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CREATION OF MATHEMATICAL MODEL OF PLATFORM-VIBRATOR WITH SHOCK, DESIGNED FOR CONCRETE PRODUCTS COMPACTION AND MOLDING

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Platform-vibrators are the main molding equipment in the production of precast concrete elements. Shock-vibration technology for the precast concrete production on low-frequency resonant platform-vibrators significantly improves the quality of the products front surfaces and the degree of their factory readiness. This technology is used to produce large elements.

We describe the creation of a mathematical model for platform-vibrator that uses shock to produce asymmetric oscillations. The values of the upper and lower accelerations of the mold with concrete have different values with shock-vibration technology.

The created mathematical model corresponds to the two-body 2-DOF vibro-impact system. It is strongly nonlinear non-smooth discontinuous system. It has some peculiar properties, namely: the upper body with very large mass breaks away from the lower body during vibrational motion; both bodies move separately; the upper body falls down onto the soft constraint; the impact that occurs is soft one due to the softness and flexibility of the constraint. The soft impact simulation requires special discussion. In this paper, we simulate a soft impact by a nonlinear contact force in accordance with the Hertz quasistatic contact law.

The numerical parameters for this system were chosen in such a way that: firstly they provide the fulfillment of requirements for real machine, and secondly they allow analyzing its dynamic behavior by nonlinear dynamics tools. The created model is well enough to fulfill a number of requirements, namely: T -periodic steady-state movement after passing the transient process; the appropriate value of mold oscillations amplitude; the satisfactory value of the asymmetry coefficient that is the ratio of lower acceleration to the upper acceleration. We believe that the created model meets all the necessary requirements.

Keywords: platform-vibrator, shock, vibro-impact, mold with concrete, upper and lower accelerations.

1. Introduction

Molding processes are one of the most important in the manufacture of reinforced concrete structures. Now vibration and shock-vibration technologies for concrete mixtures compaction and concrete products molding have the greatest distribution in the construction industry. This priority is likely to continue in the future. Therefore, the issues of optimizing vibration modes, proper selection of vibration equipment do not lose their significance [1].

At the end of the last century 60s extensive technological researches began to optimize the molding modes. Gradually, it became clear that low-frequency compaction modes have undoubted advantages: they allow obtaining high-density concrete with a shorter compaction time. The asymmetric modes turned out to be very effective. The values of the upper and lower accelerations have different values under these regimes. Such modes are called shock-vibrational, and technology – shock-vibration. For their implementation, low-frequency resonant platform-vibrators were created. In recent years, shock-vibration modes have been widely used in various fields of technology.

Shock-vibration technology for the precast concrete production on low-frequency resonant vibratory platforms significantly improves the quality of the products front surfaces and their factory readiness degree. Platform-vibrators are the main molding equipment in the production of precast concrete elements [2]. Such equipment are produced in several factories in Russia.

Studies have confirmed that low-frequency resonant vibration platforms and shock-vibration technology allow to obtain higher quality products in comparison with other vibration platforms.

A complex process of concrete mixture particles interaction with each other occurs under vibration influence. Many aspects of this process are not well understood.

The proposed models and the corresponding equations of the concrete mixture state, the criteria for the compaction effectiveness and the front surfaces quality remain debatable [3].

Many experiments have been conducted to study various aspects of the concrete compaction process.

The magnitude of the working body acceleration w was taken as one of the most important factors affecting the compaction process. It determines the values of dynamics strengths in machine elements to a large extent. So, it links technology process characteristics and strength machine characteristics.

If the working body makes asymmetric oscillations, it is advisable to take into account the upper acceleration w_U and the lower one w_L . Upper acceleration is the acceleration of the mold with concrete at its highest position, and lower acceleration is the acceleration of the mold in its lowest position [4].

Studies were carried out at various values of the working body oscillations frequency.

Experiments with concrete mixtures of other compositions have confirmed that the concrete macrostructure formation is faster and better at a lower frequency and increased amplitude of the working body vibrations.

The goal of the paper is to create a mathematical model of platform-vibrator with shock and to select its numerical parameters so that:

- a) the model maximally corresponded to the requirements for a real machine;
- b) the model made it possible to analyze its dynamic behavior using nonlinear dynamics methods.

