega_2 Cos\varepsilon + k^2 ASin\varepsilon = 0 \end{cases}$$

$$1)\ (k^2 - \omega_2^2) Cos\varepsilon + 2n\omega_2 Sin\varepsilon = h_0 \tag{37}$$

$$2)\ (k^2 - \omega_2^2) Sin\varepsilon - 2n\omega_2 Cos\varepsilon = 0$$

$$(k^2 - \omega_2^2) tg\varepsilon = 2n\omega_2$$

Oscillation phase shift:

$$tg\varepsilon = \frac{2n\omega_2}{k^2 - \omega_2^2}$$

$$\varepsilon = arctg\frac{2n\omega_2}{k^2 - \omega_2^2} \tag{38}$$

(37) , add equations 1 and 2 to the square and determine the amplitude-frequency characteristic:

$$\begin{cases} A^2 (k^2 - \omega_2^2)^2 Cos^2\varepsilon + 4(k^2 - \omega_2^2) n\omega_2 Cos\varepsilon Sin\varepsilon + 4n^2\omega_2^2 Sin^2\varepsilon = h_0^2 \\ A^2 (k^2 - \omega_2^2)^2 Sin^2\varepsilon - 4(k^2 - \omega_2^2) n\omega_2 Cos\varepsilon Sin\varepsilon + 4n^2\omega_2^2 Cos^2\varepsilon = 0 \end{cases}$$

$$A^2 (k^2 - \omega_2^2)^2 + 4n^2\omega_2^2 = h_0^2$$

Basic forced vibration amplitude:

$$A = \frac{h_0}{\sqrt{(k^2 - \omega_2^2)^2 + 4n^2\omega_2^2}} \tag{39}$$

Putting formulas (38) and (39) into (37), we get the equation of the vibration of the sieve system:

$$x_2 = \frac{h_0}{\sqrt{(k^2 - \omega_2^2)^2 + 4n^2\omega_2^2}} Sin(\omega_2 t + \delta - \varepsilon) \tag{40}$$

In order to achieve the maximum vibration amplitude, the expression under the root should be the smallest:

$$f(\omega_2) = (k^2 - \omega_2^2)^2 + 4n^2\omega_2^2 \to \min \qquad (41)$$

In that case

$$\frac{df(\omega_2)}{d\omega_2} = 0 \qquad (42)$$

$$-2(k^2 - \omega_2^2)\cdot 2\omega_2 + 8n^2\omega_2^2 = 0$$

$$-k^2 + \omega_2^2 + 2n^2 = 0 - \Rightarrow \omega_2^2 = k^2 - 2n^2$$

$$\omega_2 = \sqrt{k^2 - 2n^2} \qquad (43)$$

$$f(\omega_2)_{\min} = 4n^2k^2 - 4n^4 = 4n^2(k^2 - n^2)$$

$$A_{\max} = \frac{h_0}{2n\sqrt{k^2 - n^2}} \qquad (44)$$

By reducing formulas (36) and (37) to the following form without units, we obtain the amplitude-frequency and phase-frequency characteristics of the machine:

phase-frequency characteristic

$$\varepsilon = \operatorname{arctg}\left(\frac{2\frac{n}{k}\frac{\omega_2}{k}}{1 - \frac{\omega_2^2}{k^2}}\right); \qquad (45)$$

amplitude-frequency characteristic

$$A = \frac{\frac{h_0}{k^2}}{\sqrt{\left(1 - \left(\frac{\omega_2}{k}\right)^2\right)^2 + 4\left(\frac{n}{k}\right)^2\left(\frac{\omega_2}{k}\right)^2}}. \qquad (46)$$

Results

Calculations were performed in the "Multibody Dynamics" physics interface of the "Solid mechanics" section of the "Comsol multiphysics 6.1" platform in order to justify the characteristic parameters of the core sorting machine based on the analytical expressions obtained from the above calculation equations. The machine is attached to a fixed base by means of a support with four springs, and the lower part of the support is assumed to have a spring.

Figures 3-4 below show the oscillating motion of the designed sorting machine's oscillating part. Figure 3 shows the deflection caused by the vibrational movement in the OX, OY, OZ directions when the car vibrates three times. It can be seen from Fig. 4 that the vibration makes a rotational movement relative to the moment center in the OY and OZ directions, except for the direction of the force. Also, the sieve vibrates in the vertical direction with a small amplitude of 5 mm, the sieve moves up and down. Rotational movement also occurs in the OY direction, which is perpendicular to the sieve. In this case, its amplitude is almost twice as large as compared to the OZ direction. As a result of the periodic effect of the vibrating force OX, the sieve is in a complex circular motion.

