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CONSIDERATIONS ON KINEMATICS AND DYNAMICS OF SHAKING SIEVES ELECTROMAGNETICALLY DRIVEN

Carmen Brăcăcescu[1], Simion Popescu[2]

[1]*Transilvania* University of Brasov/Romania; simipop38@yahoo.com
[2]National Insitute for Agricultural Machinery (INMA), Bucharest/Romania; carmenbraca@yahoo.com

Abstract: *The paper presents the diagram and the operational principle of electromagnet driven vibrating feeders and the constructive and functional factors influencing the working parameters of this type of equipment. Diagrams are presented of automatic amplitude and frequency control systems of the vibrations generated by the studied electromagnetic device, in view of its utilization for the drive of vibrating feeders for bulk solid material of varied characteristics and properties*.
Keywords: *vibrating feeder, electromagnetic vibrator, elastic vibrating system, vibration amplitude control, vibration frequency control*

1. INTRODUCTION

Vibrating sieves are used as vibrating feeders at technological primary processing installations for cereals. The vibrating feeders are made of a vibrating chute (channels) that are fixed elastically on stands that receive oscillating movements from rod-crank mechanisms (fig. 1,a) or from vibrating units with eccentric masses (fig.1,b) or from electromagnets (fig.1,c) [1,2], Magnetic drive systems (Fig. 1,c), which are used as free vibrating or with steering suspension features, make use of the operation at resonance conditions with a safer excitation energy and leading to very compact units.

Vibration sieves feeders consist of a conveying device which carries the bulk solid layer and the drive unit for the vibration excitation supported by a spring suspension. The function of vibration feeders is based on the *micro-cast* effect. The vibration agitation is usually induced by a flute vibrating inclined by 20…45 ^{0} to the horizontal. The particles are accelerated starting from definite vibration frequency /amplitude; these acceleration conditions are used to execute a parabolic ejection motion inclined upwards and strike the flute again after an adequate displacement.

This effect (micro-cast) is repeatedly induced at the exciting frequency with the result that the bulk solid layer which is pre-adjusted by the outlet clearance of the feeder is discharging quasisteadily.

The necessary excitation condition for the micro-cast effect to develop is that the vertical upwards directed acceleration induced by the flute vibrations on the particles is larger than the gravitational acceleration (vertical downwards). Important parameter influences on this displacement process are the vibration frequency and amplitude, the angle between the excitation and the flute axis and the bulk solid properties such as particle size, distribution and shape as well as friction.

2. MATERIAL AND METHODS

On a world level the companies having a long tradition in this field use the electrovibrator for cereal primary processing equipment. In order to operate the shaking sieves designed to drive the technical equipment used in technological primary processing of agricultural products, the electromagnetic vibrators (named electrovibrators), manufactured by specialized companies can be used as single elements or pair elements, generating the sieve frame vibration

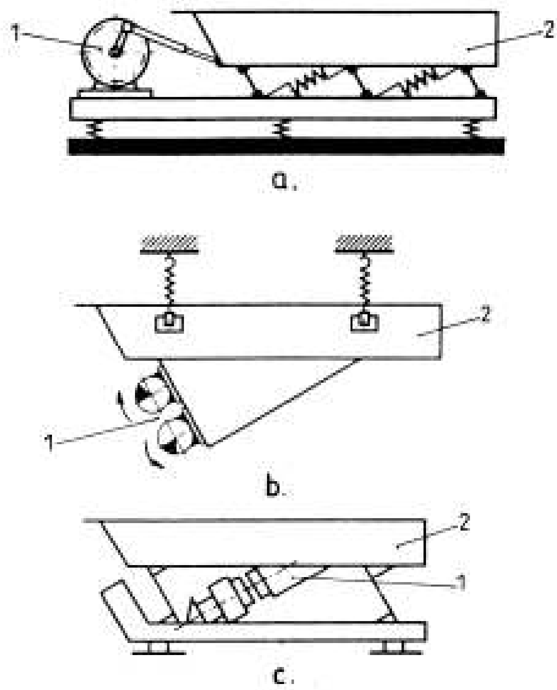

Figure 1. Basic types of vibrating feeders: a-with rod-cran mechanism; b-with eccentric mass; c- with electromagnetic vibrator: 1-vibration generating mechanism; 2- vibrating mass with transporting chute.

