SOME FACTORS INFLUENCING THE PRECISION OF TECHNOLOGIC PROCESSES IN THE VIBRATORY MACHINES

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Abstract - In the paper, the spacial dynamic model of a vibratory technologic machine with load is presented. On the basis of the systemic approach, the movement equations of the material are acquired, where in the form of the non-linear members, the spacial (non-working) oscillations of the working member caused by possible construction and other deviations of the machine are included; their influence on the movement velocity of the material is studied. The solultion of the equation shows that some spatialoscillations in combination with the working vibration contributes the increse of the material's movement velocity, and based on this, a new construction of a vibratory machine can be designed. The given approach is a novelty and it reveals possibilities of wider research of vibratory technological processes.

Keywords - Vibratory Technologic Machine, Spacial Oscillations, Vibratory Movement Velocity of Material, Mathematical Modeling.

I. INTRODUCTION

Vibratorytechnologicmachinessuchasvibro-conveyers, vibro-feeders, vibro-batchers, vibro-mixersetc., arewidely used in many spheres of industry, where transportation, mixing, separation, metered feeding etc., of homogeneous or heterogeneous friable materials take place [1, 2, 3, 4, 5].
The above-mentioned processes are performed with certain precision whose improvement depends on the various factors of the vibratory system "vibro-drive – elastic system – working member" including physical characteristics, geometric parameters, assemblage precision etc.

The electro-magnetic vibratory machines, where elastic systems are cylindrical springsmeant mainly for transfer of electro-magnetic forces in the direction of the longitudinal axis, are distinguished by greater sensitivity to the mentioned factors.

II. THE POSIBLE DESIGN ERRORS

The elastic units and elements connected to them play an important role in the mechanical systems of vibro-machines. This especially concerns to specificity of elastic elements, for example springs, which vibrate not only in the direction of the exciting force but also in the circular and lateral directions. Nonworking vibrations are usually small, they are neglected at the researches since is thought that they do not have significant influence on the technologic process. But at vibrations with great amplitudes (for example at resonant and the like) such errors can influence significantly the machine working processes.
Theanotherreasonofgenerationofthenonworkingvibrationsisinaccuracyofmanufacturingofthesurfacesofjunctionsoftheelasticelementsandmasses. Even deviations in the limits of tolerances cause nonworking vibrations at vibrations with great amplitudes that affect the quality of the technologic processes [1, 6, 7].

The next factor affecting the working process is inaccuracy of transmission of the exciting force on the working member.
In this case inaccuracy can be caused by the exciting force as well as by non correct disposition of the elastic elements.

The position of the working member obtained because of the above-mentioned deviations is shown schematically in Fig.1: system of coordinates $O_1 x_1 y_1 z_1$ corresponds to the machine ideal position. Axis O_1x_1coincides with the direction of the electromagnetic force transmitted from the vibro-drive to the working member.The real point of application $O_1^{'}$ of the force is displaced relative to O_1 by the eccentricities e_x, e_y,e_z becouse of errors of fabrication and assemblage of the vibro-mqchine. Besides the axes $O_1^{'}x_1^{'}y_1^{'}z_1^{'}$ are deflected from axes $O_1 x_1 y_1 z_1$ by the angles $\theta_0, \psi_0, \varphi_0$ (Fig.1).
The mentioned angular errors cause deviations of the exciting force Q(t) whose projections on the axes of the coordinate system $O_1^{'}x_1^{'}y_1^{'}z_1^{'}$ are determined by the directing cosines:

$$\begin{aligned} Q_{x_1^{'}} &= Q(\psi_0 \sin\alpha_1 + \cos\alpha_1); \\ Q_{y_1^{'}} &= Q(\varphi_0 \cos\alpha_1 + \theta_0 \sin\alpha_1); \\ Q_{z_1^{'}} &= Q(\psi_0 \cos\alpha_1 + \sin\alpha_1); \end{aligned} \qquad (1)$$

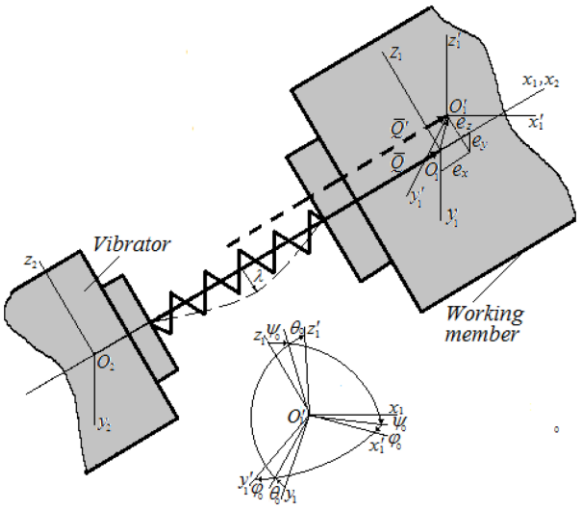


Fig.1. The possible deviations of directions of the force transmission and coordinate axes

where $\alpha_1 = \alpha + \beta$ □α is an angle of inclination of the working member; β – angle of vibrations; the values of angles $\theta_0, \psi_0, \varphi_0$ □ aresmalland for this reason linear terms will only betaken into account at developing of the directing cosinesin (1).