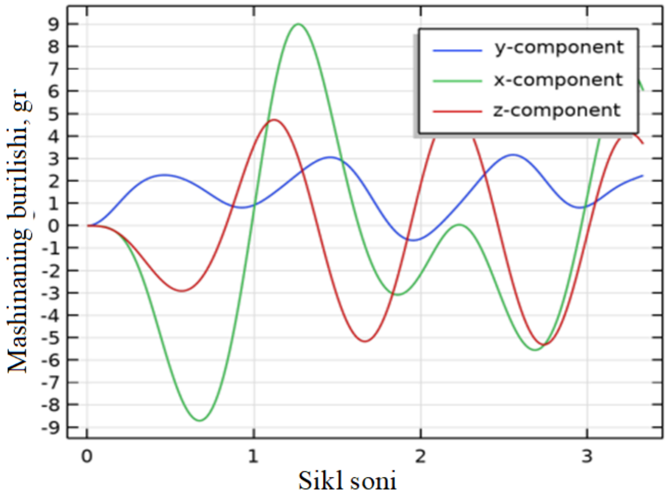

Figure 3: The Turning of the Machine Due to the Vibrational Movement in Relation to the OX, OY, OZ Directions

The character of the spatial vibration of the sieve can be seen in Fig. 4. The rapid change in the amplitude value of the sieve vibration action sharply reduces the sorting quality. Therefore, it is damped to limit its movement. It is important to consider the energy lost due to damping. Figure 5 shows the energy loss in each of the four springs in the machine under consideration.

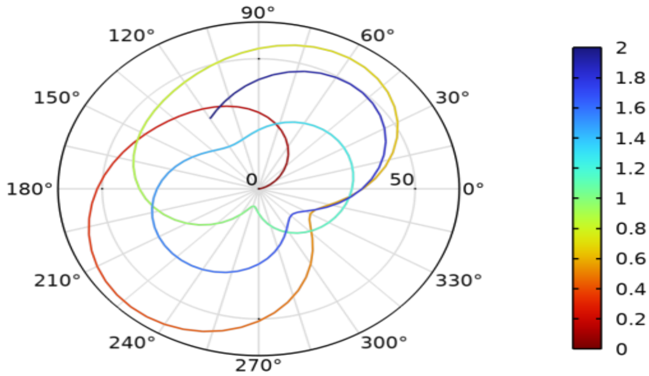

Figure 4:Spatial Oscillating Motion of Sieve

It shows the condition of the sieve vibrating three times. It can be seen from Figure 5 that the vibration of energy is the least in the spring located at the end of the machine screen, and the vibration level is greater in springs 3, 4 on the side of the hopper. Because it is given by this. If we pay attention to the degree of deformation, the most displacement is observed in springs 3, 4, as above (see Fig. 5). Figure 5 presents undamped vibration. It can be seen from the picture that the displacement amplitude changes up to 42 mm according to the uniformity of the spring. The smallest displacement will be around 12 mm. These are used to determine the level of damping.

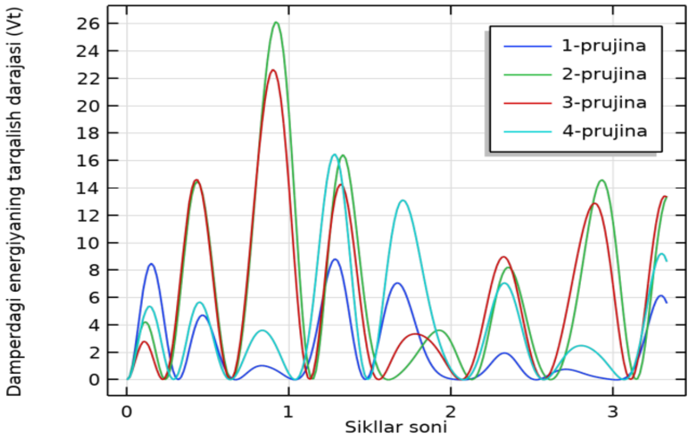

Figure 5: Energy Lost in Damping

It is known that the amplitude of the vibration process, that is, the displacement, is integrally connected to the uniformity of the spring. The dependence of the cyclic frequency of the sieve vibration obtained on the basis of formulas 22 and 43 on the stiffness of the spring in the machine is presented (see Fig. 7). From this graph calculated as a result of the analytical analysis carried out on the machine, a spring corresponding to the vibration frequency is selected.

