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Adjustable Driver of Grain Cleaning Vibro-Machine with Vertical Axis of Eccentric Masses Rotation

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Abstract: Vibration machines due to their advantages have found wide application in all fields of technology and are now regarded as the basis of the technology of the future. The researchers give preference to the superresonance vibro-machines. However, it is inherent in them to increase the resonant amplitudes during start-up and stoppage. The greatest application was found for vibro-machines with inertial vibrators driven by asynchronous motors. The main kinematic parameters of vibration are the frequency and amplitude of the vibrations of the working element. The application of vibration in the agricultural sector has been most studied in the processes of releasing seeds of cultivated plants from difficult-to-separate impurities and similar weed seeds. In this case, it is required to smoothly adjust the amplitude from the oscillation frequency according to, the hyperbolic dependence. If the regulation of the oscillation frequency of the operating element can be effectively carried out by a frequency-controlled asynchronous electric drive, the amplitude of the oscillations is established basically by changing the weight of the eccentric mass manually and only during the stoppage of the machine. Therefore, automation of amplitude reduction with increasing frequency during the operation of the machine according to the hyperbolic law is actual. A vibrator with a vertical axis of rotation of the eccentric mass has been developed which limits the resonance amplitudes but it only partially ensures the required control law. Simultaneous regulation of the oscillation amplitude of the working element with frequency change is proposed to be carried out by an automatic inertial vibrator with spring-loaded counter eccentric masses which extend in accordance with the speed. A mathematical model is obtained, a calculation technique is developed and the design parameters of the automatic vibrator are calculated. The mechanical characteristic of a vibrator with an automatic vibrator is specified and the mechanical characteristics of a vibrator with both a base and an automatic vibrator are calculated.

Key words: Vibro-machine, resonance, vibrator, separation of seed mixtures, regulation of amplitude and frequency of oscillations, eccentric mass, radius, weight, spring, moment

INTRODUCTION

In modern machines of the vibration principle, vibration (Zaika, 1977; Horgos *et al.*, 2011; Aipov and Iarullin, 2017; Frolov, 1985) is used, starting with low frequency-infrasonic (from 10 MHz) to high-frequency-ultrasonic (up to 10 kHz) frequencies. It was established that low frequencies correspond to large amplitudes (from several to tens of centimeters), high ones to small amplitudes of oscillations (millimeters and

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fractions of a millimeter) and they vary depending on the type of vibration processes and the state of the materials being processed (Iarullin, 2007; Horgos *et al*., 2011; Iarullin, 2013; Iarullin and Safin, 2014; Nikolov, 2015). Vibrating Machines (VM) have a number of advantages, the main of which are (Frolov, 1985), high productivity with minimal energy consumption, intensification in tens and hundreds of times of many technological processes; The combination of several functions in the same technological processes, the implementation of certain technological processes in which vibration is crucial or without it, processes are generally not feasible (Zhang *et al*., 2013) or unprofitable. Therefore, VMs have found wide application in all fields of technology and continue to thrive, mastering all new areas (microbiology in surgery for cutting organs with focused ultrasound (Kuang *et al*., 2014), space technology, creating new materials) including technological processes in the agricultural sector and are now being evaluated as the basis of the technology of the future (Frolov, 1985).

According to the operating mode of the VM, there are (Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011) pre-resonance, near-resonance, resonance and super-resonance. From the energy point of view, the resonance mode of operation of the VM is most effective. However, the weak point of such VMs is the unstable amplitude of vibration in connection with which complex control systems are developed (Kuang *et al*., 2014; Jiang *et al*., 2011). At the same time, for the operation of a VM with a different resonant frequency or other spectrum of oscillations, it is necessary to change the rigidity of elastic suspensions (Sharapov and Vasiliev, 2017). Therefore, researchers give preference to super-resonance VM in which the oscillation amplitude of the Working Element (WE) does not depend on the technological load of the processed material and remains constant. However, the electric drive of these machines has to overcome resonance during start-up and stoppage, accompanied by a multiple (up to 10 times) increase in the amplitude of the oscillations of the WE. This is especially, inherent in the spatial VM where the mutual influence of the main resonances takes place (Zaika, 1977). Resonance amplitudes on the one hand can complicate the acceleration of the VM, at the same time, it is the cause of the destruction of the machine. In this connection, various methods for analyzing the transient resonance exist and are being developed (Michalczyk and Cieplok, 2016; Michalczyk and Czubak, 2010; Cieplok, 2009). However, this is only a struggle with the consequence and not with the cause of the resonance itself. The main but not sufficient measure of limiting the resonance of the VM is the choice of the lowest possible frequency of the elastic suspension itself which is however, limited by the strength of the elastic elements.

To further limit the resonance during start-up in existing VMs, it is often necessary to fall back (Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011) on the forced start, at the expense of engines with an increased starting torque or uprated power and at stoppage to produce braking by opposing, resulting in increased energy costs. In our opinion, resonance can be most effectively reduced to a minimum only with a balanced state of eccentric mass (Iarullin, 2007; Horgos *et al*., 2011) at the time of passage of the resonance zone.

