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USE OF SELF-SYNCHRONIZATION EFFECT IN VIBRATION CARS

Abstract. The problems of self-synchronization of a dynamic system of a resonant type with two degrees of freedom are considered. The possibility of self-synchronization of two suited resonant vibrating machines connected by an elastic connection is considered as dynamic systems. A theory is given about the property of the natural frequencies of sectioned systems. An approximate periodic solution corresponding to the self-synchronization mode is constructed. The parameter ranges leading to a periodic re-gime are investigated. The possibility of block design of resonant vibrating machines for technological purposes has been established. Mathematical modeling of transient and stationary modes of a number of resonant vibrating machines in block design has been carried out. The results of mathematical modeling of the problem using numerical methods (Maple, Delphi) are presented. The use of the results of theoretical studies makes it possible to abandon the rigid kinematic synchronization of the rotors of vibration systems, which leads to a reduction in the energy intensity of the equipment. A methodology has been proposed and devices have been developed for experimental research of the main laws and characteristics of the technological process of vibratory compaction of concrete mixtures. The study shows that resonant single-mass vibrating machines, when combined into a single system, relatively easily enter the self-synchronization mode and this mode is stably maintained when a number of system parameters change within a relatively wide range. This makes it possible to widely use the principle of sectioning when creating resonant vibrating machines and, in some cases, create heavy resonant vibrating machines from standard modules. It has been established that this method makes it possible to obtain concrete products of higher density than with known methods. In this case, water absorption decreases by 15-25%, and strength characteristics increase by 25-31%.

Keywords: synchronization, vibration machines, vibration exciters, vibration platforms, elastic connections, autonomous systems.

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Introduction. Modern mechanical engineering puts forward as a primary task the problem of increasing the productivity of technological machines and equipment while simultaneously reducing their specific metal and energy intensity.

The introduction of vibration technology is impossible without the creation of highly efficient, reliable vibration machines and devices. Expanding the capabilities of a vibrating machine in a number of cases is associated with the use of the phenomenon of self-synchronization, which makes it possible to improve existing ones and create some fundamentally new vibration devices.

The work examines new possibilities for using the self-synchronization effect in vibration machines for technological purposes to increase the efficiency of their operation and transition, in some cases, to their sectional design.

Currently, most reinforced concrete structures are manufactured using the vibration method. At the same time, the most widespread type of equipment for compacting concrete mixtures is unified block vibrating platforms with an inertial drive in a resonant mode and operating at a frequency of 50 Hz. Along with the advantages (ease of changeover, high vibration stability), machines of this type have a number of significant disadvantages

Firstly, due to the resonant setting of the vibrating platform, high-power motors are required to excite vibrations of the working element (energy consumption of the equipment is about 5 kW per 1 ton of lifting capacity).

Secondly, due to the increased load on the bearings of the inertial drive, their durability is sharply reduced (on average, the durability of one rolling bearing is no more than 3500...4000 hours, which is less than the minimum permissible value L = 5000 hours accepted in mechanical engineering).

Thirdly, the specific design of the vibrating platform (linearity of the elastic system, harmonic nature of excitation) makes it possible to implement only symmetrical harmonic laws of vibration of the working body, which cannot ensure high quality compaction of rigid concrete mixtures. This is contrary to modern trends in the development of the precast concrete industry, aimed at the use of rigid concrete mixtures to obtain high-strength concrete and reduce cement consumption.

Fourthly, since the acceleration amplitude ranges from 3 to 6 times the acceleration of gravity, the concrete mixture, when vibrating without a significant inertiafree load, periodically bounces, separating from the bottom of the mold, and air is sucked into the resulting gap; therefore, it is not possible to completely displace air from the concrete mixture, which reduces the quality of the products.

Fifth, due to the presence of a large number of relatively short-lived elements, machine downtime due to breakdowns turns out to be unacceptably long.

Along with this, there are machines that do not have most of these disadvantages. Such vibration machines include a two-mass resonant vibration platform. In addition, such a machine has one more advantage – the ability to almost completely eliminate the transfer of dynamic forces to the base [1]. A solution to this problem can be achieved by using the self-synchronization effect.

Materials and methods. Interest in using the features of nonlinear vibrations in vibrating machines arose in connection with the need to solve three main problems that are extremely important for improving the vibrating machine. We are talking about increasing the stability of resonant modes and implementing polyharmonic oscillations of the working body, with gearless frequency conversion. The idea of creating vibration devices operating in resonant modes and realizing intense vibrations with small forcing influences has long attracted the attention of designers [2]. However, due to the strong sensitivity of the amplitude of resonant oscillations to changes in the mass of the load of the processed medium, linear vibrating machines of the resonant type are not widely used [1]. To increase the stability of resonant vibrations, it is advisable to use elastic links with a nonlinear characteristic in vibrating machines. The presence of elastic nonlinearity contributes to the slope of the frequency response in the main resonance zone and, as a consequence, to increased stability of operating modes.

The simplest option for introducing nonlinearity into the elastic system of a vibrating machine, as mentioned above, is to install elastic travel limiters (buffers). The influence of the rigidity of buffers and the gaps with which they are installed on the stability of resonant modes is considered in [2,3].

Technical problems leading to the problem of synchronization of vibrators. A significant number of vibration machines and installations use not one, but several vibrators installed either on one common supporting body (the vibrating organ of the machine), or on two or more bodies connected in one way or another.

In other cases, the use of several relatively low-power vibrators, instead of one more powerful one, is due to the need to disperse the disturbing force over a vibrating organ of considerable size.

Currently, there are three main methods of synchronizing mechanical vibrators. The simplest way is to introduce kinematic connections between the vibrator rotors, i.e. kinematic synchronization [4]. The second method of synchronizing mechanical vibrators is electrical synchronization, where the desired synchronization is achieved by using an electric shaft system or synchronous motors [5-11]. Both of the methods described above for coordinating the rotation of vibrator rotors – kinematic and electrical – refer to forced synchronization. The third method is self-synchronization. The phenomenon of self-synchronization of mechanical vibration exciters, in the simplest case, represents unbalanced rotors driven into rotation by asynchronous electric motors. The essence of the phenomenon is that several such rotors installed on a common working body of the machine rotate with the same or multiple average angular velocities and certain phase shifts, despite the absence of any kinematic connections between them, for example, gearing, chain transmissions and etc. Synchronicity and a certain phrasing are ensured automatically due to the vibrations of the bodies on which the rotors are installed.