By comparing with vibrating sieves driven by eccentric mass, the electrovibrators use leads to many advantages: simplifies the kinematic chain, by eliminating the following negative factors: many specific parts, assembling difficult and laboriously to control, inesthetic guards, rigorous maintenance, big reparation costs, reduced reliability, mechanic shocks (especially when wear appears); intensifies the separating process; diminishes the stress transmitted to foundation; their volume is minimum, which enables their mounting on active parts of equipment so that the vibration direction passes through the mass center of sieves shaker loaded with material (or by its immediate proximity); allows the modification of disturbing force direction any time. The electromagnetic vibrators advantages consists in absence of friction and rotating parts when the productivity is regulated, as disadvantages, we can mention the small amplitude (0,5….2mm) which excludes the transport of powder loads, as well as the transport little length of vibrator (up to 2,5…6m).

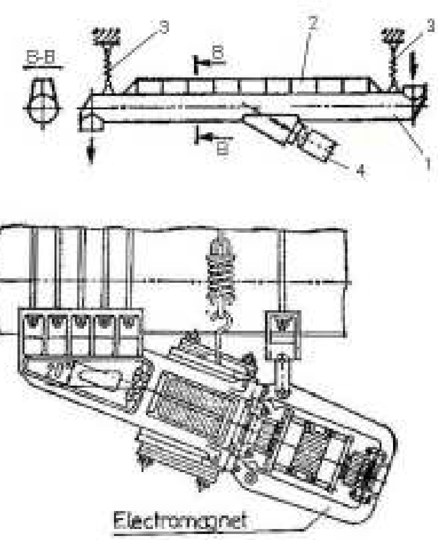

Figure 2. Vibrating feeder electromagnetically driven

The vibrating sieve electromagnetically driven comprises the frame (fig.2), freely hung on elastic elements, frame which receives oscillations transmitted by electromagnetic vibrator 8. The oscillating frame is supported by a rigid longitudinal beam 7. The electromagnetic vibrator can be with simple or double drive, with mobile (indus) and reactive (inductor) part. Being endowed with small mass and power (up to 1kW), the electromagnetic vibrators simply acted can be used to light conveyors. The connecting support of vibrating feeder bear (hang) the auger and ensure the oscillations according to dynamic calculation. There are used single sheet (lamella) springs or many sheets springs. The sheet crossing rigidity must be several times less than the longitudinal rigidity. In terms of bumpers and elastic connections there are used parts which are submitted to shearing, compression and distortion and metallic and rubber blocks. The rubber blocks are characterized by high elasticity and endurance. When metallic and rubber parts are manufactured, it is necessary to ensure the rubber free deformation, which, it is well known, is incompressible within close spaces. The helical and plane springs can also be considerate as elastic elements. The spring sheets thickness is $\delta = 2\ldots6$ mm.
The above vibrating system comprises a simple drive electromagnetic vibrator (Fig.3) including a fixed coil of electromagnet 1 with coils 2, connected to network by a rectifier, the indus 5, rigidly connected to auger 3 of conveyor through the crossing bar 6, endowed with elastic connections 7 and a set of adjusting weights 4, closed alongside with springs in housing 8.