III. A MATHEMATICAL MODEL OF THE MOVEMENT

Equations of the spatial movement of the working member will take the form:

$$m_1 \ddot{x}_1 + c_x \dot{x}_1 + k_x x_1 - k_x' \theta_1 = Q_{x_1};$$

$$m_1 \ddot{y}_1 + c_y \dot{y}_1 + k_y y_1 - k_y' \psi_1 = Q_{y_1}; \quad (2)$$

$$m_1 \ddot{z}_1 + c_z \dot{z}_1 + k_z z_1 - k_z' \varphi_1 = Q_{z_1}.$$

$$J_\theta \ddot{\theta}_1 + c_\theta \dot{\theta}_1 + k_\theta \theta_1 - k_\theta' x_1 = M_{x_1};$$

$$J_\psi \ddot{\psi} + c_\psi \dot{\psi}_1 + k_\psi \psi_1 - k_\psi' y_1 = M_{y_1}; \quad (3)$$

$$J_\varphi \ddot{\varphi}_1 + c_\varphi \dot{\varphi}_1 + k_\varphi \varphi_1 - k_\varphi' z_1 = M_{z_1},$$

where M_{x_1}, M_{y_1}, M_{z_1} - projections of the moments are obtained from the expression $M = Q.e$; $c_x, c_y, c_z, c_\theta, c_\psi, c_\varphi$ -damping coefficients of the elastic system; $k_x, k_y, k_z, k_\theta, k_\psi, k_\varphi$ - coefficients of rigidity; $k_x', k_y', k_z', k_\theta', k_\psi', k_\varphi'$ - coefficients connecting lateral rotational and longitadional torsional vibrations; $Q=Q_o f(t)$, where Q_o is a coefficient of the exciting force; f(t) – law of variation of the exciting force; J_θ, J_ψ, J_φ – moments of inertia of masses with respect to theproper axes. Incontrast to equations (1) and (2) dynamical deviations θ_1, ψ_1, φ_1 of the coordinate axes are considered in projections of the exciting force in the right-hand parts of eq-s (3).

The equations describe spatial movement of the friable material for which a classical method of the relative and translational motions and systemic approach are used [2, 4].The possible errors e_x, e_y,$e_z$and $\theta_0, \psi_0, \varphi_0$ - (1), (3, 4) shownin Fig.1 areconsideredinthepresentedequations (1), (2), (3), (4) incontrasttogeneralizedequations [7, 8, 9].

Second degree in the form of products are retained in the equations of spatial movement of the load[2, 7]:

$$m_3 \ddot{x}_2 + (\ddot{x}_1 - \ddot{z}_1 \psi_1)\cos a_1 - (\ddot{z}_1 + \ddot{x}_1 \psi_1)\sin a_1 +$$
$$+ z_3 \ddot{\psi}_1 + \varphi_1 \ddot{y}_1 + 2\dot{\psi}_1 \dot{z}_3 - 2\dot{y}_3 \dot{\varphi}_1 + \dot{\varphi}_1 y_3 -$$
$$- \ddot{\varphi}_1 - \ddot{x}_1 \psi_0 \sin a_1 - \ddot{z}_1 \psi_0 \cos a_1 + \ddot{y}_1 \varphi_0 + A\dot{x}_1 +$$
$$+ B\dot{y}_3 + B_1 \dot{x}_3 = -f(N_z - N_y) sign(\dot{x}_3) - g \sin a,$$
$$m_3 \ddot{y}_3 + \ddot{y}_1 + (\ddot{z}_1 \theta_1 - \ddot{x}_1 \varphi_1)\cos a_1 + (\ddot{x}_1 \theta_1 +$$
$$+ \ddot{z}_1 \varphi_1)\sin a_1 - \theta_1 z_3 - 2\dot{\theta}_1 \dot{z}_3 + 2\dot{\varphi}_1 \dot{x}_3 -$$
$$- \ddot{x}_1(\varphi \cos a_1 - \theta_0 \sin a_1) + \ddot{z}_1(\theta_0 \cos a_1 +$$
$$+ \varphi_0 \sin a_1) + B\dot{y}_3 - C'y_3 = -fN_z sign(\dot{y}_3);$$
$$m_3 \ddot{z}_3 + (\ddot{z}_1 + \ddot{x}_1 \psi_1)\cos a_1 + (\ddot{x}_1 - \ddot{z}_1 \psi_1)\sin a_1 -$$
$$- \theta_1 \ddot{y}_1 + y_3 \ddot{\theta}_1 + 2\dot{y}_3 \dot{\theta}_1 - 2\dot{x}_3 \dot{\psi}_1 -$$
$$- \ddot{z}_1 \psi_0 \sin a_1 + \ddot{x}_1 \psi_0 \cos a_1 - \ddot{y}_1 \theta_0 + D_1 \dot{Z}_1 +$$
$$+ E\dot{x}_1 + E'\dot{z}_3 + Cz_3 = -fN_y sign(\dot{z}_3) - g \cos a,$$