2. Two body platform-vibrator with shock

When developing resonant vibratory machines, designers have to take into account a large number of various requirements presented by modern production to new equipment. The realization of close to optimal operating mode is one of the most significant and fundamental. Its successful implementation is largely determined by the successful choice of the principle vibro-machine scheme and, as a consequence of this, its design scheme.

The specifics of many dynamic schemes was carefully analyzed in the work on resonant vibratory machines for compaction of concrete mixtures. It turned out that the conditions for optimal functioning can be satisfied by relatively simple two-mass systems. The creation of vibroforming machines with the number of main moving masses more than two is impractical, since this leads to an unjustified complication of the machines design.

The two-mass platform-vibrator with shock is one of the successful solutions for vibration equipment that implements shock-vibration technology for concrete mixtures compaction and reinforced products molding [5]. Its principled scheme is shown in Fig.1.

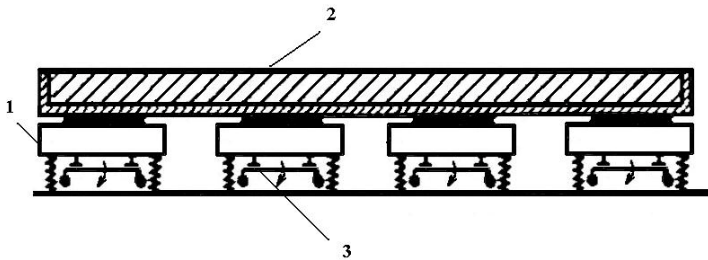


Fig.1 Principled scheme of platform-vibrator:
1 – working body; 2 – mold with concrete; 3 – vibration exciter.

The shock-vibration platform of block type consists of separate blocks on which rubber plates limiters are fixed. The mold with concrete mix is installed on the stops without fastening. With oscillations, the mold breaks away from the limiters and then falls on them. The mold collides with the limiters with oncoming movement. Two general vibration exciters are installed on each block.

One of the most important factors affecting the compaction process is the value of the working body acceleration w . More precisely, the ratio of the lower acceleration w_L to the upper w_U is important.

This can be explained as follows. With vertical oscillations, separation of the mixture from the pallet is possible only when the inertial forces applied to the particles of the concrete mixture act upward. The mixture is pressed to the mold pallet when inertial forces on the particles act downward [4].

The process of concrete mixture compaction is made more intensive due to such asymmetric vertical form vibrations. Accelerations w_U that tear off

mixture from the form pallet become smaller, and the pressing accelerations w_L become larger with such oscillations. These asymmetric vibrations can be obtained precisely in shock-vibration compaction machines [4].

It is necessary to take into account the influence of concrete mix on the machine dynamics when calculating vibration compaction machines. So, it is necessary to consider the compacting machine and concrete mixture as a single dynamic system. But the concrete mixture is a complex viscoplastic medium. It has some elastic properties in the presence of air (especially in the initial period of compaction). All this makes it extremely difficult to solve a single dynamic system "compacting machine – concrete mixture". The question about the nature of the machine interaction with the medium being processed remains the most complex and least investigated. However, the influence of the concrete mixture can be taken into account as an attached mass and some additional damping when practical calculating and, in particular, determining the amplitude-frequency responses of a vibratory machine [3].

The forces of resistance, despite their relative smallness, play a significant role in resonant and close to them oscillation modes. These forces are "used" by the system to compensate the energy coming from the external load. Therefore, it is necessary to take into account the resistance force in the resonance zone. They have not significant effect on the result outside this zone.

The attenuation coefficients are calculated from experimental data. One can assume that the dissipation energy is relatively small and significantly affects only on the resonating harmonic. Such an assumption may simplify the problem. However, accounting for energy dissipation in elastic elements is very important when studying the vibrations of an elastic system in the resonance region. Internal friction is determined by a number of factors. Their influence is difficult to take into account directly. There are many hypotheses to describe dissipative forces. The most widely used is the Kelvin-Voigt hypothesis. It is often called the viscous friction hypothesis. It suggests that dissipative forces are proportional to the strain rate of elastic bonds [6].

The parameters of the exciting force and of the system (the mass of the frame with the mold and concrete mixture, the stiffness coefficients of the vibration limiters, etc.) are selected so that the machine operating mode is close to resonant. That is, so that the natural frequency of the system is close to the frequency of the exciting force.

Resonant vibration platforms have wide possibilities for regulating the modes of working body oscillations. One can change both the amplitude and frequency, and the very law of working body oscillations in the machine tuning process.