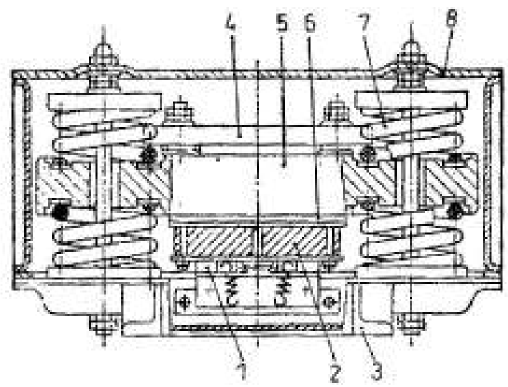

Figure 3. Construction of electromagnetic vibrator

In compliance with the equivalent scheme calculating the vibration of one mass vibrating system shown in figure 4, the system movement equation is:

$$m\ddot{S} + \mu c\dot{S} + c_1 S = F(t) \qquad (1)$$

where: m is the auger reduced mass (equivalent) which includes the indus mass and the adjusting mass; $c_1 = (c + c_0)$ – reduced rigidity (equivalent) of the system, formed of principal elastic elements' rigidity c and rigidity c_0 of vibrator elastic connections.

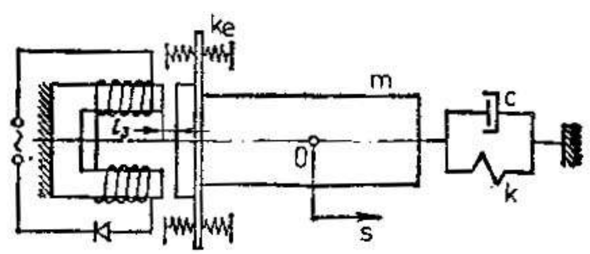

Figure 4. Equivalent scheme for calculating the one mass system vibration system

By replacing the expression of traction force $F(t)$ in equation (1) and taking into account the sinusoidal low of feeding current variation and influence of air variability (interstice) l_j we obtain a differential movement equation which is very difficult to solve. In order to simplify this problem it is supposed a certain constant size of air. Then, the relation (1) acquires the form of a non-homogenous linear equation:

$$m\ddot{S} + \mu c\dot{S} + c_1 S = F(1 + \cos 2 p_T t) \qquad (2)$$

where: $F = k_F B_m^2 S_c / 2$ is steady component of vibrator excitation force, in N;

$k_F = 3{,}98 \cdot 10^5$ – proportionality coefficient;
B_m – amplitude of magnetic induction, in T;
S_c- the area of crossing section of coil core, in m^2;
p_T – feeding alternating current frequency.
The equation particular solution (2) is:

$$S = A_0 + A_1 \cos(2 p_T t - \varphi) \quad (3)$$

Where: $A = F/c_1$ is the steady displacement of mass center O of mass m; A_1 – amplitude of oscillations of center O; φ – angle of phase difference between variable parts of mass center O and disturbing force, given by the relations:

$$A_1 = (F/m)\sqrt{(c_1/m - 4p_1^2)^2 + 4\mu^2 c^2 p_T^2 / m^2} \quad 4)$$

$$\varphi = arctg \frac{2\mu c p_T}{c_1 - 4mp_r^2} \quad (5)$$

The resonance of oscillating system appears for pulsation: $p_T = \sqrt{c_1/m}$.
For current of industrial frequency, f = 50 Hz, the part supporting the load (the auger) oscillates by a double frequency, namely f = 6000 1/min, which is generally inadmissible. In order to diminish the auger frequency up to 3000 1/min, a semiperiod rectifier (Fig.3) is introduced, and the rectified voltage modifies the magnetic flow action and shock movement type. This movement is described as another equation of a disturbing force represented by a circular function Fourier.

Vibration feeders can be considered as two masses spring systems (Fig, 5) [3;4]: the mass of the part feeding the bulk solids m_a consists of the vibrating flute or pipe 1, the bulk solid mass 10 and the magnetic vibrator (positions 2, 3 and 4). The second mass m_f involved is that of the free side consisting of the mobile body 5 and 6. Both masses are connected by springs 7. The electronic control unit 8 is connected with the normal AC voltage supply. Normal 50 Hz excitation will yield a 100 Hz vibration frequency, with thyristor control 25 and 50: efficient control of the vibration amplitude is possible