wherecoefficients A, B, B_1, C', D, E, E_1, C characterize position of the loud relative to the working member surface [2,3] (movement either together with the surface or separately from it in the air); N_y, N_z - normal reactions of the load relative to the working member surfaces; equations (4) differ from general equations of spatial movement of the load [4, 10, 11, 12] with the terms that are generated as a result of errorsθ_0, ψ_0, φ_0.

Equations (3) and (4) togetherwith the expression of the exciting force of the electro-magneticvibro-drive weresolved for the following values of the errors : $\theta_0, \psi_0, \varphi_0 = 0 \div 9^0$;$e_x$, e_y, $e_z = 0 \div 0{,}015$ mm(5)

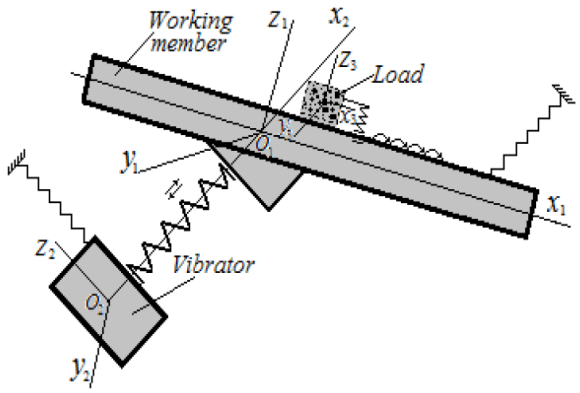


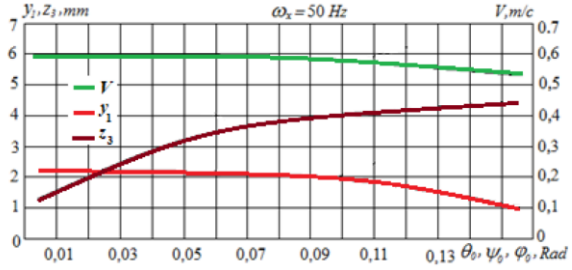
Fig.2.The vibratory feeding

IV. SOME RESULTS OF THE MATHEMATICAL MODELING

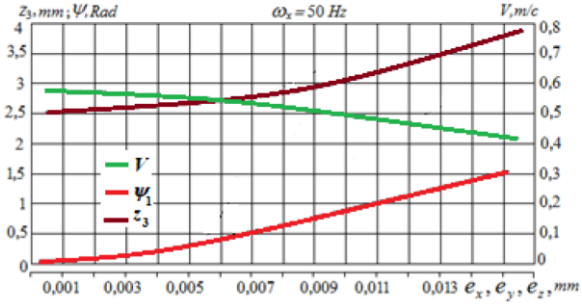
Mathematical modeling was carried out for the following resonant working regimes of the electromagnetic vibro-drive: basic (50 Hz), sub-harmonic (25 Hz) and super-harmonic (100 Hz). The real physical data in the form of coefficients were put in the equations of movement of the basic and sub-harmonic electromagnetic vibro-feeders and the conditinal data – in the equations of movement of the super-harmonic machine.

At normal functioning of a vibro-machine when parasitic frequencies are far from the resonant ones, the errors in the given limits do not have a significant influence on the transportation process. Therefore, the following approach was used: resonant vibrations were generated in the mathematical model of the vibro-machine in all nonworking spatial directions at the numerical experiment and only after that, rise of the errors could have a significant influence on the transportation process.