The presence of two main moving masses allows us to solve two problems: to provide the necessary law of working body movement and to create an effective vibration isolation system. This can be achieved by appropriate selection of the elastic bonds characteristics.

One of the main disadvantages of resonance modes is associated with high system sensitivity to a change in its parameters, which is due to its strong nonlinearity. One can observe such sensitivity in real vibratory machines due to changes in the technological load mass, the characteristics of shock dampers and so on.

The calculation schemes for determining the amplitude-frequency responses of resonant vibration machines are based on assumptions usual to most applied problems of the oscillations theory. The main moving masses are assumed to be absolutely rigid. The masses of elastic bonds are not taken into account due to their relative smallness. Conditions guaranteeing single-axis motion are also fulfilled.

Such assumptions turn out to be quite acceptable and do not introduce significant errors in the final results of resonant vibratory machines calculating.

3. The mathematical model of platform-vibrator with shock

The calculation scheme of platform-vibrator is shown in Fig.2. Exciting force $F(t) = P \cos(\omega t + \varphi_0)$, its period is $T = 2\pi/\omega$.

Platform shock table with mass m_1 is attached to the base by linear vibration isolating spring of stiffness k_1 and a linear dashpot with damping factor c_1 . Exciting external periodic force $F(t)$ is generated by electric motors mounted under the table. Elastic rubber gasket with thickness h and stiffness k_0 is attached to the table. A linear dashpot with damping factor c_0 is placed between the table and the mold. Mold with concrete with mass m_2 is placed on the gasket but is not fixed both to the gasket and to the table. So it can tear herself away from the gasket and bounce. The machine starts their movement when the electric motors begin their work. First, the table and the mold move vertically together. Then the mold comes off from the gasket. The table and the mold are moving separately until the mold falls down onto the rubber gasket. Impact occurs. The bodies move together again until the mold comes off the gasket and so on.

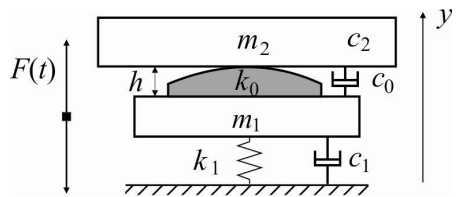


Fig.2

This shock-and-vibration machine is two-body 2-DOF vibro-impact system. The model has the following features: a large mass of the falling body – mold with concrete; softness, flexibility of one contacting surface – rubber gasket; separation of the one body (mold) from another (table with gasket) and their separate motion. One can consider the impact between mold and table with gasket as soft one because of the softness, flexibility of the rubber gasket.

Thus, we consider three states of the platform vibrator: the initial joint movement of both bodies, separate movement in case of loss of contact between

them, and joint movement during the impact due to the form falling onto the gasket.

The forces acting on the bodies are as follows.

The elastic force in spring is

$$F_{k1} = k_1 \Delta l_1 = k_1 (y_1 - \lambda_{st})]. \quad (1)$$

The elastic force in rubber gasket is

$$F_{k0} = k_0 \Delta l_0 = k_0 [h - (y_2 - y_1)]. \quad (2)$$

The origin of coordinate y is chosen in the table centre in the state of static equilibrium. The static deformation of spring is

$$\lambda_{st} = \frac{(m_1 + m_2)g}{k_1}. \quad (3)$$

The damping forces are taken to be proportional to the first degree of velocity:

$$F_{damp1} = c_1 \dot{y}_1, \quad F_{damp0} = c_0 \dot{y}_1. \quad (4)$$

The influence of the concrete mixture can be taken into account as some additional damping $c_2 \dot{y}_2$.