The conditions for the existence and stability of synchronous movements of vibration exciters are usually obtained based on the analysis of differential equations of motion of the system. To do this, it is necessary to determine under what conditions the original system has a solution of the form:

$$\varphi_i = \delta_i \omega_t + \alpha_1, i = (1,...,S) \tag{1}$$

where: S is the number of vibration exciters in the system; φ_i – Angular coordinates of vibration exciter rotors; ω – synchronous frequency; δ_i – a number that takes on a value $\pm i$, depending on the direction of rotation of the *i*-th rotor; α_i – Phase angle of the *i*-th rotor (in the general case α_i =const).

To find out the conditions under which stable solutions of the form (1) exist in the original system, the Poincare-Lyapunov small parameter method is most often used [6]. Its essence in relation to the problem of self-synchronization is that some terms of the equations of the original system are assumed to be arbitrarily small and correspondingly assigned to them an infinitely small order μ. In this case, the parameter must be introduced into the terms of the equations describing the interaction of vibration exciters in the system. Then, neglecting the small terms of the equations (assuming μ =0), proceed to solving a simplified, so-called generating system, from which periodic solutions with a period are determined $T = 2/\pi$ up to arbitrary constants α_i i=(1,…, S) The latter are found from special equations obtained taking into account the assumption that μ can take arbitrarily small values. These conditions, which establish a certain relationship between the single angles of rotation of the rotors, are called the conditions for the existence of synchronous movements of the original system.

The above methods for solving the problem of self-synchronization of vibration exciters show that the difficulty of finding the conditions for the existence and stability of synchronous movements is determined, first of all, by the type of the original system. For quasilinear systems, these conditions can often be obtained in analytical form. In some cases, analytical solutions are also possible for some nonlinear systems. In the general case, the solution can only be obtained by calculating the system on a computer. A systematic presentation of the theory of synchronization of vibration exciters, as well as a review of relevant studies, are given in the works of I.I. Blekhman [4].

The use of the phenomenon of self-synchronization opens up the possibility of significant improvements in drive devices of vibration machines, making it possible to eliminate highly noisy and quickly worn-out gears, chain drives, and cardan shafts. It turns out to be possible to place vibration exciters at significant distances from each other, etc.

Along with this, in vibration technology there is another wide area of technical applications of the self-synchronization effect. We are talking about the possibility of connecting individual resonant vibration machines into a single system that performs periodic oscillations. This task turns out to be extremely relevant when creating heavy resonant vibrating machines for technological purposes. If it is resolved positively, it becomes possible to quickly change the increase in the size of the working bodies of resonant vibration machines, as well as move to their sectioning. To solve this problem, the following theorem is used [12-21].

Let us consider two elastic discrete one-dimensional systems described by the following linear differential equations with constant coefficients:

$$X_i + \sum_{j=1}^{n} \alpha_{ij} x_j = 0, \ \ (i = 1,\dots, n) \tag{2}$$

$$X_i + \sum_{j=n+1}^{n+m} \alpha_{ij} x_j = 0, \ \ (i = n+1,\dots, n+m) \tag{3}$$

In the future, through $\|\alpha\|$ and $\|\beta\|$ we will denote the coefficient matrices of the system (2) and (3)

Theorem. Let there be two elastic systems defined by coefficient matrices $\|\alpha\|$ and $\|\beta\|$ let there be a frequency P_0 such that $\det\left\|\alpha - EP_0^2\right\| = \det\left\|b - EP_0^2\right\| = 0,$ where E is the identity matrix (i.e. P_0 is the natural frequency of each system).

Then, when any two masses belonging to different systems are connected by an elastic connection of arbitrary rigidity, the frequency P_0 will be the frequency of natural oscillations of the combined system. Then the system turns out to be connected and its characteristic equation can be written in the form:

$$\Delta(P_0) = \det\left\|\frac{\alpha - EP_0^2 + \alpha J_{VV} \qquad -\alpha J_{VS}}{-в J_{SV} \qquad в - EP_0^2 + \beta J_{SS}}\right\|,$$

where, α and β are some constants, and J_{ij} is a matrix located at the intersection of the *i*-th row and the *j*-th column, equal to one. The remaining elements of this matrix are equal to zero.

The characteristic matrix in this case is presented in coagulated form. It is required to show that P_0 – is the frequency of natural oscillations of the system, i.e., that

$$\Delta(P_0) = 0 \tag{4}$$

Relations (4) allow us to obtain such a linear combination of the first η row, which, when added to the *S*-th row, brings the characteristic determinant to the form

$$\Delta(P_0) = \det\left\|\frac{\alpha - EP_0^2 + \alpha J_{VV} \qquad -\alpha J_{VS}}{0 \qquad в - EP_0^2 + \beta J_{SS}}\right\| \tag{5}$$

Determine the coagulated matrix can be calculated based on the relation $\det\left\|\frac{A\ B}{C\ D}\right\| = \det\left\|D - CA^{-1}\right\|$ provided that $\det\|A\| = 0$.

Since in case (5) the matrix is zero, and det $\|D\|$, according to (3), is equal to zero, then $\Delta(P_o) = 0$ and the theorem is proven.

This theorem states that if there are two vibration machines with resonant tuning to the same frequency, then the resonant tuning is preserved even after these machines are in some way combined into one common system.

Mathematical model of the system. As dynamic systems, we consider the possibility of self-synchronization of two suited resonant vibrating machines connected by an elastic connection. The diagram of the system under consideration is presented in Figure 1.

