Analysis of the beats of separation sieve pans

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Abstract. To ensure high productivity of grain production reliable operation of grain cleaning equipment is required. The process of cleaning seed material is connected to significant vibration loads some of which have an impact on the separation process and the operational reliability of the machine components. To combat harmful vibrations a theoretical analysis of the beats occurring during the sieve separation of grain was made. The law of motion of the mass center of the sieve pan is derived as a function from time over a period of oscillating. The analytical solution of the obtained equation allows determining the horizontal and vertical displacements of the mass center at every moment of time. Based on this motion, accelerations and related inertial forces are determined. These forces take on sufficiently large values, which lead to beats of the indicated separation devices. To determine the forces leading to these beats, using the d'Alembert's principle, equilibrium equations are compiled, the solution of which allows determining the total horizontal force and reaction force in the suspension. The equations of equilibrium include the forces of tension on bending of the suspension rods. It has been established that a rational choice of the geometric characteristics of these suspensions can significantly reduce the forces acting on the drive of the sieve pan and, as a result, on the horizontal and vertical runouts.

1. Introduction
Grain production in our country is constantly increasing its volumes. It results from the competent selection of seed material, the use of chemicals that stimulate the growth and development of plants, organic and mineral fertilizers, as well as the continuous improvement of the technical equipment of the technological line for the production of grain crops. The post-harvest treatment based on various separation machines is important for the quality of the harvested material. Sieve pans with a set of flat perforated sheets are used as the main working body in the majority of grain cleaning machines [1]. Their perforation is made in the form of various geometric holes of different sizes and locations. The work of the suspended on flat metal suspensions housing with sieves is carried out by the drive mechanism [2, 3]. This working process is a reciprocating movement of a given amplitude and frequency. At the same time the machine parts experience serious alternating loads, which lead to malfunction of the entire unit. The elimination of this problem is highly relevant for designers and consumers of grain cleaning equipment [4].

2. Theoretical analysis of the separation sieve pans
The operation of the grain cleaning machine is closely related to the work of flat sieves located in the sieve pan. The work efficiency depends on a large number of factors [5, 6]. While the influence of
regime parameters are the most studied at the moment, the effect of vibration transmitted from the sieve pan to the machine remains largely unexplored. Such impact is only negative in nature, since, firstly, it destroys the components of the machine, and, secondly, it is transmitted through the bottom to the operator’s working place negatively affecting his health.

In order to eliminate this problem it is necessary to study the mechanics of the working process and establish the nature of the forces affecting the formation of harmful vibrations.

The separation sieve pan suspended on two pairs of elastic suspensions oscillates due to an eccentric of radius R, m centered at point D. Moreover, it performs translational movement along a circle of radius L = |OA|, m, equal to the length of the suspensions. When the eccentric is rotated through an angle \( \alpha = \omega t \), where \( \omega \) is the angular velocity of rotation, c-1, point C moves to point M, and point A moves to point B. Then \( |OA| = |OB| = L \), the radius of the eccentric \( |DC| = |DM| = R \), connecting rod length \( |AC| = |BM| = S \), m [7].

Let \((x_1, y_1)\) be the coordinates of M, and let \((x, y)\) be the coordinates of B. Then \( x^2 + (L - y)^2 = L^2 \) whence \( y = L - (L^2 - x^2)^{1/2} \).

**Figure 1.** Diagram of the drive mechanism of the sieve pan.

The coordinates \( x_1 \) and \( y_1 \) are determined by the formulas: \( x_1 = \sqrt{S^2 - (h - R)^2 + R \cdot \sin \omega t} \), \( y_1 = R \cdot \cos \omega t - h \). Using the coordinates of points B and M, we’ll determine the length of the segment BM.

\[
((S^2 - (h - R)^2)^{1/2} + R \cdot \sin \omega t - x)^2 + (R \cdot \cos \omega t - h - y)^2 = S^2,
\]

\[
((S^2 - (h - R)^2)^{1/2} + R \cdot \sin \omega t - x)^2 + (R \cdot \cos \omega t - h - L + (L^2 - x^2)^{1/2})^2 = S^2 \tag{1}
\]