Then the primary joint movement of the table and the form until the first separation is described by the equations:

$$\begin{aligned} m_1 \ddot{y}_1 &= -m_1 g - F_{k1} - F_{damp1} - F_{k0} + F_{damp0} + F(t), \\ m_2 \ddot{y}_2 &= -m_2 g - c_2 \dot{y}_2 + F_{k0} - F_{damp0}. \end{aligned} \quad (5)$$

We introduce the standard notation:

$$\frac{k_1}{m_1} = \omega_1^2, \quad \frac{k_0}{m_2} = \omega_2^2, \quad \frac{c_0}{m_2} = 2\xi_0 \omega_2, \quad \frac{c_1}{m_1} = 2\xi_1 \omega_1, \quad \frac{c_2}{m_2} = 2\xi_2 \omega_2, \quad \frac{m_2}{m_1} = \chi. \quad (6)$$

The equations of primary joint movement will be written as follows;

$$\ddot{y}_1 = g \chi - \omega_1^2 y_1 - \omega_2^2 \chi [h - (y_2 - y_1)] - 2\dot{y}_1 (\xi_1 \omega_1 - \xi_0 \omega_2 \chi) + \frac{1}{m_1} F(t), \quad (7)$$

$$\ddot{y}_2 = -g + \omega_2^2 [h - (y_2 - y_1)] - 2\omega_2 (\xi_2 \dot{y}_2 + \xi_0 \dot{y}_1).$$

The initial conditions are:

$$\text{at } t = 0 \text{ we have } \varphi_0 = 0, \quad y_1 = 0, \quad \dot{y}_1 = 0, \quad y_2 = h - \lambda_0, \quad \dot{y}_2 = 0. \quad (8)$$

The static deformation of gasket is: $\lambda_0 = \frac{m_2 g}{k_0}$.

The equations during the separate movement of bodies are:

$$\begin{aligned} \ddot{y}_1 &= \chi g - \omega_1^2 y_1 - 2\xi_1 \omega_1 \dot{y}_1 + \frac{1}{m_1} F(t), \\ \ddot{y}_2 &= -g - 2\xi_2 \omega_2 \dot{y}_2. \end{aligned} \quad (9)$$

Impact occurs when the mold falls on the gasket. The mold with concrete and the table are moving jointly during impact. The equations of this movement are:

$$\begin{aligned}
\ddot{y}_1 &= g\chi - \omega_1^2 y_1 - 2\xi_1 \omega_1 \dot{y}_1 + \frac{1}{m_1} F(t) + \\
&+ H(z) \left\{ 2\xi_0 \omega_2 \chi \dot{y}_1 - \omega_2^2 \chi [h - (y_2 - y_1)] - \frac{1}{m_1} F_{con}(y_1 - y_2) \right\}, \quad (10) \\
\ddot{y}_2 &= -g - 2\xi_2 \omega_2 \dot{y}_2 + \\
&+ H(z) \left\{ \omega_2^2 [h - (y_2 - y_1)] - 2\xi_0 \omega_2 \dot{y}_1 + \frac{1}{m_2} F_{con}(y_1 - y_2) \right\}.
\end{aligned}$$

Here $H(z)$ is Heaviside step function relatively bodies' rapprochement $z = h - (y_2 - y_1)$.

$$H(z) = \begin{cases} 1, & z \geq 0 \\ 0, & z < 0 \end{cases}. \quad (11)$$

One can see that equations (10) include equations (9). When an impact occurs, $H(z) = 1$ and the terms in figure brackets begin to act.

$F_{con}(y_1 - y_2)$ is contact interactive force that simulates an impact and acts only during an impact. The type of this function requires the special consideration. It can be either a linear or non-linear function. Previously, we studied the impact simulation in vibro-impact systems with various impact types [7,8]. We'll describe the simulation of soft impact in detail in another paper. We'll compare the simulation by linear forces with different proportionality coefficients and the simulation by nonlinear Hertz' force. Now, when we'll choose the numerical system parameters in Section 4, we'll use the nonlinear Hertz's force [9,10] for calculations.

It is worth to point out that Hertz' contact theory requires that the strains in the contact region be sufficiently small to be within the scope of the linear theory of elasticity. "Metallic solids loaded within their elastic limit inevitably comply with this latter restriction. However, caution must be used in applying the results of the theory to low modulus materials like rubber where it is easy to produce deformations which exceed the restriction to small strains" [11]. For an example we have calculated the impact duration in three cases. We have compared the values obtained by known formula [11,12] and by numerical integration of the motion equations (10). The results are shown in Table 1.

Table 1

Moduli of elasticity of contacting bodies		Impact duration by formula [11,12], s	Impact duration by integration of equations (10), s
$E_1, \text{N}\cdot\text{m}^{-2}$	$E_2, \text{N}\cdot\text{m}^{-2}$		
3.5 E+06	2.0 E+11	0.152	0.0135
3.5 E+08	2.0 E+11	0.0256	0.0125
2.0 E+11	2.0 E+11	0.00257	0.00259

Nevertheless, Hertz's contact theory is widely used to model impact. It provides fairly good and reliable results.