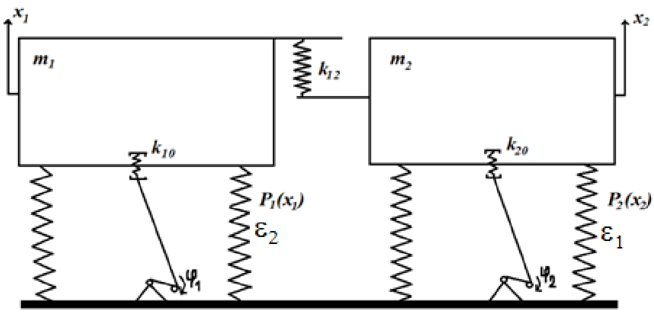

Fig. 1. Design diagram of a sectioned resonant vibrating machine

We will assume that each block is an elastic system with one degree of freedom. In the system under consideration, vibrations are excited by eccentric vibrators with elastic elements in the connecting rod. The elastic restoring characteristics of the main elastic bonds will be described by relations of the form:

$$P_0(x_i) = k_i x_i + \beta_i x_i^3 \ (i = 1,2). \tag{6}$$

We will assume that energy dissipation in the main elastic bonds obeys the viscous Kedwin-Voigt hypothesis. We will assume the elastic characteristics of the drive shock absorbers to be linear, and we will neglect the energy dissipation in them. The indices for the coefficients will indicate that the characteristics of the corresponding elastic connection belong to each other. We will accept the characteristics within the framework of the V.O. Kononeiko model. Then, for the design scheme presented in Figure 1, under the assumptions made above, the differential equations of motion can be written in the form:

$$\begin{aligned} &m_1\ddot{x}_1 + \varepsilon_1\dot{x}_1 + (k_{10} + k_1)x_1 + \beta_1 x_1^3 + k_{12}(x_1 - x_2) = k_{10}\rho\sin\varphi_1, \\ &m_2\ddot{x}_2 + \varepsilon_2\dot{x}_2 + (k_{20} + k_2)x_2 + \beta_2 x_2^3 - k_{12}(x_1 - x_2) = k_{20}\rho\sin\varphi_2, \\ &J_1\ddot{\varphi}_1 + g_1\ddot{\varphi}_1 = M_{10} - b_1\dot{\varphi}_1 + k_{10}\rho(x_1 - \rho\sin\varphi_1)\cos\varphi_1, \\ &J_2\ddot{\varphi}_2 + g_2\ddot{\varphi}_2 = M_{20} - b_2\dot{\varphi}_2 + k_{20}\rho(x_1 - \rho\sin\varphi_2)\cos\varphi_2, \end{aligned} \tag{7}$$

where: m_1, m_2 are the masses of the working parts; J_1, J_2 – moments of inertia reduced to the cranks; k_{10}, k_{20} – rigidity of drive elastic connections; φ_1, φ_2 – angles of rotation of the cranks; x_1, x_2 – movements of the centers of mass of the working bodies; k_{12} is the rigidity of the elastic connection connecting the working bodies; M_{10} - b_1 φ_1, M_{20} - b_2 φ_2 torque characteristics of engines in the working area, reduced to cranks; ρ – crank radius (eccentricity).

Energy dissipation in elastic elements is also taken into account. In this case, the hypothesis of viscous friction was accepted and the dissipation coefficients were assumed to be equal ε_1 and ε_2. To calculate the torque reduced to the crank developed by the electric motor at a constant gear ratio between the engine and the crank, you can use the expression $M_{gbi} = M_{io}U + b_i\dot{\varphi}_i U^2$, where U is the gear ratio. To speed up calculation work, a table of coefficients is proposed in the work M_{i0}, b_i for different engine powers.

Approximate solution to the problem of self-synchronization of a dynamic system with two degrees of mobility. System of equations (7) is autonomous. Determining the parameters of periodic solutions of such systems is significantly complicated by the need to clarify the period of this solution at each stage of calculations. At the same time, there is always the possibility of reducing these problems to constructing a periodic solution of some auxiliary non-autonomous system, the order of which is one unit lower than the order of the original system.

$$\ddot{x}_i = \frac{d^2 x_i}{dt^2} = \frac{d}{dt}\left(\frac{dx_i}{dt}\right) = \frac{d\varphi_1}{dt}\frac{d}{d\varphi_1}\left(\frac{d\varphi_1 dx_i}{dt\, d\varphi_1}\right) = \omega_1^2\frac{d^2 x_i}{d\varphi_1^2} + \omega_1\frac{d\omega_1}{d\varphi_1}\frac{dx_i}{d\varphi_1},$$

$$\ddot{\varphi}_i = \frac{d^2\varphi_i}{dt^2} = \omega_1\frac{d^2\varphi_i}{d\varphi_1^2} + \omega_1\frac{d\omega_1}{d\varphi_1}\frac{d\varphi_i}{d\varphi_1} \quad \text{(i=1, 2)} \tag{8}$$

After transforming the system, equation (7) taking into account (8) will take the form:

$$m_1\left(\omega_1^2\frac{d^2x_1}{d\varphi_1^2}+\omega_1\frac{d\omega_1}{d\varphi_1}\frac{dx_1}{d\varphi_1}\right)+\varepsilon_1\omega_1\frac{dx_1}{d\varphi_1}+(k_{10}+k_{12})x_1+\beta_1x_1^3+k_{12}(x_1-x_2)=k_{10}\rho\sin\varphi_1,$$
$$m_2\left(\omega_1^2\frac{d^2x_2}{d\varphi_1^2}+\omega_1\frac{d\omega_1}{d\varphi_1}\frac{dx_2}{d\varphi_1}\right)+\varepsilon_2\omega_1\frac{dx_2}{d\varphi_1}+(k_{20}+k_{12})x_2+\beta_2x_2^3-k_{12}(x_1-x_2)=k_{20}\rho\sin\varphi_2,$$
$$J_1\omega\frac{d\omega_1}{d\varphi_1}+g_1\omega_1=M_{10}-b_1\omega_1+k_{10}\rho(x_1-\rho\sin\varphi_1)\cos\varphi_1, \tag{9}$$
$$J_2\left(\omega_1\frac{d^2\varphi^2}{d\varphi_1}+\omega_1\frac{d\varphi_2}{d\varphi_1}\frac{d\omega_1}{\varphi_1}\right)+g_2\omega_1\frac{d\varphi_2}{d\varphi_1}=M_{20}-b_2\omega_1\frac{d\varphi_2}{d\varphi_1}+k_{20}\rho(x_2-\rho\sin\varphi_2)\cos\varphi_2,$$

where, $\omega_1=\frac{d\omega_1}{dt}$.