The motivators of vibration of the VM are inertial Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011; Gonzalez-Carbajal and Dominguez, 2017; Moiseev *et al*., 2015; Zorawski and Dzikowska, 2015), electromagnetic (Iarullin, 2007; Horgos *et al*., 2011; Olaru *et al*., 2017), eccentric (Iarullin, 2007), hydraulic (Li *et al*., 2014), pneumatic (Iarullin, 2007; Horgos *et al*., 2011), piezoelectric (Kuang *et al*., 2014; Sun *et al*., 2015) and other vibrators. Of the known types of vibrators, electromagnetic vibrators are of the greatest interest because of the simplicity of obtaining vibrations. However, the main drawback of them is the difficulty in controlling the amplitude of the oscillations. Hydraulic and pneumatic vibrators are complex in design and require pumps and compressors. The most universal for the VMs with frequency and the amplitude of the WE oscillations regulation are considered (Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011; Frolov, 1985) inertial vibrators. Therefore, from more than one hundred studies of vibro processes in the agricultural sector (Horgos *et al*., 2011) it has been established that mainly (more than 63%) vibrational sources are inertial vibrators, 12.1% are eccentric and 9.73% are electromagnetic.

It was also found (Horgos *et al*., 2011) that Asynchronous Motors (AM) with a mechanical method for regulating the speed of intermediate transmission are used primarily among the VM drive engines (more than 71%). Adjustment of the amplitude of the oscillations in this case is carried out mainly manually by changing the eccentric mass and only during the stoppage of the unit.

The analysis shows that 57% of the studies (Iarullin, 2007; Horgos *et al*., 2011) known to us are devoted to post-harvest processing of grain mixtures in which I class seeds can be obtained in 1-2 passes, effectively separating difficult-to-separate impurities and weed seeds from the seeds of the main crop. It was established that for effective processing of various grain mixtures on one VM it is necessary to smoothly regulate both the frequency and the amplitude of the oscillations of the WE, maintaining their nonlinear interrelation according to the hyperbolic law (Iarullin, 2007; Horgos *et al*., 2011). In our view, it is advisable to smoothly control the oscillation frequency by changing the current frequency of the drive AM by semiconductor converters taking into account the static mechanical characteristics of the motor. Whereas during the start-stop transient processes of the resonance VM, the AM should be investigated taking into account its dynamic mechanical characteristics (Horgos *et al*., 2011; Iarullin and Linenko, 2013). On the basis of the foregoing in our further investigations, a superresonance VM withan inertial vibrator with a drive from a frequency-controlled AM is considered.

The amplitude of the oscillations of the WE of the superresonance VMs (Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011) is independent of the frequency of the oscillations and is controlled by a change in both the weight m_0 and radius of the center of weight r of the eccentric mass and can vary with the oscillating weight m_*. The experiments carried out (Zaika, 1977; Iarullin, 2007; Horgos *et al*., 2011) have shown that the oscillating mass m_* does not have a noticeable influence on the amplitude of the oscillations of the WE of the VM and can be neglected. It was established (Iarullin, 2007; Horgos *et al*., 2011) that the most effective amplitude control is achieved by changing the radius of the center of weight ρ of the mobile eccentric mass. This change is achieved both by the angle α of the mutual location of the two eccentric masses and the radius ρ of the eccentric mass itself (Iarullin, 2007).

According to, the technological requirements for superresonance VMs, vibrators are suitable that simultaneously limit resonant amplitudes and automatically change the amplitude of the oscillations according to hyperbolic dependence (Iarullin, 2007; Horgos *et al*., 2011; Iarullin, 2013; Iarullin and Safin, 2014; Iarullin, 2015; Iarullin and Safin, 2017).

To solve these problems, we developed a VM vibrator which can be adjusted manually (both at rest and on the move) with a vertical axis of rotation of two fixed eccentric masses $m_{б1}$ and m_{H1} (Fig. 1) (Iarullin, 2007; Iarullin, 2015; Iarullin and Safin, 2017). The vibrator eccentric masses are located with a displacement $Z_б$ and Z_H vertically from the origin and a certain angle α of mutual position on the horizontal plane. Changing during operation, the distance Z_H and angle α between eccentric masses is limited, balancing the eccentric masses at $Z_H = 0$ and $\alpha = 180°C$, resonance within the limits of operational requirements and also, adjusting vertical $\alpha_б$ aux and horizontal $\alpha_ς$ amplitudes, respectively. However, it is established (Iarullin, 2015) that due to the change in

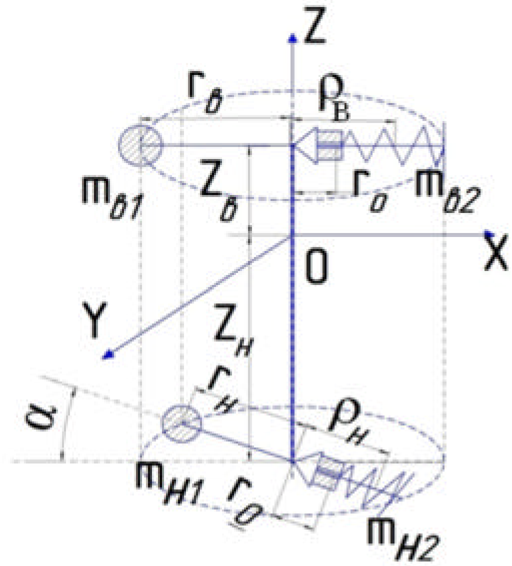

Fig. 1: Vibrator VM with a vertical axis of rotation of fixed and movable counter eccentric masses

the eccentric mass angle α, it is not possible to provide the required amplitude control law. By reducing the distance between the eccentric masses Z_H, the required law is provided only for 82% of the operating speed range, reducing the angle of direction of the oscillations and thereby worsening the separation of the seeds.