In the Fig.3 are shown dependences of velosity V, trajectory z_3 and lateral vibrations y_1 N_z upon the errors of direction of the coordinate axes $\theta_0, \psi_0, \varphi_0$ The graphs correspond to bringing the nonworking vibrations θ_1 $y_1$of the working member into resonance at simultaneous variation of the errors. Resonant vibrations in the direction of O_2x_2(Fig.2) ($\omega_{exc}=\omega_x=50$ Hz) represent a working regime of the machine that is constant during the experiment.



Fig,3. Dependences of the Vandz$_3$on the y_1 at variation of the $\theta_0, \psi_0, \varphi_0$



Fig, 4. Dependences of the V and z_3 on the □□ at variation of the e_x, e_y,e_z

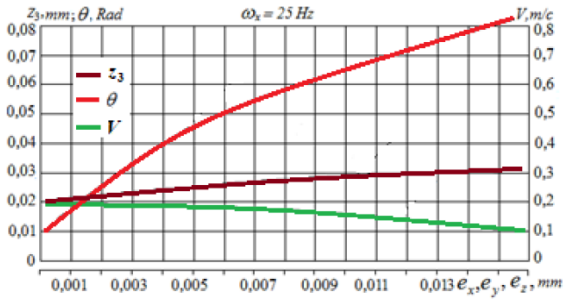


Fig.5. Dependences of the V and $z_3$on the □□ at variation of thee$_x$, e_y,e_z

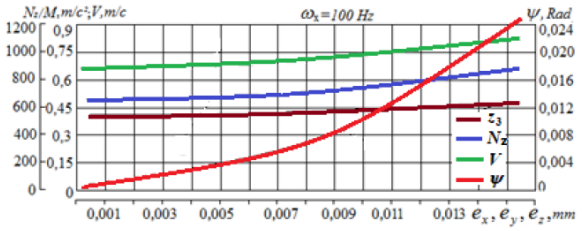


Fig.6. Dependences of the V, $z_3$and N_z on the □ at variation of the e_x, e_y,e_z

The spatial vibrations which are brought into resonance for their enhance and study of their influence on the process, are mainly pre-resonant vibrations; their phases either coincide with-or differ little from the phases of the working resonant vibrations.

It is evident that the same near-resonant vibrations whose proper frequencies are in the after-resonance area, will have different influence on the same working process of the vibro-machine.

In Fig.4 are shown the graphs describing dependence of speed V and trajectories (z_3) of the friable load upon variation of the point of transmission of the exciting force of the vibro-machine on the working member at sub-harmonic working regime. The graphs correspond to bringing of the non-working vibrations □□□ which, in turn, changes according to the deviations□□ of the working member into the resonance at simultaneous variation of the errors. Influence of the errors upon the transportation process in the basic and sub-harmonic resonant regimes has a dissimilar character.

Dependences of speed V and trajectory z_3 upon angular displacement θ of the working member and errors e_x, e_y,$e_z$at operating of the vibrator in the sub-

harmonic (25 Hz) resonant regime are shown in Fig.5. Similar dependences for the super-harmonic resonant regime are shown in Fig.6 where dependences of V, z_3 and normal vertical reaction N_z upon variation of the amplitude ψ of the working member rotary vibrations are given. It should be noted that in contrast to sub-harmonic machines super-harmonic resonant machines could not find practical use. This is explained by their increased dynamical loads, power expenses and metal-consumption. Therefore, research into the vibro-transportation process in the super-harmonic regime for the present has a theoretical character. In spite of the fact that such regimes can practically be realized in the existent electro-magnetic vibratory machines by the small constructional alteration.

The carried out model researches have shown, that the errors of the manufacturing and assemblage and parasitic vibrations generated on their basis disturb the normal course (envisaged by the project) of the technologic processes and in many cases they worsen main dynamical and exploitation parameters. However, it should be also noted that in some cases the transportation speed retains constant value or tends to rise. On this ground, we can consider it expedient that instead of decreasing the errors down to minimum (which is a quite difficult problem) create such construction of the vibro-drive, where controlling of the exciting force and moment will be possible. In this case it will be expedient to enhance those vibrations, whose combination with the basic working vibrations promotes improvement of the technologic process – rise of the speed and intensity of the load movement.

V. CONCLUSIONS

Deviation of directions of the exciting force and coordinate axes from their normal position caused by the errors of manufacturing and assemblage of the vibro-machine are considered.
The mentioned deviations are put in the system of equations of the movement which enables to study influence of the spatial nonworking vibrations upon behavior of the load to be transported.
Graphs of variation of the load speed are obtained with the help of mathematical modeling at variation of the mentioned errors in the certain limits.
It is ascertained that some parasitic vibrations have positive influence upon speed of the load that may become a base for constructional modernization.

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