The main features of the behavior of such a system can be established by analyzing approximate solutions of relatively low orders. To a first approximation, the periodic solution (9) has the form:

$$X_1=A\cos\varphi_1+B\sin\varphi_1=H_1\cos(\varphi_1-\delta_1);\ X_2=C\cos\varphi_1+D\sin\varphi_1=H_2\cos(\varphi_1-\delta_2);$$
$$\omega_1=M\cos2\varphi_1+N\sin2\varphi_1+\omega_0;\ \ \varphi_2=L\cos2\varphi_1+G\sin2\varphi_1+\varphi_1+\alpha; \tag{10}$$

where: α – phase shift angle, $\alpha\mathrm{H}_1=\sqrt{A^2+B^2}$, $\mathrm{H}_2=\sqrt{C^2+D^2}$.

The determination of the parameters of the approximate solution is determined based on the Galerkin-Bubnov method. The conditions of harmonic balance lead to a comprehensive system of equations for $A, B, C, D, M, N\ G, L, \omega_0, \alpha$.

$$\left[k_{10}+k_1+k_{12}-m_1\omega_0^2+\frac{1}{2}m_1(M^2+N^2)+\frac{3}{4}\beta_1(A^2+B^2)\right]A+\varepsilon_1\omega_0B-k_{12}C=0;$$
$$\left[k_{10}+k_1+k_{12}-m_1\omega_0^2+\frac{1}{2}m_1(M^2+N^2)+\frac{3}{4}\beta_1(A^2+B^2)\right]B-\varepsilon_1\omega_0A-k_{12}D=k_{10}\rho;$$
$$\left[k_{20}+k_2+k_{12}-m_2\omega_0^2+\frac{1}{2}m_2(M^2+N^2)+\frac{3}{4}\beta_2(C^2+D^2)\right]C+\varepsilon_2\omega_0D-k_{12}A=$$
$$=k_{20}\rho\sin\alpha+\frac{1}{2}k_{20}\rho\ G\sin\alpha-\frac{1}{2}k_{20}\rho\ L\cos\alpha;$$
$$\left[k_{20}+k_2+k_{12}-m_2\omega_0^2+\frac{1}{2}m_2(M^2+N^2)+\frac{3}{4}\beta_2(C^2+D^2)\right]D-$$
$$-\varepsilon\omega_0C-k_{12}B=k_{20}\rho\cos\alpha+\frac{1}{2}k_{20}\rho\ L\cos\alpha-\frac{1}{2}k_{20}\rho\ G\sin\alpha;$$
$$2J_1\omega_0N+B_1M=\frac{1}{2}k_{10}\rho\ A;\ \ -2J_1\omega_0M+B_1N=\frac{1}{2}k_{10}\rho\ B-\frac{1}{2}k_{10}\rho^2;$$
$$(g_1+b_1)\omega_0-M_{10}=\frac{1}{2}k_{10}\rho\ A;$$
$$2J_2\omega_0N+b_2M-\left[2J_2(M^2+N^2)+4J_2\omega_0^2\right]L+2b_2\omega_0G=\frac{1}{2}k_{20}\rho\ (G-DL)\cos\alpha+$$
$$+\frac{1}{2}k_{20}\rho\ (D-CL)\sin\alpha-\frac{1}{2}k_{20}\rho^2\ \sin2\alpha;$$

$$-2J_2\omega_0 M + b_2 N - \left[2J_2(M^2+N^2)+4J_2\omega_0^2\right]G - 2b_2\omega_0 L = \frac{1}{2}k_{20}\rho\,(G-CD)\sin\alpha -$$
$$-\frac{1}{2}k_{20}\rho\,(L-DG)\cos\alpha - \frac{1}{2}k_{20}\rho^2\cos 2\alpha;$$
$$(g_2+b_2)\omega_0 - M_{20} + b(MG-NL) - 2J_2\omega_0(ML+NG) = \frac{1}{2}k_{20}\rho(C-CG)+$$
$$+\frac{1}{2}k_{20}\rho\left(D-DG+\frac{1}{2}CL\right)\sin\alpha - \frac{1}{2}k_{20}\rho^2(L\cos 2\alpha - G\sin 2\alpha).$$

In order to simplify the calculation on a computer, the system (11) is collapsed. At the same time, the system of equations obtained by condensing (11) turned out to be very cumbersome and did not lead to the desired result. Therefore, the following assumptions are made in the form of solution (10):

$$\omega_1 = \omega_0 = const;\ \ \varphi_2 = \varphi_1 + \alpha. \tag{12}$$

After this, the system will look like this:

$$\begin{aligned}
&\left[k_{10}+k_1+k_{12}-m_1\omega_0^2+\frac{3}{4}\beta_1(A^2+B^2)\right]A+\varepsilon_1\omega_0 B - k_{12}C = 0;\\
&\left[k_{10}+k_1+k_{12}-m_1\omega_0^2+\frac{3}{4}\beta_1(A^2+B^2)\right]B-\varepsilon_1\omega_0 A - k_{12}D = k_{10}\rho;\\
&\left[k_{20}+k_2+k_{12}-m_2\omega_0^2+\frac{3}{4}\beta_2(C^2+D^2)\right]C+\varepsilon_2\omega_0 D - k_{12}A = k_{20}\rho\sin\alpha;\\
&\left[k_{20}+k_2+k_{12}-m_2\omega_0^2+\frac{3}{4}\beta_2(C^2+D^2)\right]D-\varepsilon\omega_0 C - k_{12}B = k_{20}\rho\cos\alpha;\\
&(g_1+b_1)\omega_0 - M_{10} = \frac{1}{2}k_{10}\rho\, A;\ \ (g_2+b_2)\omega_0 - M_{20} = \frac{1}{2}k_{20}\rho(C\cos\alpha - D\sin\alpha).
\end{aligned} \tag{13}$$

It is possible to transform system (13) to a system with fewer equations. As a result of simple but cumbersome transformations, system (13) is reduced to a system of two nonlinear algebraic equations with two unknowns ω_0, A^2+B^2.

