THEORETICAL CONSIDERATIONS ON CHOOSING THE VIBRATION METHOD AND SPEED OF ELECTROVIBRATOR FOR MILLING MACHINES

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ABSTRACT

The simplification of the kinematic chain for the equipment in the milling industry has been an ongoing concern for the manufacturers in this field.
Thus, the aim was to replace the eccentric drive with electric vibrators. The use of electric vibrators to drive the equipment used in the technological processes of cleaning leads to a number of advantages compared to eccentric drive: simplifying the kinematic chain, intensifying the separation process, reducing stress on the foundation. The use of electric vibrators in grain cleaning machines is increasingly common worldwide in companies with a tradition in the field. Therefore, the choice of the vibration method and the required number of vibrations per minute capable of achieving the best efficiency is an important issue that will be addressed in this paper.

INTRODUCTION

The removal of foreign bodies, before the cereals are subjected to transformations to obtain the finished product, is a complex and important operation for further processing.

The machines used for this purpose have various constructive features, due to the characteristics of foreign bodies, which indicate the difference from the seeds of the basic culture.

Certain characteristics (magnetic properties, dimensions, specific mass etc.) lead to the use of certain separation methods (magnetic separation, separation by aerodynamic properties, sieving, etc.).

The working process for pre-cleaning and cleaning machine is based on the combined action of:

• successive sieving on superimposed sieves with holes of different shapes and sizes, suitable for the intended purpose and the species being processed, for the fractional separation by width or thickness of the various components;

• the kinematic regime of the machine (crank-connecting rod or electric vibrator drive);

• aspiration of light components and their evacuation in fractions depending on their own floating speed;

• retention of ferromagnetic components by means of permanent magnets placed at the entrance or exit of the product from the machine (Jinescu, 1989).

Sieves are subsystems of machines intended for the separation of granular and pulverulent materials as well as the solid phases of the pulps (germs) in products of different granulations with the help of sieving surfaces with calibrated holes. From the point of view of the kinematic regime, the sieves within the pre-cleaning and cleaning machines in the milling field are oscillating sieves and vibrating sieves.

The combined classifier aspirator TC 600 is a modern machine, intended for the pre-cleaning or cleaning of both straw cereals (wheat, barley, rye) and corn, legumes and oilseeds, species with a large share in the overall agricultural production obtained annually in small and medium farms, operated by two electric vibrators.

In line with the most modern trends, incorporating state-of-the-art constructive solutions, the combined classifier aspirator, as its name suggests, uses two combined principles in the pre-cleaning process of grain seeds: sieving with the aspiration of the product to be processed in counter-flow (Păun, 2003).

MATERIAL AND METHOD

The specific objectives that marked the proposed research were:

Sieve operating ways

In the case of cleaning and sorting machines used in production, the sieve frame can be operated:

• by classic crank- connecting rod system (oscillating sieves) (fig. 1);

• by vibration generators - electrovibrators (vibrating sieves) (fig. 2).

The classic system involves the suspension of the sieve frame by using leaf springs made of wood or steel, embedded at both ends. In general, the inclination of the suspension springs negatively influences the separation coefficient. However, at low values of the inclination angle, this influence can be neglected. Therefore, in the case of sorting seeds with a humidity and a percentage of impurities within the normal limits specified in the specialized standards, it is recommended to have an inclination of the springs between 10˚-12˚, at an inclination of the sieve frame of 4˚ to the horizontal, situation that does not influence the separation process (Brăcăcescu et al., 2014).

In modern machines, with vibration generators, the plane - parallel motion induced by them gives the machine a high efficiency and an increased separation efficiency. In these, the leaf springs were replaced with helical or rubber springs, thus removing the negative influence of the classical springs' inclination on the separation (Munteanu, 1986).

Fig. 1. Separator cleaner with brushes SAP 815

Fig. 2. Combined classifier aspirator TC 600

Kinematic regime of oscillating sieves

It is considered that (fig. 3): the oscillation direction is perpendicular to the leaf springs, inclined at an angle α to the horizontal; the length of the crank is very small compared to that of the connecting rod ($\frac{l}{L} << 1$).