MATERIALS AND METHODS

Analytical dependencies in the work are obtained on the basis of the equations of mechanics. During solving we applied the package of programs MATLAB, the program Excel included in the package of Microsoft Office.

RESULTS AND DISCUSSION

In connection with this, it is proposed to adjust the amplitude of the oscillations according to the hyperbolic law by the change in the radius of the center of weight by setting two movable counter eccentric masses $m_{б2}$ and m_{H2} (Iarullin, 2007; Horgos *et al*., 2011; Iarullin, 2013; Iarullin and Safin, 2014; Iarullin, 2015; Iarullin and Safin, 2017) pinned to the rotation axis by springs with a nonlinear stiffness characteristic (Fig. 1) (Iarullin and Safin, 2017).

To determine the design parameters of the automatic vibrator and the mechanical characteristics of the VM with the vertical axis of rotation of the eccentric masses, we obtain analytical dependencies.

Expressions of the oscillation A amplitude for a vibrator with fixed and movable eccentric masses through vertical $\alpha_в$ az and horizontal $\alpha_ς$ terms have the form (Iarullin and Safin, 2017):

$$A^2 = a_в^2 + a_ς^2 = \{0,25 J_y^{-2} R^2 \left[(m_{в1} r - m_{в2}\rho)^2 z_в^2 + (m_{H1} r - m_{H2}\rho)^2 z_H^2 + \right.$$
$$\left. +2(m_{ш1} r - m_{в2}\rho)(m_{H1} r - m_{H2}\rho) z_в z_H \cos\alpha \right\} + \{ m_*^{-2} \left[(m_{в1} r - m_{в2}\rho)^2 + \right.$$
$$\left. +(m_{H1} r - m_{H2}\rho)^2 + 2(m_{в1} r - m_{в2}\rho)(m_{H1} r - m_{H2}\rho)\cos\alpha \right]\} = V^2/\omega^2 \quad (1)$$

Where:

- J_y = The moment of inertia of the vibrating part relative to the central horizontal axis (Y)
- R = The radius of the working element (m)
- $r_в$ and r_H = Radiuses of the center of weight of the upper $m_{в1}$ and lower m_{H1} fixed eccentric masses, respectively (m)
- $\rho_в$ and ρ_H = The are radiuses of the center of weight of the upper and lower mobile eccentric masses, respectively (m)
- $\varepsilon = \text{arctg}\, a_в/a_ς$ = The directional angle of oscillation, rad
- V = The velocity factor depending on the type of VM (Larullin, 2007) equal to (V = A.ω)
- ω = The angular velocity of eccentric masses (rad/sec)

Taking into account that A. sin ε = $a_в$ и A. Cos ε = $a_ς$ (Zaika, 1977) and for border zones ($\omega_{min} \le \omega \le \omega_{max}$) $A^{max} = V/\omega_{min}$ and $A^{max} = V/\omega_{max}$ (Horgos *et al.*, 2011; Iarullin and Safin, 2017), then with $\omega = \omega_{min}$ where $\rho = r_0$ from component $a_в$ and $a_ς$ Eq. 1 we have:

$$a_{в.max}^2 = (m_{в1} r - m_{в2} r_0)^2 z_в^2 + (m_{H1} r - m_{H2} r_0)^2 z_H^2 +$$
$$2(m_{в1} r - m_{в2} r_0)(m_{H1} r - m_{H2} r_0) z_в z_H \cos\alpha =$$
$$4V^2 J_y^2 \omega_{min}^{-2} R^{-2} \sin^2\varepsilon \quad (2)$$

$$a_{в.max}^2 = (m_{в1} r - m_{в2} r_0)^2 + (m_{H1} r - m_{H2} r_0)^2 +$$
$$2(m_{в1} r - m_{в2} r_0)(m_{H1} r - m_{H2} r_0)\cos\alpha$$
$$= V^2 m_*^2 \omega_{min}^{-2} \cos^2\varepsilon \quad (3)$$

where r_0 is the radius of the center of weight of mobile eccentric masses in the state of rest, m. Analogically with $\omega = \omega_{max}$ (for compact vibrator) (Iarullin and Safin, 2017) from Eq. 2 and 3 it follows:

$$a_{в.min}^2 = (m_{в1} - m_{в2})^2 r^2 z_в^2 + (m_{H1} - m_{H2})^2 r^2 z_H^2 +$$
$$2r^2 (m_{в1} - m_{в2})(m_{H1} - m_{H2}) z_в z_H \cos\alpha =$$
$$4V^2 J_y^2 \omega_{max}^{-2} R^{-2} \sin^2\varepsilon \quad (4)$$

$$a_{ς.min}^2 = (m_{в1} - m_{в2})^2 r^2 + (m_{H1} - m_{H2})^2 r^2 +$$
$$2r^2 (m_{в1} - m_{в2})(m_{H1} - m_{H2})\cos\alpha =$$
$$V^2 m_*^2 \omega_{max}^{-2} \cos^2\varepsilon \quad (5)$$