$$\begin{aligned}
&f_1(\omega_0 A^2+B^2) = \left[\frac{(-a_1^2+\varepsilon_1^2\omega_0^2)(A^2+B^2)-2\varepsilon_1\omega_0 a_4 + k_{12}^2 a_5 - k_{10}^2\rho^2}{2a_1}\right]+\\
&+a_2 - k_{10}^2\rho^2(A^2+B^2) = 0\\
&f_2(\omega_0 A^2+B^2) = \left[\frac{(a_6^2+\varepsilon_2^2\omega_0^2)a_5 + 2\varepsilon_2\omega_0 a_4 + 2a_1 a_6 + k_{12}^2(A^2+B^2) - k_{10}^2\rho^2}{2a_6}\right]^2+\\
&+a_2 - k_{10}^2\rho^2(A^2+B^2) = 0,
\end{aligned} \tag{14}$$

where $a_1 = k_{10} + k_1 + k_{12} - m_1\omega_0^2 + \frac{3}{4}\beta_1(A^2 + B^2),\ a_2 = (g_1 + b_1)\omega_0 - M_{10};$

$a_3 = (g_2 + b_2)\omega_0 - M_{20};$

$a_4 = 2a_2 + \varepsilon_1\omega_0)(A^2 + B^2);\ a_5 = \frac{-2a_2 - 2a_3 - \varepsilon_1\omega_0)(A^2 + B^2)}{\varepsilon_2\omega_0};$

$a_6 = k_{20} + k_2 + k_{12} - m_2\omega_0^2 + \frac{3}{4}\beta.$

Research results and discussion. *Analysis of the results of mathematical modeling of the system.* Solutions of the system of equations (14) make it possible to determine the main parameters of the oscillation process to a first approximation. If the system of equations (14) has real solutions, then this indicates the fundamental possibility of the existence of a self-synchronization effect in the system.

A numerical solution to the system of equations (14) was obtained using a Delphi program. The bisection method was used to solve the system.

The specific object of research was a system with the following parameter values: m_1, m_2 – masses of working bodies; J_1, J_2 – moments of inertia reduced to the cranks; k_{10}, k_{20} – rigidity of drive elastic connections; φ_1, φ_2 – angles of rotation of the cranks; x_1, x_2 – movements of the centers of mass of the working bodies; k_{12} is the rigidity of the elastic connection connecting the working bodies; elastic connection stiffness k_1, k_2.

$m_2 = 7800\ kg;\ k_1 = k_2 = 6.05 \cdot 10^7\ N/m;$

$k_{10} = k_{20} = 8 \cdot 10^7\ N/m;\ \beta_1 = \beta_2 = 10^{11}\ N/m^3;\ \varepsilon_1 = \varepsilon_2 = 8 \cdot 10^5\ s \cdot N/m;$

$\rho = 0.015m;\quad J_1 = J_2 = 5.85 H \cdot м/s^2;\ g_1 = g_2 = 2.2\ N \cdot m \cdot s;$

$M_{10} = M_{20} = 10773.6\ N \cdot m;\ b_1 = b_2 = 173.86\ N \cdot m \cdot s.$

Values of stiffness k_{12}, mass m_1 varied within the following limits: $k_{12} = 10^7\ N/m \div 2 \cdot 10^8\ N/m$ in increments $5 \cdot 10^6\ N/m,\ m_1 = 3800\,kg \div 7800\,kg$ in increments of 500 kg.

This was carried out with the aim of assessing the possibility of locking the system into self-synchronization mode when parameters change. Varying the value k_{12} is due to the fact that when drawing up the equation of motion, it was assumed that the form installed on the working bodies with (or without) technological load does not ensure their transformation into one absolutely rigid body.

In this regard, an elastic connection with stiffness was introduced into the mathematical model k_{12}, which can take any value, given the practical impossibility of its control.

It was found that for all the considered parameter values, the system reached a periodic oscillation mode. Figure 2 shows the dependence of the phase shift angle α rotation of the cranks in a steady state of motion of the system due to rigidity k_{12} at different values of λ.

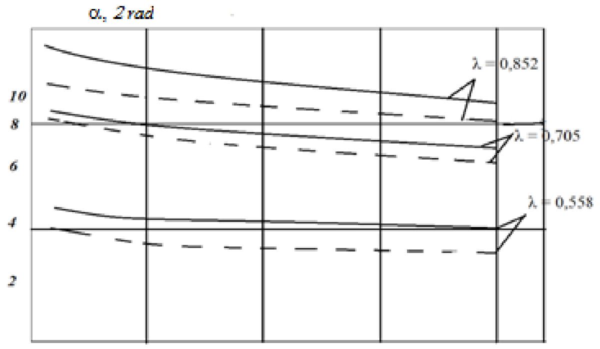

Fig. 2. Dependence of phase shift angle λ on stiffness k_{12} at different values of λ

Here λ mass ratio, i.e. $\lambda = \frac{m_1}{m_2}$. Analysis of the results obtained shows that with an increase in connecting stiffness k_{12} there is a decrease in the phase shift angle α. In the limit during the transition to an absolutely rigid connection of masses m_1 and m_2 The operating mode of vibration exciters tends to be in phase. This fact should be taken into account in the practical use of the self-synchronization effect in resonant machines, especially in heavy machines for technological purposes.

The graphs provided in Figure 3 shows that the average angular velocity of rotation of the cranks in steady motion changes relatively little when the mass changes m_1. A general increase in the masses of the working bodies leads to a slight decrease in speed ω_0. During calculations it was found that the rigidity k_{12} has virtually no effect on the average angular speed of rotation of the cranks. Change in hardness and magnitude $\lambda = \frac{m_1}{m_2}$ also does not have a significant effect on the vibration amplitudes of the working bodies.

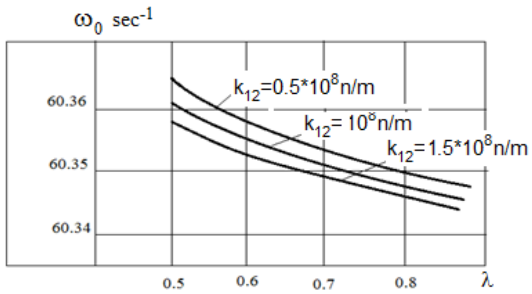

Fig. 3. Dependence of the average angular velocity ω_0 on the ratios mass $\lambda=m_1/m_2$ at different values of k_{12}

Since there was a problem of assessing the acceptability of representing approximate solutions in the form (7), direct integration of the original system of differential equations was also carried out with the solutions reaching the mode of steady motion.