Equation of motion:

$$s = l\cos\omega t = A\cos\omega t \qquad (1)$$

Differential equation of motion:

$$m\ddot{s} + c\dot{s} + ks = F \quad (2)$$

where:

m = total mass of the sieve frame, [kg];

s = oscillation elongation, [m];

c = overall damping coefficient in the s direction of the oscillation [kg/s];

k = overall elastic constant of the elastic elements [kg/s^2];

α = angle to the horizontal of the oscillation or vibration direction, [degrees];

ω = pulsation of the drive element, [s^{-1}];

m_0 = mass of vibration generator counterweights [kg];

F - the force that appears in the connecting joint of the connecting rod with the working part [N];

$$F = F_0 \cos(\omega t + \varphi) \quad (3)$$

where:

F_0 - the disturbing force generated by unbalanced masses, [N]:

$$F_0 = m_0 r \omega^2 \quad (4)$$

The following notations are made:

- own pulsation [s^{-1}]

$$p = \sqrt{\frac{k}{m}} \quad (5)$$

- dimensionless drive pulsation: $\gamma = \frac{\omega}{p}$

where the pulsation of the drive element [s^{-1}] is: $\omega = \frac{\pi n}{30}$, (6)

- dimensionless damping coefficient

$$\zeta = \frac{c}{2mp} \quad (7)$$

- static value of the amplitude of the force F, force required to start [N]:

$$F_{0s} = mlp^2 \quad (8)$$

At the resonance frequency:

$$\gamma_r = \sqrt{1 - 2\zeta^2} \quad (9)$$

the amplitude of the force will have a minimum:

$$F_{0\min} = 2F_{0s}\sqrt{1 - \zeta} \quad (10)$$

The average power output will be:

$$N_{med} = \frac{1}{2} A^2 c \omega^2 \quad (11)$$

and for the resonance frequency:

$$N_r = ml^2 p^4 \zeta^2 (1 - 2\zeta) \quad (12)$$

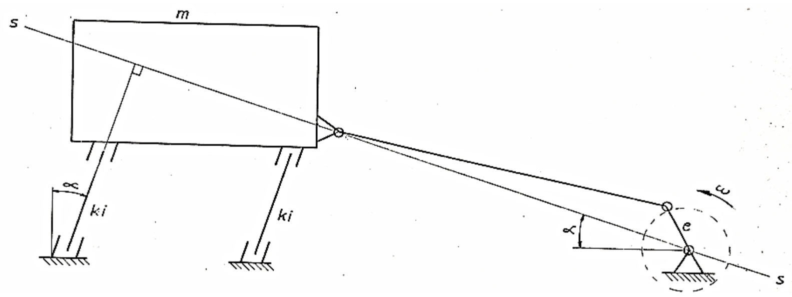

Fig. 3. Kinematic regime of oscillating sieves

Dynamic regime of vibrating sieves

Consider the case of a monomassic vibrating machine (fig.4), driven by vibration generators directed by unbalanced masses that rotate in opposite directions. The direction of vibration is perpendicular to the supporting springs and inclined at an angle α to the horizontal and passes through the mass centre of the product-loaded frame. It is considered that the drive motors ensure a sufficiently fast passage through the resonance range and a constant speed. The continuous and uniform feeding of the sieve is ensured, which leads to the constancy of the mass of the product that is on the feeding surface at any moment of time.

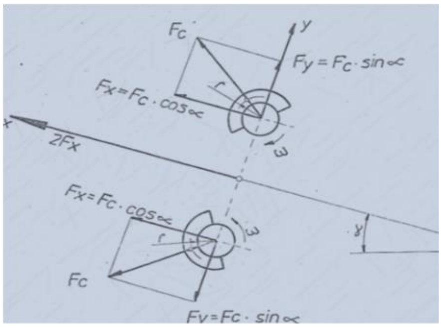

Fig. 4 - Dynamic regime of vibrating sieves

The differential equation of motion is:

$$(m+m_0)\ddot{s}+c\dot{s}+ks=F_0\cos\omega t \qquad (13)$$

where:

m = total mass of the feeding surface [kg]

m_0 = mass of vibration generator counterweights [kg]

s = vibration elongation [m]

c = overall vibration damping coefficient [kg/s]

k = overall elastic constant of the elastic elements [kg/s^2]