The calculation scheme of shock-and-vibration machine corresponds to two-body 2-DOF vibro-impact system. It is strongly nonlinear non-smooth discontinuous system. It changes its structure during oscillatory motion. The right-hand sides of motion differential equations are discontinuous.

4. Selection of numerical system parameters

The basis for the choice of numerical parameters was "Recommendations for vibratory molding of reinforced concrete products" [5]. But, as we wrote in Section 2, this system is very sensitive to changes in its parameters due to strong nonlinearity. The parameters of the exciting force and of the system (the mass of the mold with concrete mixture, the stiffness coefficients of the vibration limiters, etc.) are selected in the machine tuning process. In particular, they are selected in such a way that the operation mode of the machine is close to resonant.

First of all, it is necessary to set damping ratios ξ_1, ξ_2, ξ_0 . We can assume that the damping ratio should remain in the range $0 \leq \xi \leq 1$ [13].

The damping effect plays a key role when the excitation of oscillations occurs at a frequency close to the natural system frequency. With precise resonance, the amplitude of the oscillations will tend to infinity until damping is taken into account. The actual amplitude in resonance is determined in fact by the magnitude of such damping.

We choose the damping ratios so as to obtain firstly, steady-state T -periodic oscillatory process after transient period, and secondly, the amplitude of mold oscillation, close to the required 0.8 – 1 mm.

Then we select the parameters of stiffness for spring k_1 and for gasket k_0 . It is known that the parameters of rigidity are relevant ones. They strongly affect the oscillatory process. They also determine the natural system frequency. When selecting these parameters, we are guided by the same principles as when choosing damping coefficients.

Note 1. Elastic moduli of mold and gasket, Poisson's ratios, and gasket radius are included in the expression of contact Hertz's force. The gasket surface is flat. But we consider it as sphere of large radius in order to use the expression for contact Hertz' force.

It must be said that we cannot calculate the natural frequency of the oscillatory system. It changes its structure during movement because the mold comes off the table and then falls down on it. Its oscillatory motion is described by the equations (10). It is seen that Heaviside function $H(z)$ provides a change in structure. Separate motion of bodies and their joint motion during an impact are described by different equations. Therefore, the stiffness matrixes are different for these equations. We'll see a resonance after the formation of amplitude-frequency responses. Then we can clarify some system parameters

After many numerical experiments, we take the numerical parameters in vibro-impact system as shown in Table 2.

Table 2

Mass of table m_1 , kg	7400
Mass of mold with concrete m_2 , kg	15000
Stiffness of rubber gasket k_0 , $\text{N}\cdot\text{m}^{-1}$	$3.0\cdot 10^8$
Stiffness of spring k_1 , $\text{N}\cdot\text{m}^{-1}$	$2.6\cdot 10^7$
Thickness of gasket h , m	0.0275
Damping ratio of dashpot 1 (spring) ξ_1	0.5
Damping ratio of dashpot 0 (gasket) ξ_0	0.02
Damping ratio in concrete mixture ξ_2	0.03
Elastic modulus of mold E_2 , $\text{N}\cdot\text{m}^{-2}$	$2\cdot 10^{11}$
Elastic modulus of rubber gasket E_1 , $\text{N}\cdot\text{m}^{-2}$	$3\cdot 10^7$
Poisson's ratio of mold ν_2	0.3
Poisson's ratio of rubber gasket ν_1	0.4
Radius of gasket R , m	10
Amplitude of exciting force P , N	$2.44\cdot 10^5$
Frequency of exciting force ω , $\text{rad}\cdot\text{s}^{-1}$	157

Note 2. After integrating the equations of motion (7) and (10), we get a complete picture of the system motion, including the impact time. It should be noted that we were forced to significantly reduce the integration step during an impact. This decrease was much stronger than we did before when we considered a rigid (hard) impact between solids. We think this is due to the great softness of the rubber gasket. Its modulus of elasticity is small; therefore, its deformations during an impact may be not small.