In this case, the integration was carried out by the Runge-Kutta method, which turns out to be satisfactory in terms of calculation stability. To solve on a computer, system of equations (7) is used in conical form. For this, the following notation has been introduced:

$$x_1 = y_1; \quad \varphi_1 = y_5; \quad x_2 = y_2; \quad \varphi_2 = y_5; \quad \dot{x}_1 = y_3; \\ \dot{\varphi}_1 = y_7; \quad \dot{x}_2 = y_4; \quad \dot{\varphi}_2 = y_8. \tag{15}$$

After substituting (15) into (7), performing some transformations, we obtain:

$$F_1 = y_3; \; F_2 = \frac{1}{m_1}\left[k_{10}\rho \sin y_5 - (k_{10} + k_1)y_1 - \beta_1 y_1^3 - \varepsilon_1 y_3 - k_{12}(y_1 - y_2)\right];$$
$$F_3 = y_4; \; F_4 = \frac{1}{m_2}\left[k_{20}\rho \sin y_6 - (k_{20} + k_2)y_2 - \beta_2 y_2^3 - \varepsilon_2 y_4 + k_{12}(y_1 - y_2)\right];$$
$$F_5 = y_7; \; F = \frac{1}{J_1}\left[k_{10}\rho(y_1 - \rho \sin y_5)\cos y_5 + M_{10} - b_1 y_7 - g_1 y_7\right]; \tag{16}$$
$$F_6 = y_8; \; F_8 = \frac{1}{J_2}\left[k_{20}\rho(y_2 - \rho \sin y_6)\cos y_6 + M_{20} - (b_2 + g_2)y_8\right].$$

To implement the Runge-Kutta method scheme, subroutines were used to enter the initial parameters of the problem and initial conditions. The subroutine, composed of standard elements, uses the Runge-Kutta method scheme. The subroutine also includes a procedure for calculating the right-hand sides of the system of equations (7).

Solutions of the system of equations (7) correspond to the following initial conditions:

$$x_1 = x_2 = 0.002м; \quad \varphi_1 = \varphi_2 = 0; \quad \dot{\varphi}_1 = \dot{\varphi}_2 = 60c^{-1}; \quad \dot{x}_1 = \dot{x}_2 = 0;$$

The initial integration step was taken equal to 0.005 s for small values; 0.001 s for large k_{12}, k_{10}, k_{20}, k_1, k_2, β_1, β_2.

The results of calculations carried out on the basis of approximate relations and obtained as a result of direct integration of the original system of differential equations showed good agreement. Some of the calculation results are shown in Fig. 2.

In Figure 2 solid lines show the results obtained by the approximate method, and dotted lines show the results of direct integration.

The conducted research shows that resonant suited vibrating machines, when combined into a single system, relatively easily enter the self-synchronization mode and this mode is stably maintained when a number of system parameters change within a relatively wide range. This makes it possible to widely use the principle of sectioning when creating resonant vibrating machines and, in some cases, create heavy resonant vibrating machines from standard modules.

Experimental research. To carry out the experiments, a specially made steel mold *2* measuring 10x30x10 was rigidly attached to the vibration stand table. The steel mold was filled to the required level with concrete mixture *3*, after which the surface of the mixture was leveled (Fig. 4). On top, to compact the concrete mixture, a steel plate *4* was placed with two rigidly welded posts *5*, onto which springs with different diameters and stiffness coefficients were placed. Specially made cylindrical weights *6* are installed on the springs, freely moving back and forth along the racks;

the source of sound frequency oscillations for the VEDS-400 vibration stand (Fig. 5) was a GB-27 type generator.

The signal coming from the generator was amplified and the operation of the vibration stand *1* was controlled, respectively, using a stabilized amplifier and a SUPV-01A control panel.

DU-5 acceleration sensors were attached to the cylindrical weights. The same sensors are attached to the vibration table table. Using shunt and resistance magazines, the sensors were calibrated so that their readings, with the same specified mechanical parameters of the system, were equal to each other.

The acceleration of vibrations of the vibration stand table using the DM-162 vibration station varied from 0 to 4.5.

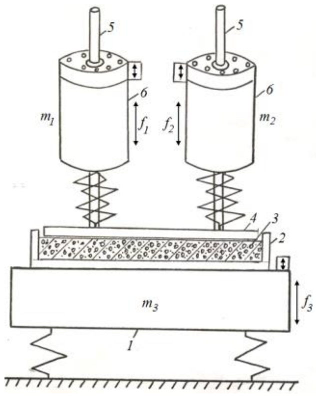

Fig. 4. Vibration stand


Fig. 5. Vibration stand VZDS-400

The experimental part of the work, which was carried out on a laboratory installation, can be divided into 3 main stages:

First stage: Conducting experiments usingweights installed on the plate (body).

Second phase: Conducting experiments using weights installed on equal plates (bodies).

Third stage: Experimental testing of the possibility of using self-synchronizing weights for compacting more rigid concrete mixtures.

First stage. At the beginning of the stage, weights with the same masses and with nonlinear elastic connections operating in a resonant mode were considered. Tuning to the resonant mode of oscillation of weights was carried out visually by rotating the knob of the frequency regulator of the sound generator in the range from 1 to 100 Hz. The resonant frequency value was measured with a 43-32 frequency meter. At the same time, stable synchronous and in-phase movement of the loads was observed. Even after an artificial stop of one of the loads, synchronism and phase alignment were instantly restored; to establish the phase relationships of the loads, similar experiments were carried out for various mass ratios of the loads (m_1/m_2 = 1.05; 1.1; 1.2; 1.3) and for different accelerations of kinematic excitation (Ag=0.3g; 0.5g; 1g; 1.5g).

At the end of the stage, similar experiments were repeated for a system with linear elastic connections. At the same time, in the resonant mode, such violations of the synchronism and in-phase movement of the weights were observed, such as the damping of the oscillations of one of the weights until a complete stop or the movement of the weights in ant phase. In conclusion of the experiment, it should be noted that the non-resonant oscillation mode of a system with two weights does not ensure self-synchronization of these loads. And in this mode there can be no question of using weights to compact more rigid concrete mixtures.