F_0 = disturbing force generated by unbalanced masses [N]

$$F_0 = m_0 r\omega^2 \qquad (14)$$

r = radius of mass centre of the vibration generator counterweights [m]

ω = pulsation of the drive element [s^{-1}]

$$\omega = \frac{.\pi n}{30} \qquad (15)$$

n = speed of the drive element [rpm]

α = angle to the horizontal of the vibration direction [º]

i = angle of incidence i=90- α

The following notations are made:

- own pulsation

$$p = \sqrt{\frac{k}{(m+m_0)}}\ [\mathrm{s}^{-1}] \qquad (16)$$

- own damping coefficient

$$h = \frac{c}{2(m+m_0)}\ [\mathrm{s}^{-1}] \qquad (17)$$

The differential equation becomes:

$$\ddot{s}+2h\dot{s}+p^2 s = \frac{F_0}{m+m_0}\cos\omega t \qquad (18)$$

The stationary solution of the equation will be:

$$s = A\cos(\omega t-\varphi) \qquad (19)$$

where:

$$A = \frac{m_0}{m+m_0} r \frac{\omega^2}{\sqrt{(p^2-\omega^2)^2+4h^2\omega^2}} \qquad (20)$$

The following notations are made:

- dimensionless damping coefficient

$$\zeta = \frac{c}{2(m+m_0)p} = \frac{h}{p} \qquad (21)$$

- dimensionless drive pulsation

$$\gamma = \frac{\omega}{p} \qquad (22)$$

- dimensionless mass coefficient

$$\varepsilon = \frac{m_0}{m+m_0} \qquad (23)$$

- dimensionless amplitude coefficient

$$\xi_A = \frac{A}{\varepsilon_r} \qquad (24)$$

Using the above notations, we obtain:

$$\xi = \xi_A\cos(\omega t-\varphi) \qquad (25)$$

with $\xi_A = \dfrac{\gamma^2}{\sqrt{(1-\gamma^2)^2+4\zeta^2\gamma^2}}$ (26)

and $\xi_{A\max} = \dfrac{1}{2\zeta\sqrt{1-\zeta^2}}$ (27)

From the study of the function **$\xi_A(\gamma)$,** it results that **ξ_A** stabilizes tending towards the value 1. It is recommended to avoid the influences of higher-order harmonics of vibration (Piven, 2019).

Systems that use the free oscillation vibration technique include two vibration methods (Darie, 1986): rotary method (fig. 5) and unidirectional method (fig.6).

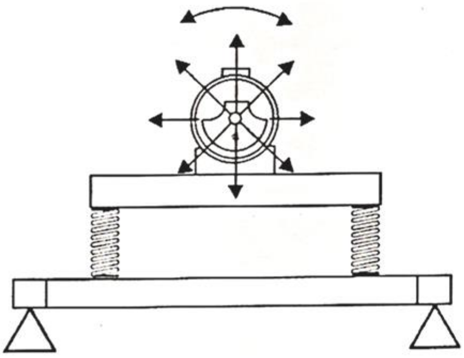

Fig.5. Rotary method

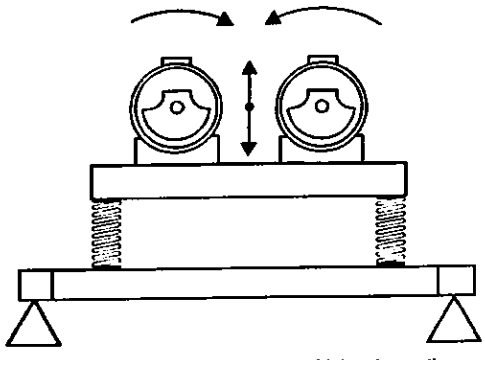

Fig. 6. Unidirectional method

Rotary method - The vibrating force is driven in all directions on 360º, either clockwise or counterclockwise (fig. 5). The rotary method is obtained using a single electrovibrator.