Based on the equations obtained, a calculation of the weights of fixed and mobile eccentric masses for the grain cleaning VM of A.I. Petrusov ВВМ-П with the following constructive-kinematic parameters of the machine (Zaika, 1977) was carried out:

$$m_* = 144_{КГ} \quad J_y = 13,05_{КГ} \cdot M^2$$
$$\alpha = 60° \quad m_{?1} = 1,15_{КГ}$$
$$m_{H1} = 0,726_{КГ} \quad r_в = r_H = r = 0,11_{М}$$
$$Z_в = -0,013_{М} \quad Z_H = 0,375_{М} \quad (6)$$

$R = 0,45_{М}$; $V = A\omega = 0,231_{М/с}$ is the speed factor for frictional seed purifying VM; ω_{min}-ω_{max} = 135-314рад/с is the range of working angular speeds; the vibration parameters for which the VM is tuned $\omega_{раб}$ rad/sec; $A_{раб}$ = 1.35 mm. The system of Eq. 2-5 solved by the Gauss method has the following solution (Iarullin and Safin, 2017):

$$m_{в1} = 1,64372_{КГ}, \; m_{H1} = 1,03767_{КГ}$$
$$m_{в2} = 1,01648_{КГ}, \; m_{H2} = 0,64169_{КГ} \quad (7)$$

To find the characteristics of the rigidity of nonlinear springs and the mechanical characteristics of a VM, an equation for the determination of ρ which is obtained by transforming Eq. 1 (Iarullin and Safin, 2017):

$$\rho^2 \begin{bmatrix} (m_{в2}^2 z_в^2 + m_{H2}^2 z_H^2 + 2m_{в2} m_{H2} z_в z_H \cos\alpha) \cdot \\ 0.25R^2 J_y^{-2} m_*^2 + (m_{в2}^2 + m_{H2}^2 + 2m_{в2} m_{H2} \cos\alpha) \end{bmatrix}$$
$$-\rho \begin{bmatrix} (m_{в1} m_{в2} z_в^2 + m_{H1} m_{H2} z_H^2 + m_{в1} m_{H2} z_в z_H \cos\alpha + \\ m_{в2} m_{H1} z_в z_H \cos\alpha) \cdot 0.5rR^2 J_y^{-2} m_*^2 + 2r(m_{B1} m_{B2} + \\ m_{H1} m_{H2} + m_{в1} m_{в2} \cos\alpha + m_{в2} m_{в1} \cos\alpha) \end{bmatrix}$$
$$+ \begin{bmatrix} 0.25r^2 R^2 J_y^{-2} m_*^2 \begin{pmatrix} m_{в1}^2 z_в^2 + m_{H1}^2 z_H^2 + \\ 2m_{в1} m_{H1} z_в z_H \cos\alpha \end{pmatrix} \\ + r^2 (m_{в1}^2 + m_{H1}^2 + 2m_{в1} m_{H1} \cos\alpha) \end{bmatrix} - V^2 \omega^{-2} m_*^2 = 0 \quad (8)$$

Having calculated for different values ω in the range from 135-314 rad/sec we obtain two values ρ but one of them is greater r = 0.11 m and the other is less r = 0.11 m.

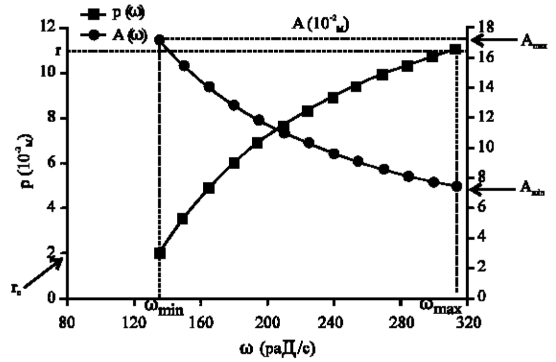

Fig. 2: Frequency response of BBM-II with an automatic vibrator in the range of operating frequencies

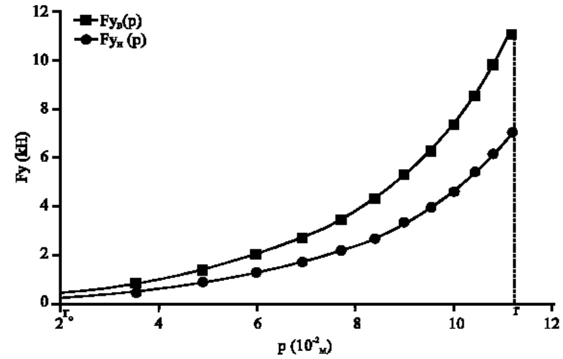

Fig. 3: Characteristics of the elasticity of nonlinear springs from deformation

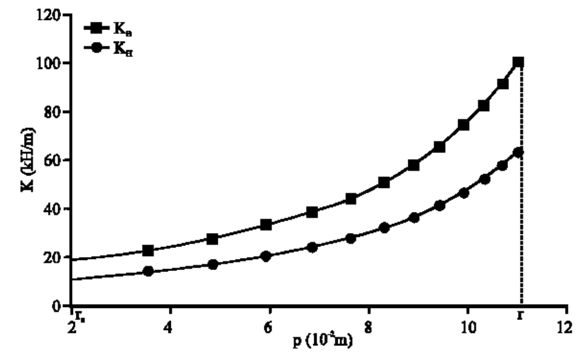

Fig. 4: Dependence of the stiffness coefficient of nonlinear springs on deformation