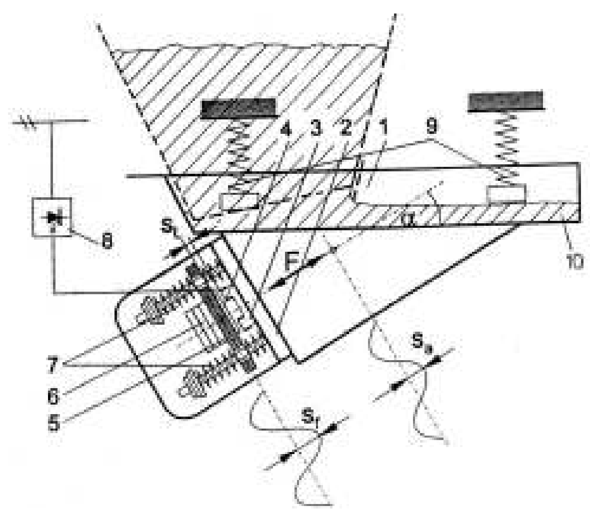

Figure 5. Magnetic vibration feeder: 1-vibration flute; 2 - vibrator housing and other components; 3-electromagnet; 4- mobile magnetic coreplate; 5, 6- additional masses; 7 - spring;8 - control device; 9- suspension springs;10-transported materialS_f- vibration amplitude (free side); S_a -vibration amplitude (working side).

Amplitude s_0 of the chute depends on the constructive and functional parameters, as well as on the type of the transporting material.

The displacement of the vibrating mass can be considered with sufficient approximation, as a harmonic sinusoidal oscillating movement, with elongation s given by relation:

$$s = s_0 . \sin(\omega t) \quad (6)$$

where: s_0 is the oscillating amplitude; ω – oscillating pulsation: $\omega = 2\pi f$ (were f is the oscillation frequency, in Hz.)

The velocity of the oscillating movement is given by relation:

$$v = s_0.\omega \cos(\omega t) \tag{7}$$

where $v_0 = s_0.\omega$ is the amplitude of the velocity, and the acceleration of the oscillating movement is expressed by relation:

$$a = -s_0.\omega^2 \sin(\omega t), \tag{8}$$

where $a_0 = -s_0.\omega^2$ is the amplitude of acceleration.

The oscillating frequency f of the vibrator is given by the well known relation: $f=\omega/2\pi$.

If the value of the amplitude's acceleration a_0 is in ratio with the gravitation acceleration g, is obtained the characteristic of the machine K_M, given by relation:

$$K_M = \frac{a_0}{g} = -\left|\frac{s_0.\omega^2}{g}\right|. \tag{9}$$

The natural frequency f_e of the spring-mass system without attenuation can be determined with the masses m_a and m_t and the spring constant c.

$$f_c = \frac{1}{2\pi} \cdot \sqrt{\frac{c}{m_r}} \quad ; \tag{10}$$

$$m_r = \frac{m_a \cdot m_f}{m_a + m_f}. \tag{11}$$

The vibrating system responds to an excitation frequency f_a with adequate amplitudes which grow the nearer f_a approaches the natural frequency f_e (f_a/f_e=1, resonance). The natural frequency decreases with growing attenuation (Fig. 6), the amplification factor V is very strongly influenced by attenuation. In order to operate vibration feeders close to resonance the system has to be adequately tuned (masses 6, Fig. 5). The effective attenuation is induced by friction, e.g. internally in the bulk solids, and externally at surfaces.

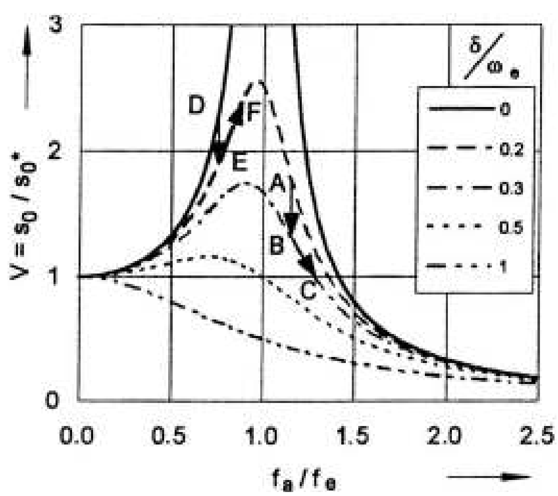

Figure 6. Amplification factor of vibration amplitudes V depending on the excitation frequency ratio (f_a/f_e).