In the case of this method, two theoretical speeds of movement of the product are established:

V_{tp} = theoretical speed of the product [m/h];

V_{tpc} = corrected theoretical speed of the product depending on the inclination of the sieve frame [m/h]

$$V_{tpc} = (V_{tp} + V_i) / F_\alpha \quad (28)$$

where:

α = angle to the horizontal of the vibration direction [º];

i = angle of incidence i=90- α;

V_i = incidence rate [m/h];

F_α = correction factor for sieve frame inclination angle.

The movement of the product in the case of this method is shown in figure 7.

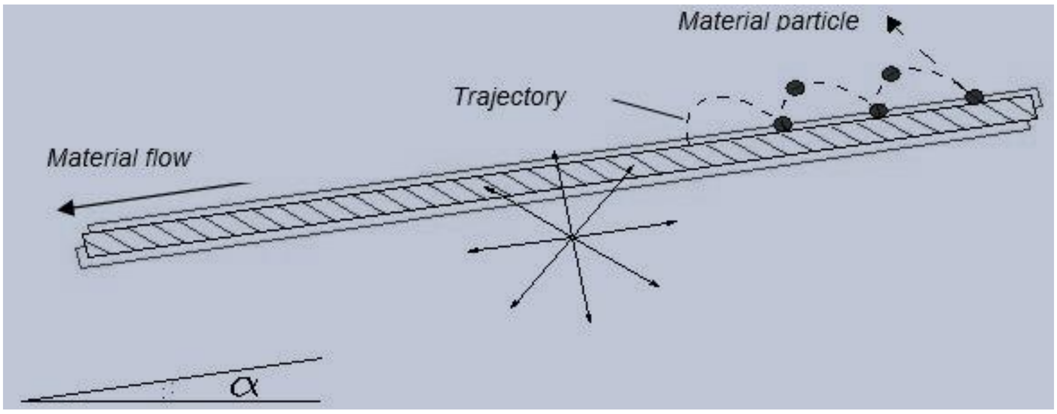

Fig. 7. Movement of particles on inclined surfaces in the case of rotary method

Unidirectional method - The vibrating force is driven in one direction only.

This method is obtained using two electrovibrators with the same electro-mechanical characteristics that rotate in opposite directions. The movement of the product in the case of this method is shown in figure 8.

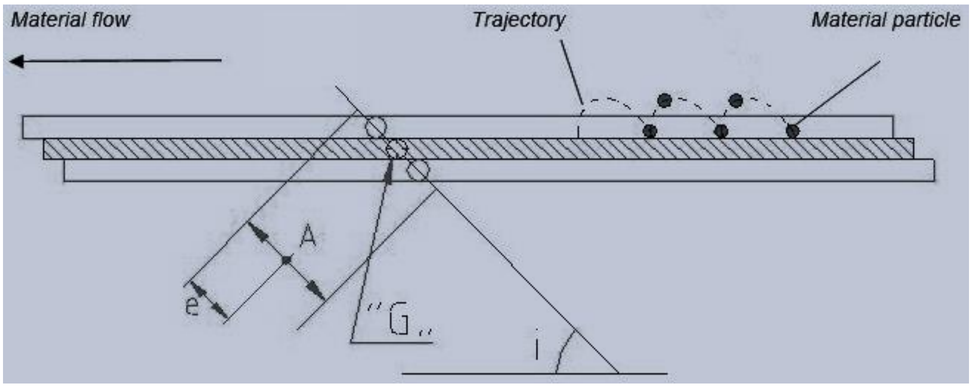

Fig. 8. Movement of particles in the case of unidirectional method

i = angle of incidence i=90 - α, [º]
e = eccentricity [mm];
A = amplitude, A= 2 x e [mm].

In the case of the unidirectional method, the angle of incidence "i" must be taken into account regardless of the support position of the frame on elastic elements (helical springs). It is very important for this method that the force line for any angle of incidence passes through the centre of gravity "G" of the machine elastically suspended (with springs).