For a compact vibrator ($\rho = r$) the value ρ can not exceed 0.11 m, so, we choose the value $\rho<0.11$ m (Fig. 2).

It can be seen from Fig. 2 that before reaching the frequency ω_{min}, the mobile eccentric masses are at rest, pressed in pre-compressed springs, at a radius of rest r_0. As the frequency increases from ω_{min} to ω_{max}, the counter eccentric masses gradually extend to the extreme position equal to the radius of the center of weight of the fixed eccentric mass $\rho = r$, ensuring a decrease in the oscillation amplitude of the working element from the maximum $A_{mix} = 1.71$ mm to the minimum value $A_{min} = 0.74$ mm.

To design an automatic vibrator, it is also necessary to know the nature of the change in the elastic force and the stiffness coefficient of nonlinear springs of mobile eccentric masses. The elastic force F_y and the stiffness coefficient of the counteracting nonlinear springs of mobile eccentric masses in the static modes of operation of an SPM are determined by the centrifugal forces of these eccentric masses $F_{yв} = m_{в2}\,\rho\omega^2$; $F_{yH} = m_{H2}\,\rho\omega^2$ and $K_B(\rho) = 1/\rho\, F_{yв}$.

The obtained values of the elastic force and the stiffness coefficient of nonlinear springs depending on the radius of the center of weight of the mobile eccentric masses are shown in Fig. 3 and 4.

As can be seen from Fig. 3 and 4, the elasticity and stiffness characteristics for the non-linear spring of the upper eccentric mass are more important. This is due to the greater weight of the upper mobile eccentric force relative to the lower one. The elastic and stiffness characteristics obtained in numerical expressions allow one to calculate the parameters of nonlinear springs. It is also seen from the graphs that with $\rho = r_0$ the springs are to be mounted pre-compressed with a forceo f 370.5 H at the upper and of 233.9 H at the lower eccentric masses.

When calculating the power and selecting the type of motor for the resonant VM, it is necessary to take into account the reliable resonance passage, stable operation in the working speed zone, sufficient overload capacity and permissible thermal conditions. This is determined by the mechanical characteristics of the VM. The mechanical characteristic of a VM with a vertical axis of rotation with two fixed eccentric masses has the form (Zaika, 1977):

$$M_C = M_K + M_{TP} = \omega^2 \left\{ \frac{S^2 \sin^2\gamma_s}{2m_*\left[\left(p_z^2-\omega^2\right)^2+4h_z^2\omega^2\right]} + \frac{L^2\sin^2\gamma_L}{2J_L\left[\left(p_L^2-\omega^2\right)^2+4h_L^2\omega^2\right]} \right\} + \mu_0 \frac{Д}{2} \left\{ S\sqrt{1+\frac{\sigma_1\left[\sigma_1\omega^4+2\omega^2\left(P_z^2-\omega^2\right)\right]}{\left(p_z^2-\omega^2\right)^2+4h_z^2\omega^2}} + \frac{2L}{z_L}\sqrt{1+\frac{\sigma_2\left[\sigma_2\omega^4+2\omega^2\left(P_L^2-\omega^2\right)\right]}{\left(p_L^2-\omega^2\right)^2+4h_L^2\omega^2}} \right\} \quad (9)$$

where, $M_в$ and $M_{тр}$ are the components of the moment of resistance, respectively for the vibrational and rotational

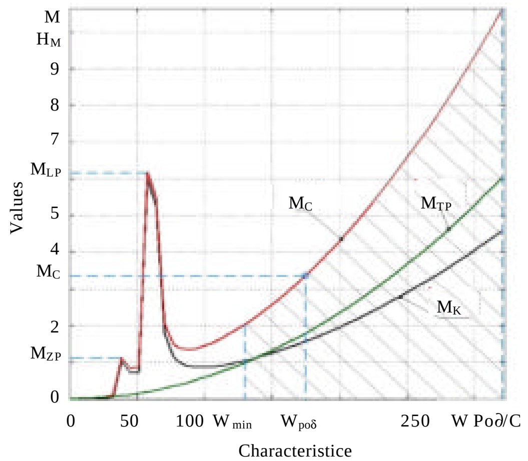

Fig. 5: Mechanical Characteristics MC of the MC and the terms for the vibrational M_K and rotational M_{TP} (friction) motion of the VM with the base vibrator for unbalanced eccentric masses (PZ = 40 rad/sec and PL = 60 rad/sec)