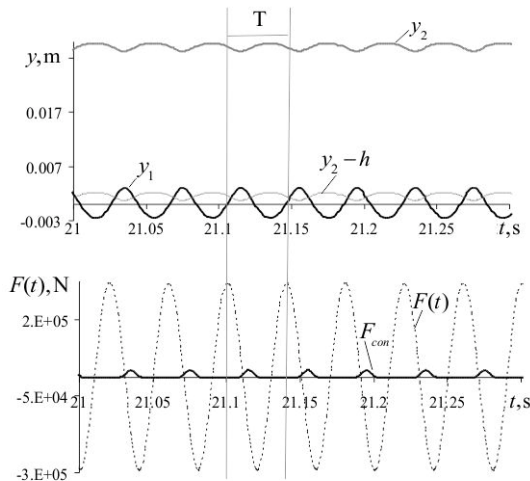


Fig. 3

We get a graph of contact forces and graphs of all other forces (Fig.5,6) acting in the system during movement. We give the graphs of both bodies' velocities and accelerations.

Time histories and contact force graph (Fig.3) show that the movement is T -periodical with one impact per cycle. We can clearly see the great penetration one body into another when we look at the curve ($y_2 - h$). The impact is very soft due the softness and suppleness of the rubber gasket and the penetration is quite large. Amplitude of mold oscillations is $A=0.76$ mm.

The phase trajectories (Fig.4) show that T -periodical movement is steady-state after passing the transient process.

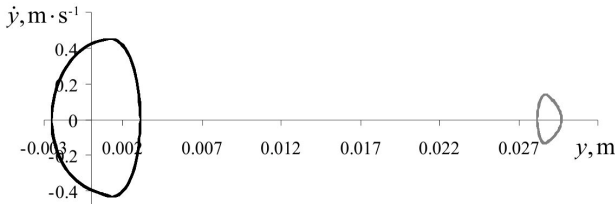


Fig. 4

The speed graph (Fig.5) shows the gradual, not instantaneous reverses in velocities during an impact because the impact is not instantaneous, its duration is long. Impact time T_{impact} is marked by vertical lines in Fig.5.

And finally, the acceleration graph (Fig. 5) gives the picture of asymmetric accelerations. We have the lower acceleration $w_L = 39.4 \text{ m} \cdot \text{s}^{-2} \approx 4 \text{ g}$ and the upper acceleration $w_U = 11.2 \text{ m} \cdot \text{s}^{-2} \approx 1.1 \text{ g}$ (g is the gravitational acceleration).

Its ratio is $\frac{w_L}{w_U} = 3.6$.

5. Conclusions

The created mathematical model of platform-vibrator with shock corresponds to the two-body 2-DOF vibro-impact system. It is strongly nonlinear non-smooth discontinuous system. It has some peculiar properties, namely: the upper body with very large mass breaks away from the lower body during vibrational motion; both bodies move separately; the upper body falls down onto the constraint that is on the rubber gasket; the impact that occurs is soft one due to the softness and flexibility of the rubber gasket. The numerical parameters for this system were chosen in such a way that: firstly, they provide the fulfillment of requirements for real machine, and secondly, they allow analyzing its dynamic behavior by nonlinear dynamics tools. The created model provides: T -periodic steady-state movement after passing the transient process; the appropriate value of mold oscillations amplitude $A= 0.76$ mm; the satisfactory

value of the asymmetry coefficient – the ratio of lower acceleration to the upper acceleration is $\frac{w_L}{w_U} = 3.6$.

Thus, we believe that the created model meets all the necessary requirements.