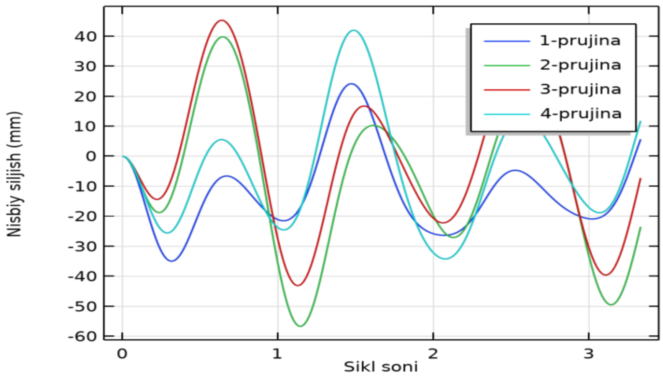

Figure 6: Displacement of the Springs Relative to the Equilibrium Position

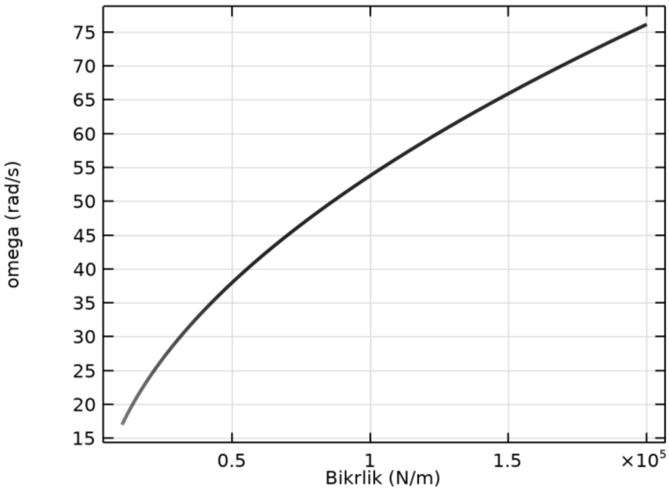

Figure 7: Dependence of the Cyclic Frequency of the Sieve Vibration on the Stiffness of the Spring in the Machine

The damping level of the sieve vibration at this frequency, determined on the basis of the external parameters of the sieve given in Table 1, can be seen in Fig. 8. The external harmonic force and the level of damping depend on the damping decrement of this vibration. The vibration graph of the sieve calculated on the basis of the above parameters of the machine is presented in Fig. 8.

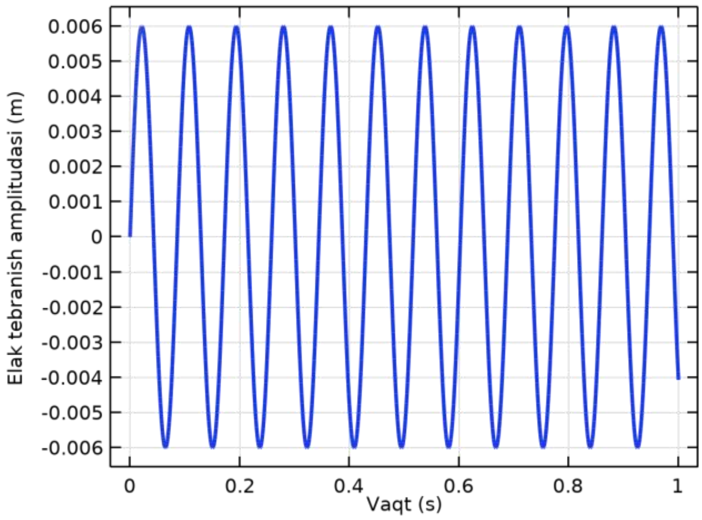

Figure 8:Vibration Characteristics of the Sieve

Summary

Most electric motors operate at frequencies of 1000 rpm and above. It is necessary to use reducers or pulleys to use them in sorting machines operating at low frequency. In this case, additional energy is lost in pulleys or reducers, as well as technical inconvenience, and the cost increases. Therefore, in order not to use such a multi-cascade pulley or reducer, it is planned to develop a machine that works at a frequency of 750 rpm. The cyclic frequency of the machine operating at this frequency is 73.26 rad/s. The spring tension corresponding to this frequency should be 200,000 N/m. If we pay attention to Fig. 8, in the graph, the longitudinal vibration amplitude of the sieve in the horizontal plane along the OX axis was 12 mm. This quantity is equal to the projection of the position of the sieve hole at an angle α with respect to the horizontal direction.

As a result of the calculated formulas and graphs, the length of the fine sieve is 0.2 m, the length of the 1/8 part sieve is 0.5 m, the length of the 1/4 part sieve is 0.5 m, the full length of the sieve is 1.2 m, the elevation angle of the sieve is 0.34907 rad, the width of the sieve is 0.6 m, the number of holes in the smallest cleaning sieve is 2857.1 pieces, the number of holes in the 1/8 part is 2083.3 pieces, the diameter of the holes in the 1/8 part is 0.01 pieces, the cleaning part, the diameter of the holes in the sieve is 0.005 m, the 1/4 part is the hole diameter in the sieve is 0.015 m, 1/ It was determined that the number of holes in the 4th sieve is 980.39 pieces, the profile metal thickness is 0.002 m, and the full mass of the sieve is 19.163 kg.

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