motions, H_m, S is the resultant perturbing force of the eccentric masses, equal to $s=\sqrt{L_B^2+S_H^2+2S_bS_H\cos\alpha}$, H; $S_б$ and S_H is the perturbing force of the upper and lower eccentric masses, respectively, equal to $S_B = m_{B}r_{B}\omega^2$ and $S_H = m_{H1}r_H\omega^2$, H; γ_S and γ_L are phase angles equal to γ_S = arctg $S_h \sin\alpha/S_B+S_H \cos\alpha$ and γ_L = arctg $L_H \sin\alpha/L_B+L_H \cos\alpha$, degrees; L is the resultant disturbance moment, equal to $L=\sqrt{L_B^2L_H^2+2L_BL_H\cos\alpha}$, H_m; L_B and L_H are the perturbing moments of the upper and lower eccentric masses, respectively, equal to $L_H = M_{H1}r_HZ_H\omega^2$ H_m; p_zp_L and h_zh_L are the frequencies of the natural oscillations and the suspension in the parallel and perpendicular direction of the spring axis, rad/sec; μ_0 is the reduced coefficient of friction of bearings of the vibrator; Д is the diameter of the vibrator shaft in the bearing seat, m; z_L is the distance between bearings, m; $\sigma_1 = m^{-1}*(m_{B1}+m_{H1})$; $\sigma_2 = J^{-1}_L(m_{B1}z^2_B+m_{H1}z^2_H)$.

According to the above parameters, the mechanical characteristics of BBM-II with unbalanced eccentric masses of the base vibrator were calculated with the following combinations of natural frequencies P_Z/P_L = 20/40; 30/50; 40/60 (Fig. 5); 60/80; 80/100 и 100/120. The remaining parameters in the calculations are assumed to be equal: $J_L = J_y$; r_0 = 0.02 M; Z_L = 0.388M; μ_0 = 0.008; $2h_z = 2h_L = 5$:

From Fig. 5 it can be seen that in there sonance zone M_c is completely determined by the term of the vibrational motion M_K where as in the working zone the term M_{TP} for rotational motion is of the same order as M_K.

Analysis of the resonance moments M_{zp} and M_{Lp} with respect to the resistance M_c at the operating speed

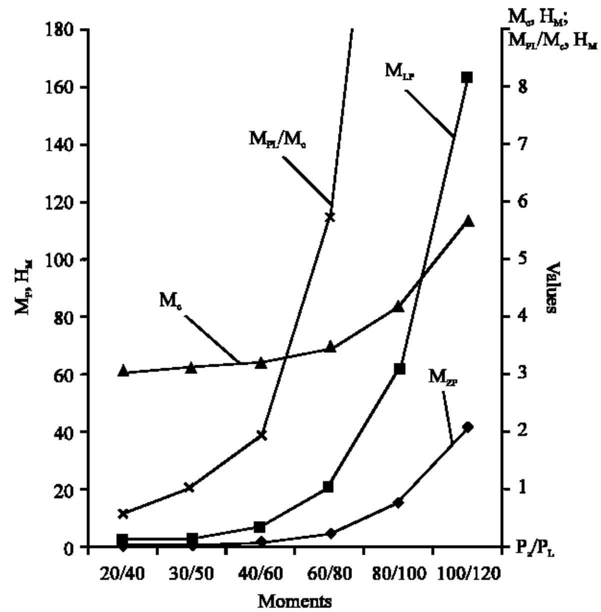

Fig. 6: Dependence of the resonance moments M_{ZP} and M_{LP} and the resistance MC at ωpaδ = 171 rad/sec on the natural frequencies of the VM with the base vibrator for unbalanced eccentric masses

$\omega_{раб}$ = 171.2 rad/sec with the amplitude $A_{раб}$ = 1.35 mm (Iarullin and Safin, 2017), on which the weights of the VM imbalances with the base vibrator are tuned shows (Fig. 6) that firstly $M_{zp} \ll M_{Lp}$, secondly as the natural frequencies P_Z/P_L like M_{zp}/M_c and M_{Lp}/M_c PZ/PL increase sharply from 0.55-43.4 times. Taking into account the multiplicity of the starting moments of AM which are within the limits of 1.7-2.2 of the nominal values, it can be seen that only with P_Z/P_L = 40/60 a VM with a base vibrator is reliably triggered with unbalanced eccentric masses.

For other cases, the resonant amplitudes and thus, the moments can be reduced to the permissible limits by launching a VM with balanced ecentric masses. Thus, for example, at P_Z/P_L = 80/100 (Iarullin, 2015), the exceeding of the WE amplitude in this VM could be reduced relative to the operating one from 42.86-2.3 times, i.e., in 18.6 times. Obviously, this also corresponds to a decrease in the resonance moment M_{LP} to the level M_c which is what we need to find out next. For a VM with an automatic vibrator, the components of the static resistance moment Eq. 7 take the form:

$$S_б = (m_{б1}r_б - m_{б2}\rho)\cdot\omega^2 \quad S_H = (m_{H1}r_H - m_{H2}\rho)\cdot\omega^2 \quad (10)$$

$$L_б = (m_{б1}r_б - m_{б2}\rho)\cdot\omega^2 \quad L_H = (m_{H1}r_H - m_{H2}\rho)\cdot\omega^2 \quad (11)$$