The determination of the angle of incidence of the force line depends on the type of process and must be within the indicated range (e.g. from 6º to 12º for machines in the milling industry, from 25º to 30º for transport, unloading, sorting, etc. machines).

The manufacturer of electrovibrators indicates in the product catalogues, for each vibrator separately, a series of parameters that constitute the input data:

- constructive elements, overall dimensions, mass;
- speed, n [rpm]
- centrifugal force, F_{cf} [N]
- drive moment, M_{mot} [Nm]
- power, N [kW]
- electric motor power (power consumed), [kW].

Based on these input data one determines:

- the vibration isolation index, ensured by the elastic elements:

$$I_i = 1 - \frac{1}{f_s\left(\frac{n}{950}\right)^2 - 1}$$, with $0<I_i<1$ (29)

where f_s= static arrow (elastic elements' arrow under the influence of the frame weight, the vibrator and the product on the sieve at a given moment) [mm]

For the value (considered satisfactory) I_i=0,95 the elastic elements can be dimensioned so that:

- vibration amplitude:

$$A = \frac{1}{2}\frac{M_{mot}}{n} \quad (30)$$

and $A \le \frac{26{,}9}{2}\left(\frac{1000}{n}\right)^2$ (31)

- power required:

$$N_{nec} = \frac{AF_c n}{7800g} \quad (32)$$

where:
N= vibrator power value

- eccentric mass (unbalanced):

$$m_0 = 91{,}19\frac{F_c}{rn^2} = \frac{Fc}{r\omega^2} \quad (33)$$

Between M_{mot} and F_c there are the relation

$$M_{mot} = \frac{F_C}{0{,}56\left(\frac{n}{1000}\right)^2 \cdot g} \quad (34)$$

- total force transmitted to the foundation

$$F_{trans} = (1 - I_i)F_c \quad (35)$$

- the force exerted by the vibrator on the frame is calculated with the relation:

$$P = P_0\cos(\omega t - \theta) \quad (36)$$

where:

$$P_0 = F_0 f \quad (37)$$

(*f* = dimensionless factor of transmissibility of the disturbing force to the frame).

RESULTS AND DISCUSSIONS

The choice of vibration method and the required number of vibrations per minute capable of achieving the best efficiency for each type of process depends on the specific weight and granulometry of the product subjected to processing.

Electrovibrator manufacturers draw up diagrams that take into account the eccentricity, amplitude, angle of incidence and theoretical speed of movement of the product. With the help of these diagrams, the vibrator suitable for the type of process is chosen.

Regardless of the method chosen, the electrovibrators can be mounted on the machine elastically suspended. From those presented above, it can be concluded:

- the drive pulsation doesn't have a major influence on the resulting amplitude (in case of operation by electrovibrators);
- the resulting amplitude is decisively influenced by the mass of the eccentric and its location radius (especially for oscillating sieves operated by crank-connecting rod mechanism);
- the vibration generator exerts a disturbing force directed rectilinearly to the sieve frame;
- the force exerted by the unbalanced mass shaft on the sieve frame is a function of the radius at which the mass centres of the unbalanced masses are located and of the drive element pulsation.

CONCLUSIONS

The use of electrovibrators to drive the machines used in technological cleaning processes leads to a number of advantages compared to eccentric drive:

• simplification of the kinematic chain, implying the removal of the following disturbing factors:

- to the producer:

-many and high precision parts;

-fastidious and difficult to control assembly;

-protection structures often anaesthetic

- to the user:

- rigorous maintenance;
- high repair costs;
- low reliability;
- mechanical shocks in the system, in particular in the event of wear

• intensification of the separation process;

• reduction of loads to the foundation;

• allowing the variation of the centrifugal force, this being achieved by changing the reciprocal position of the unbalanced masses mounted on the rotor extremities;

• electrovibrators occupy a minimum volume, which facilitates their mounting on the active parts of the machine so that the direction of vibration passes through the mass centre of the sieve shaker loaded with material;

• the use of this type of electrovibrators allows changing the direction of the disturbing force at any time.

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