Second phase. The formulation of such an experimental problem aims to study the influence of the fact of separation of the plate on which weights are installed on the effect of self-synchronization of their oscillations. At this stage, the entire cycle of experiments of the first stage was repeated and results similar to the results of the first stage were obtained. That is, stable synchronous and in-phase movement of the loads was observed in the resonant mode and with nonlinear elastic connections.

In addition, in both the first and second stages, the influence of the rigidity and thickness of the concrete mixture layers on the synchronism and in-phase movement of the loads was considered. It was found that these factors did not have a significant impact on the self-synchronization effect.

Third stage. To compact rigid concrete mixtures, effective external mechanical action (vibration) is necessary, and for particularly rigid (heavy, fine-grained, soil concrete) concrete mixtures, the use of vibration compaction in combination with a weight is required.

Experience in using vibration compaction technology with an inertial load indicates its high efficiency and opens up wide possibilities for application on existing and designed lines for the production of reinforced concrete products.

The question of the possibility of using two or more small-sized weights in the manufacture of long-length reinforced concrete products in order to increase their strength characteristics is of interest.

For strength testing, control samples were made from more rigid concrete mixtures, compacted using standard technology and using technology using two self-synchronizing weights. Two concrete mixture compositions were used: a normal mixture and a mixture with a reduced cement content. Using standard technology (on a VZDS-400 vibration stand, Fig. 5), five samples were molded from a mixture of normal composition; using the proposed technology, five samples from a mixture of normal composition and five samples from a mixture with a reduced cement content.

After 28 days of normal hardening, the samples were tested for compression on a PSU-50 testing machine. Based on the test results, the compressive strength limits of the specimens Рсж were determined, which are presented in Table. 1.

Analysis of tabular data indicates an increased efficiency of the vibration-impact mode of vibration seals compared to the standard harmonic one. This is expressed as follows.

Firstly, the use of shock vibration, achieved with the help of self-synchronizing loads, instead of the usual harmonic one, allows (without changing the composition of the concrete mixture) to increase the compressive strength of concrete by an average of 25-31%.

Secondly, the shock mode of vibration makes it possible to reduce consumption by 10.1 (compared to its content in a mixture of normal composition) and at the same time not “lose” in the strength of concrete.

Thirdly, the proposed method of vibration compaction of the concrete mixture ensures the production of concrete products of a higher density than with known methods, as well as a decrease in water absorption and an increase in the frost resistance of products.

Table 1

Strength test results

	Vibration compaction using standard technology		Vibration compaction using weights			
	Normal mixture		Normal mixture		Mixture with reduced cement content	
	400 kg/m^3	300 kg/m^3	400 kg/m^3	300 kg/m^3	360 kg/m^3	270 kg/m^3
1	20.1	13.5	30.2	20.4	21.4	14.1
2	19.4	14.2	29.4	19.2	21.1	15.0
3	22.2	12.8	28.6	21.3	23.4	13.6
4	22.0	14.3	28.0	20.6	21.3	14.5
5	23.3	14.4	27.8	19.4	21.8	15.3
Average	21.4	13.6	28.8	20.18	21.8	14.56

A Delphi program was compiled that allows visual observation of the movements of the dynamic system under consideration (Fig. 6). The program allows you to vary the basic parameters of the system. In addition, a program has been compiled in the Maple language that allows obtaining, on the basis of the scheme under consideration, the kinematic parameters of a dynamic system. Figures 7 and 8 show the synchronous movement of blocks.

The results show that the self-synchronizing mode in these machines is maintained by the device when changing a number of system parameters within a relatively wide range.