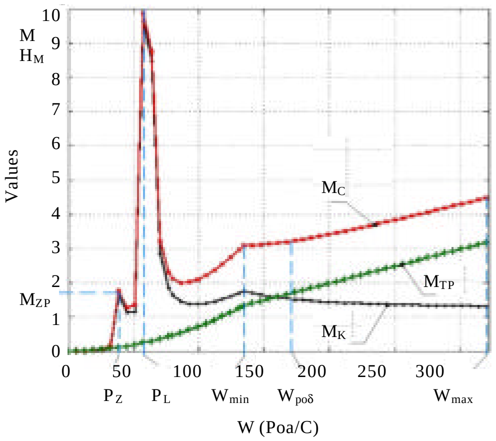

Fig. 7: Mechanical characteristics of the M_C and the components for the vibrational M_K and rotational M_{TP} (friction) motion of a VM with an automatic vibrator with unbalanced eccentric forces

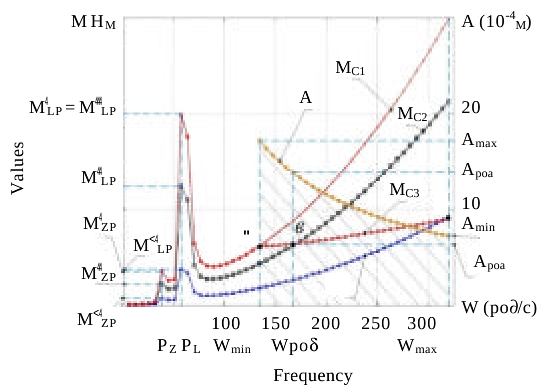

Fig. 8: Amplitude-frequency A (ω) and mechanical characteristics MC (ω) of the VM with unbalanced eccentric masses of the base vibrator at operating speeds $\omega_{Pa\delta}$ = 171 rad/sec-MC_2, boundary velocities ω_{min}-MC_1, ω_{max}-MC_4 and with automatic vibrator-MC_3

$$\sigma_1 = \frac{m_{\theta 1}+m_{\theta 2}+m_{H1}+m_{H2}}{m_*}$$
$$\sigma_2 = \frac{(m_{\theta 1}+m_{\theta 2})\cdot z_{\theta}^2+(m_{H1}+m_{H2})\cdot z_H^2}{J_L} \tag{12}$$

Taking into account that the automatic vibrator works only in the range of operating speeds ω_{min}:ω_{max} rad/sec in Eq. 8 and 9 in the range of speeds $0\leq\omega\leq 135$ rad/sec is accepted $\rho = r_0$ (Fig. 7-9).

And in the working range of speeds the radius of mobile eccentric masses ρ is determined by an equation (Frolov, 1985) for the case of compact vibrator (Iarullin and Safin, 2017) which has $r_0\leq\rho\leq r$. From Fig. 7-9 it can be seen that the mechanical characteristic

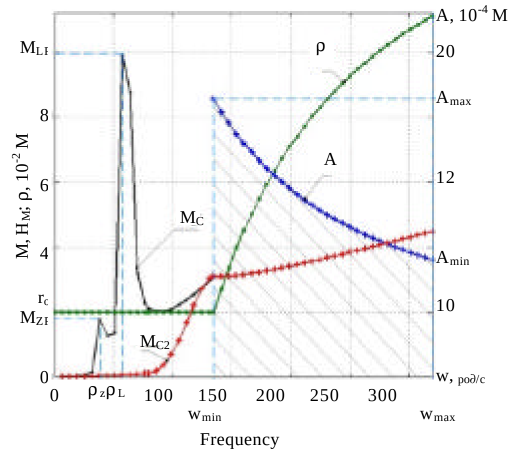

Fig. 9: Amplitude-frequency characteristic A(ω) p (ω) and mechanical characteristics M_C (ω) of the VM with an automatic vibrator for unbalanced eccentric masses

of a VM with an automatic vibrator for unbalanced eccentric masses (α = 60 °C; z_H = 0.375_M) in the resonance zone has the same character of change (Fig. 5). However, the resonance moment M_{lp} increases from 6.3-10 H_m (Fig. 8), i.e., in M_{Lp}/M_c = 3.1 times which in turn leads to the choice of AM of higher power. This is due to the selection of a larger weight of fixed eccentric masses of an automatic vibrator designed for operation over the entire range of operating speeds beginning already with ω_{min} and A_{max} (Fig. 7-9).