The solution of this equation allows determining the dependence of the \( x \) coordinate of point B on time, that is, the horizontal movement of the sieve pan \( x(t) \). Vertical movements in accordance with Fig. 1 are determined by the function \( y(t) = L - (L^2 - x(t)^2)^{1/2} \). The analytical solution of equation (1) even with numerical values of the input parameters is so cumbersome that we can not give it in this article. The dependences of the horizontal and vertical displacements on the angle of rotation of the eccentric at \( L = 0.7 \) m, \( S = 0.6 \) m, \( h = 0.065 \) m, \( R = 0.03 \) m, \( \omega = 35 \) c^{-1} are shown in Fig. 2. As can be seen from this graph, the asymmetry of the movement of the sieve pan is observed.
Differentiating the functions $x(t)$ and $y(t)$ twice in time, we’ll find the horizontal and vertical accelerations of the sieve pan $a_x(t)$ and $a_y(t)$. The dependence of the magnitude of these accelerations on the angle of rotation of the eccentric is shown in Fig. 3. As we can see from this graph, horizontal accelerations are especially large. Their maximum and minimum values are 38.6 and -35.3 m/s$^2$, respectively. Obviously, they cause large inertial forces causing beats of sieve pans. Due to the fact that the line of action of the drive connecting rod does not pass through the mass center of the sieve pan $E_1$, the moments arising during operation also cause significant vertical beats.

Applying the d’Alembert’s principle, we’ll write down the equilibrium equations of the sieve pan taking into account the inertial forces (Fig. 1).

\[
\begin{align*}
-(N_1 + N_2) \cdot \sin \gamma + F_x - F_{up} + T \cdot \cos \beta &= 0 \\
(N_1 + N_2) \cdot \cos \gamma + F_y - G - T \cdot \sin \beta &= 0 \\
\left[ (N_1 + N_2) \cdot \sin \gamma + F_{up} \right] \cdot h_1 + (N_2 - N_1) \cdot \cos \gamma \cdot b + T \cdot \cos \beta \cdot h_1 &= 0
\end{align*}
\]
Here $N_1$ and $N_2$ are reactions of two pairs of suspension rods; $F_x = ma_x(t)$ and $F_y = ma_y(t)$ are the horizontal and vertical inertial forces, respectively, where $m$ is the mass of the sieve pan, kg; $T$-rod reaction force; $G = mg$ — pan weight, $g$ — gravity acceleration, m / s$^2$; $F_{\text{фр}}$ is the total bending strength of four rods; $h_i$ - half the height of the mill, $b$ - the distance from the rods to the point $E_1$ (mass center).

The solution to this system is:

$$N_1 = \frac{1}{2} \left[ 2 \cdot F_{\text{уп}} \cdot \cos \gamma \cdot h_1 + F_{\text{уп}} \cdot \cos \gamma \cdot \sin \beta \cdot b - F_x \cdot \cos \gamma \cdot \sin \beta \cdot h_1 - F_x \cdot \cos \gamma \cdot h_1 + G \cdot \cos \gamma \cdot \cos \beta \cdot b + G \cdot \sin \gamma \cdot \cos \beta \cdot h_1 \right] \cdot [\cos \gamma \cdot b (\cos \beta \cdot \cos \gamma - \sin \gamma \cdot \sin \beta)]^{-1}$$

$$N_2 = -\frac{1}{2} \left[ 2 \cdot F_{\text{уп}} \cdot \cos \gamma \cdot h_1 - F_{\text{уп}} \cdot \cos \gamma \cdot \sin \beta \cdot b - F_x \cdot \cos \gamma \cdot h_1 + F_x \cdot \cos \gamma \cdot \cos \beta \cdot b + F_y \cdot \sin \gamma \cdot \cos \beta \cdot h_1 - G \cdot \cos \gamma \cdot \cos \beta \cdot b + 2G \cdot \sin \gamma \cdot \cos \beta \cdot h_1 \right] \cdot [\cos \gamma \cdot b (\cos \beta \cdot \cos \gamma - \sin \gamma \cdot \sin \beta)]^{-1}$$