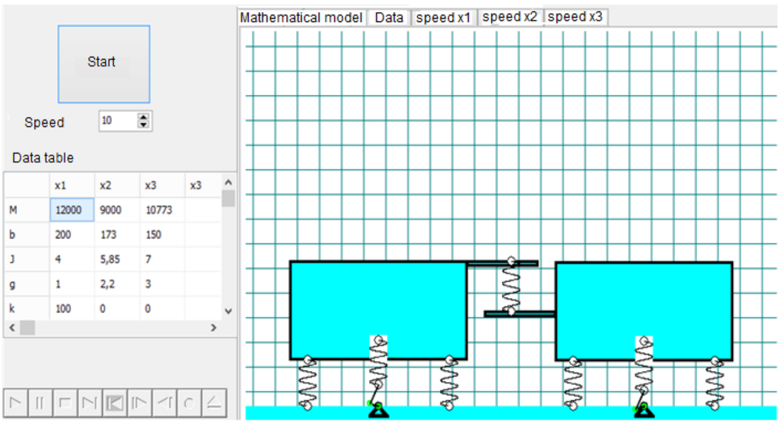

Fig. 6. Observation of the movements of the dynamic system under consideration

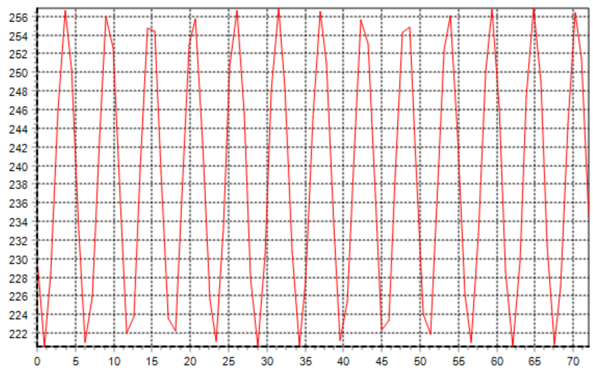

Fig. 7. Moving the first block

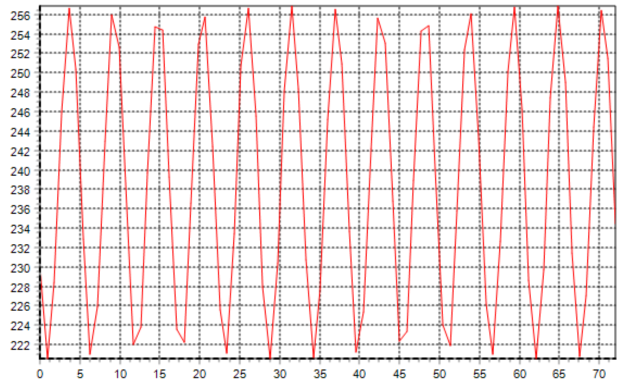

Fig. 8. Moving the second block

Conclusion. The problems of self-synchronization of a dynamic system of a resonant type with two degrees of freedom are considered. A theory is given about the property of the natural frequencies of sectioned systems. An approximate periodic solution corresponding to the self-synchronization mode is constructed. The parameter ranges leading to a periodic regime are investigated. The possibility of block design of resonant vibrating machines for technological purposes has been established. Mathematical modeling of transient and stationary modes of a number of resonant vibrating machines in block design has been carried out. The results of mathematical modeling of the problem using numerical methods (Maple, Delphi) are presented. The use of the results of theoretical studies makes it possible to abandon the rigid kinematic synchronization of the rotors of vibration systems, which leads to a reduction in the energy intensity of the equipment. A methodology has been proposed and devices have been developed for experimental research of the basic laws and characteristics of the technological process of vibratory compaction of concrete mixtures. It has been established that this method makes it possible to obtain concrete products of higher density than with known methods. In this case, water absorption decreases by 15-25%, and strength characteristics increase by 25-31%.

Experience in using vibration compaction technology with an inertial load indicates its high efficiency and opens up wide possibilities for application on existing and designed lines for the production of reinforced concrete products.

The conducted research shows that resonant single-mass vibrating machines, when combined into a single system, relatively easily enter the self-synchronization mode and this mode is stably maintained when a number of system parameters change within a relatively wide range. This makes it possible to widely use the principle of sectioning when creating resonant vibrating machines and, in some cases, create heavy resonant vibrating machines from standard modules.

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ТЕРБЕЛІС МАШИНАЛАРЫНДАҒЫ ДИНАМИКАЛЫҚ ЖҮЙЕНІ ӨЗДІГІНЕН СИНХРОНДАУ

Аңдатпа. Екі еркіндік дәрежесі бар резонанстық үлгідегі динамикалық жүйені өздігінен синхрондау міндеттері қарастырылады. Серпінді жүйелер ретінде серпінді байланыспен байланысқан бір пластиналы резонансты екі тербелісті машинаны өздігінен синхрондау мүмкіндігі қарастырылған. Секцияланған жүйелердің меншікті тербеліс жиіліктерінің қасиеті туралы теория келтірілген. Өзін-өзі синхрондау режиміне сәйкес келетін жақындатылған мерзімдік шешім құрылған. Кезеңдік режимге әкелетін параметрлер аумағы зерттелген. Технологиялық мақсаттағы резонансты тербеліс машиналарын блоктап орындау мүмкіндігі белгіленген. Блоктық орындаудағы резонансты тербеліс машиналарының бірқатар конструкцияларының өтпелі және стационарлық режимдерін математикалық модельдеу жүргізілген. Жұмыста сандық әдістермен математикалық модельдеу (Maple, Delphi) нәтижелері келтірілген. Теориялық зерттеулердің нәтижелерін пайдалану тербеліс жүйелерінің роторларын қатаң кинематикалық синхрондаудан бас тартуға мүмкіндік береді, бұл жабдықтың энергия сыйымдылығын азайтуға алып келеді. Бетон қоспасын тербеліс тығыздаудың технологиялық процесінің негізгі заңдылықтары мен сипаттамаларын эксперименттік зерттеуге арналған әдістеме ұсынылып, құрылғылар әзірленген. Жүргізілген зерттеу резонансты бір массалы тербеліс машиналары оларды бірыңғай жүйеге біріктірген кезде салыстырмалы түрде өздігінен синхрондау режиміне оңай кіретінін және бұл режим салыстырмалы түрде кең ауқымда жүйенің бірқатар параметрлері өзгерген кезде тұрақты ұсталатынын көрсетген. Бұл резонансты тербеліс машиналарын жасау кезінде секциялау қағидатын кеңінен пайдалануға және бірқатар жағдайларда стандартты модульдерден ауыр резонансты тербеліс машиналарын жасауға мүмкіндік береді. Бұл әдіс белгілі тәсілдерге қарағанда неғұрлым жоғары тығыздықтағы бетон бұйымдарын алуға мүмкіндік беретіні анықталды. Бұл ретте су сіңіру 15-25%-ға азаяды, ал беріктік сипаттамалары 25-31%-ға ұлғаяды.

Тірек сөздер: синхрондау, тербелмелі машиналар, тербеліс күшейткіштер, тербеліс алаңдары, серпімді байланыстар, автономды жүйелер.

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САМОСИНХРОНИЗАЦИИ ДИНАМИЧЕСКОЙ СИСТЕМЫ В ВИБРАЦИОННЫХ МАШИНАХ

Аннотация. Рассматриваются задачи самосинхронизации динамической системы резонансного типа с двумя степенями свободы. В качестве динамических систем рассматривается возможность самосинхронизации двух одномастных резонансных вибромашины, связанных упругой связью. Приводится теория о свойстве частот собственных колебаний секционированных систем. Строится приближенное периодическое решение, соответствующее режиму самосинхронизации. Исследуются области параметров, приводящие к периодическому режиму. Установлена возможность блочного исполнения резонансных вибромашин технологического назначения. Проведено математическое моделирование переходных и стационарных режимов ряда конструкции резонансных вибромашин в блочном исполнении. Приводится результаты математического моделирования задачи численными методами (Maple, Delphi). Использование результатов теоретических исследований позволяют отказаться от жестких кинематических синхронизации роторов вибросистем, что приводит к уменьшению энергоемкости оборудования. Предложена методика и разработаны устройства для экспериментального исследований основных закономерностей и характеристик технологического процесса виброуплотнении бетонной смеси. Проведенное исследование показывает, что резонансные одномассные вибромашины при объединении их в единую систему сравнительно легко входят в режим самосинхронизации и этот режим устойчиво удерживается при изменении ряда параметров системы в сравнительно широких пределах. Это позволяется при создании резонансных вибромашины широко использовать принцип секционирования и в ряде случаев создавать тяжелые резонансные вибромашины из стандартных модулей. Установлено, что данный метод позволяет получить бетонные изделия более высокой плотности, чем при известных способах. При этом водопоглащение уменьшается на 15-25%, а прочностные характеристики увеличивается на 25-31%.

Ключевые слова: синхронизация, вибрационные машины, вибровозбудители, виброплощадки, упругие связи, автономные системы.