At the work section with an increase in the speed from ω_{min} up to ω_{max} mobile eccentric masses m_{B2} and m_{H2} moving out with a radius ρ from r_0 to r, reduce the amplitude A_{max} to A_{min} according to the hyperbolic law (Fig. 9). At the same time, the resistance moment in the automatic vibrator grows less intensively M_{c3} than M_{c1} (Fig. 8) as well as its components M_K and M_{Tp} (Fig. 7). It can be seen from Fig. 8 that the points a, b and c in the working section M_{c3} of the work of the automatic vibrator correspond to the resistance moments of certain technological processes with parameters, respectively ω_{min} and A_{max} and $\omega_{pa\delta}$ and $A_{pa\delta}$; ω_{max} and A_{max}, inter connected according to the hyperbolic law A = V/ω. When the VM operates with basic vibrators for the same modes mechanical characteristics correspond to curves M_{c1}, M_{c2} and M_{c4}. The mechanical characteristics of a VM with an automatic vibrator at start-up with balanced eccentric masses (α = 180°; Z_H = $0{,}05_M$) in the range of speeds $0\leq\omega\leq 80$ rad/sec are shown in Fig. 9.

It can be seen from Fig. 9, that the resonance moment is practically absent. At the same time, the power of the variable-frequency AM is selected only for ω_{max} non-taking into account the moments of the starting branch of the motor characteristics.

As a result of the research, an automatic vibrator for a VM with a vertical axis of rotation of eccentric masses is proposed. The proposed vibrator, due to movable counter eccentric masses pressed against the axis of rotation by springs with a nonlinear stiffness characteristic with a change in the oscillation frequency, automatically provides in the resonant region the rational amplitude of the oscillations of the technological processes according to the hyperbolic law. To do this, the mobile eccentric masses in a state of rest and until the lower limit of the working speed ω_{min} is reached are fixed and pressed to the axis of rotation by pre-compressed springs. Analytical dependencies are obtained for a vibrating grain-cleaning machine BBM-II with a basic version reconstructed on a VM with an automatic vibrator for which the following are calculated weights of fixed and mobile eccentric masses, the nature of the change in the radius of the center of weight of mobile eccentric masses, characteristics of elastic forces and stiffness coefficients of nonlinear springs, the mechanical characteristics of a VM with a base vibrator and an automatic vibrator both with limitation and without limitation of resonance.

The obtained results allow, both for existing and projected VMs with vertical axis of rotation of eccentric masses, to develop an automatic vibrator having preliminarily established the parameters of the hyperbolic law of interrelation of rational vibration parameters of technological processes carried out on the VM with the same law of motion of the WE.

The mechanical characteristics of a VM with an automatic vibrator provide clarity, convenience in analyzing the operation of a VM electric drive both in steady-state conditions, starting and stopping processes and when adjusting the speed within the operating range. They also allow estimating the overload capability of the drive motor and static stability which increases due to the greater rigidity of the mechanical characteristics of the VM. A VM with an automatic vibrator, limiting the resonance in the start-up and stoppage processes, allows to reduce the power of the drive AM which is calculated only for ω_{max}. Thus, for ω_{max} = 314 rad/sec and M_c = 4.5 H_M, the calculated power will be $P_p = M_c, \omega_{max}$ = W. The results obtained are comparable to the current power of the existing BBM-II with a conventional vibrator from fixed ecentric masses without a resonance limitation mechanism (Horgos *et al*., 2011) equal to 1.7 kW which with a V-belt variator provides operation in the speed range within 178.5-252 rad/sec. Then the power at a speed of ω_{max} = 252 rad/sec with resonance limitation would be $P_P = M_c .\omega_{max}$ = 3, 9.252 = 982.8 W. Therefore, the power of 1.7 kW of AM is selected taking into account the passage of the resonance. The foregoing testifies to the reliability of the results obtained.

The results obtained in the course of the study relate to the controlled drive of the working elements of vibrating machines.

CONCLUSION

Vibration machines with their advantages are widely used in all fields of technology and are now regarded as the basis of the technology of the future. Researchers give preference to resonant vibro-machines for which the amplitude is stable. However, they have to constantly overcome the resonance. The limitation of resonance can be most effectively achieved by balancing the eccentric masses of inertial vibrators in the resonance zone.

The drive of known vibrators in the agricultural sector is mainly carried out by inertial vibrators 63% from asynchronous motors 71% with mechanical speed control and manual setting of the amplitude by changing the weight of eccentric masses. In the agricultural sector, 57% of studies are related to vibroseparation of seeds of cultivated plants from difficult-to-separate impurities and similar weed seeds. For the effective operation of vibrators, smooth regulation of the frequency and amplitude of the oscillations according to the hyperbolic law is required.

For a vibrator with a vertical axis of rotation of eccentric masses, a vibrator with a manual restriction of resonances and automatic amplitude control in the working zone of velocities according to hyperbolic law was developed by setting two movable counter eccentric masses pressed to the axis of rotation by nonlinear springs.

Analytical dependencies on which the weight of eccentric masses are calculated, the radius of the center of weight of mobile eccentric masses, the elastic force and the stiffness coefficient of nonlinear springs and the mechanical characteristic with different vibrator variants for the BBM-II vibrator of A.I. Petrusov are obtained.

The obtained results allow to determine the design parameters of the automatic vibrator for both existing and projected vibromachines with the vertical axis of rotation of the eccentric masses. To do this, it is necessary to preset the speed factor of hyperbolic dependence for the current technological processes.

The obtained mechanical characteristic of a vibrator with an automatic vibrator makes it possible to select a rational frequency-controlled asynchronous electric drive with a more precise law for regulating the voltage and frequency of the current of the frequency converter.

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