The total lateral force is determined by the formula:

$$BOK = T \cdot \cos \beta = \frac{[F_{\text{уп}} \cdot \cos \gamma - F_x \cdot \cos \gamma - F_y \cdot \sin \gamma + G \cdot \sin \gamma] \cdot \cos \beta}{\cos \beta \cdot \cos \gamma - \sin \gamma \cdot \sin \beta}$$

According to Fig. 1, $\sin \gamma = x(t)/L$, $\cos \gamma = (1 - \sin^2 \gamma)^{1/2}$, $\sin \beta = (h + y(t) - R \cdot \cos \omega t)/S$, $\cos \beta = (1 - \sin^2 \beta)^{1/2}$

Due to the rigid fastening of the suspension rods to the pan and the body, when they bend, an alternating horizontal force of tension arises. It is determined by the well-known [7] formula $F_{\text{уп}}(t) = \frac{-3EJx(t)}{I^3}$, where $E$ is the elastic modulus of the rod material, $P$. For steel $E = 2.1 \times 10^{11}$ Pa. $I$ is the moment of inertia of the cross section of the rod, m$^4$.

In the case of a rectangular section of the rods, the moment of inertia is equal to:

$$I = H \cdot a^3 \cdot 12^{-1},$$

where $a$ and $H$ are respectively the width and length of the cross section of the rod, m

Without taking into account the forces of tension at $h_1 = 0.1$ m and the mass of the sieve pan $m = 100$ kg, the graph of lateral force is shown in Fig. 4. As can be seen from this graph, during fluctuations, the lateral force reaches very large values - more than 3500 N. In turn, due to the arising moments, significant reactions appear in the suspension rods, causing vertical beats. In this case the total force in the suspension $N_1 + N_2$ varies from 390 N to 1360 N.

**Figure 4.** Dependence of lateral force on the angle of rotation of the eccentric excluding force of tension.
With a fixed length of the cross section of the suspension rods $H = 0.05$ m, an increase in the width of the cross section leads to an increase in the force of tension. Its direction is opposite to the lateral inertia, which reduces the total lateral force and, as a consequence, horizontal and vertical runouts. The table shows the maximum and minimum values of lateral force at various values of $a$.

| $a$, m | $BOK_{\text{max}}, H$ | $BOK_{\text{min}}, H$ |
|-------|----------------|----------------|
| 0     | 3800           | -3500          |
| 0.01  | 3000           | -2500          |
| 0.012 | 2350           | -1785          |
| 0.014 | 1500           | -750           |
| 0.015 | 975            | -135           |
| 0.016 | 625            | 210            |
| 0.017 | 1450           | -320           |
| 0.018 | 2370           | -1100          |

From the table that, at $a = 0.016$ m, the lateral forces have the lowest modulus value, which is six times less than these forces in the case of an inelastic suspension. This is reflected in the total force in the suspension, which ranges from 870 N to 1180 N, as shown in the graph in Fig. 5. Since in the absence of inertial forces and moments the total force in the suspension is equal to 1000 N. At this value a vertical runouts are not more than 180 N. Thus, with the correct choice of the geometry of the rods, both horizontal and vertical runouts of separation sieve pans can be significantly reduced.

**Figure 5.** Dependence of the total force in the suspension on the angle of rotation of the eccentric

### 3. Conclusion

The theoretical studies and the obtained analytical dependences allow determining the horizontal and vertical displacements of the mass center of the sieve pan at every moment of time, acceleration and inertial force. The correct choice of the geometric characteristics of the suspension of the sieve pan and the corresponding material for their manufacture will significantly reduce the forces acting on its drive and the degree of vibration. The obtained results can be used in the designing of new grain cleaning machines for post-harvest processing installed in the technological lines of grain cleaning units, as well as in the calculation of structural-operational parameters of the sieve pans.

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