For metering purposes close to resonance conditions the automatic control of the vibration amplitude is strongly recommended to keep the disturbance potential within narrow limits [4; 5]. The measurement of the set vibration amplitude can be achieved by acceleration sensors (Fig.7) or by stationary vibration displacement transducers (Fig.8). With the combined control system for the vibration amplitude which automatically keeps the operational conditions close to the resonance frequency it is possible to obtain good linear and reproducible vibration feeder characteristics.

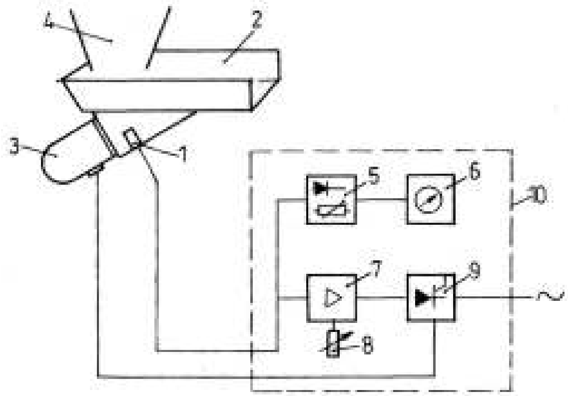

Figure 7. Automatic control of the vibration amplitude: 1 - acceleration sensor; 2, 4 - vibration feeder; 3- vibrator; 5,…, 10- controller.

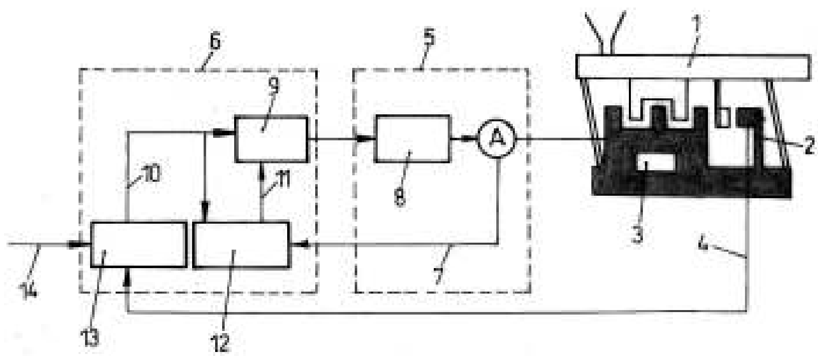

Figure 7. Combined control of vibration amplitude and resonance frequency: 1 - vibration feeder; 2 - vibration transducer; 3 - magnetic drive; 4 - feedback signal;5 - power unit; 6 - control system; 7 - current signal; 8 - amplifier; 9 - AD converter;10 - set-point amplitude; 11 - set-point frequency; 12 - frequency control (slow); 13 - amplitude control (fast); 14 - input signal

CONCLUSIONS

1. Taking into account the technical and economic advantages of the vibrating systems with electromagnets (electrovibrators) they are used not only at vibrating conveyors for grain matters (including seeds) but also at driving the vibrating sieves of technological primary processing installations for cereals;
2.The theoretical study of system of sieves mounted on vibrating frame operated by electrovibrators can be achieved by replacing the real systems with equivalent vibrating dynamic systems with external excitation, comprising one or two vibrating masses;
3. By solving the differential movement equations of vibrating systems with one or two vibrating masses and their applicability for concrete situations, we can study by computer simulation the kinematic and dynamic behaviour of systems endowed with existing or in course of designing vibrating sieves;
4. The results of theoretical researches performed on basis of those presented in the paper allow the optimization of constructive and functional parameters of systems of vibrating sieves, in view of manufacturing state-of the art installation.

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