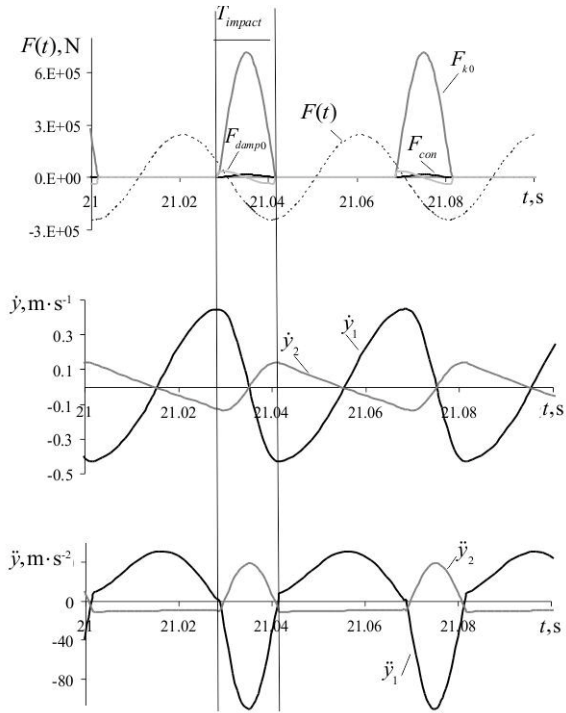


Fig. 5

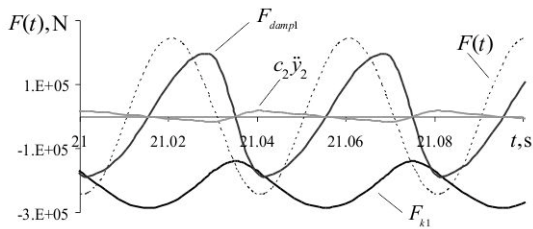


Fig. 6

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СТВОРЕННЯ МАТЕМАТИЧНОЇ МОДЕЛІ УДАРНО-ВІБРАЦІЙНОГО МАЙДАНЧИКА ДЛЯ УЩІЛЬНЕННЯ ТА ФОРМУВАННЯ БЕТОННИХ ВИРОБІВ

Вібраційні майданчики є головним обладнанням при виробництві бетонних та залізобетонних виробів. Ударно-вібраційна технологія при виробництві збірного залізобетону на низькочастотних резонансних вібро-майданчиках значно поліпшує якість виробів та ступінь їхньої заводської готовності. Ця технологія використовується для виробництва великогабаритних виробів.

В статті описується створення математичної моделі ударно-вібраційного майданчика, де реалізується режим асиметричних коливань, у якому верхнє та нижнє прискорення форми з бетоном мають різні значення. Створена математична модель відповідає двох-масовій вібро-ударній системі з двома ступнями вільності. Це сильно нелінійна негладка розривна система, яка має такі особливості: верхнє тіло дуже великої маси відривається під час коливального руху від нижнього тіла, і тоді тіла рухаються окремо; потім верхнє тіло падає на м'який обмежник; відбувається м'який удар. Моделювання м'якого удару потребує окремого обговорення. У цій статті удар моделюється нелінійною контактною силою відповідно до квазістатичного контактного закону Герца.

Числові параметри системи вибиралися таким чином, щоб по-перше, вони забезпечували виконання вимог до реальної машини, та по-друге, дозволили виконати аналіз її динамічної поведінки засобами нелінійної динаміки. Створена модель достатньо добре забезпечує

виконання низки вимог, а саме: T -періодичний усталений рух після перехідного процесу; додатне значення амплітуди коливань форми; задовільну величину коефіцієнту асиметрії, а саме відношення нижнього прискорення до верхнього.

Ключові слова: ударно-вібраційний майданчик, вібро-ударна система, форма з бетоном, верхнє та нижнє прискорення.

UDC 539.3

Bazhenov V.A., Pogorelova O.S., Postnikova T.G. Creation of mathematical model of platform-vibrator with shock, designed for concrete products compaction and molding Lyapunov exponents estimation for strongly nonlinear nonsmooth discontinuous vibroimpact system // Strength of Materials and Theory of Structures: Scientific-and-technical collected articles. – K.: KNUBA, 2020. – Issue 104. – P. 103-116.

Platform-vibrators are the main molding equipment in the production of precast concrete elements. Shock-vibration technology for the precast concrete production on low-frequency resonant platform-vibrators significantly improves the quality of the products front surfaces and the degree of their factory readiness. This technology is used to produce large elements.

We describe the creation of a mathematical model for platform-vibrator that uses shock to produce asymmetric oscillations. The values of the upper and lower accelerations of the mold with concrete have different values with shock-vibration technology.

The created mathematical model corresponds to the two-body 2-DOF vibro-impact system. It is strongly nonlinear non-smooth discontinuous system. It has some peculiar properties, namely: the upper body with very large mass breaks away from the lower body during vibrational motion; both bodies move separately; the upper body falls down onto the soft constraint; the impact that occurs is soft one due to the softness and flexibility of the constraint. The soft impact simulation requires special discussion. In this paper, we simulate a soft impact by a nonlinear contact force in accordance with the Hertz quasistatic contact law.

The numerical parameters for this system were chosen in such a way that: firstly they provide the fulfillment of requirements for real machine, and secondly they allow analyzing its dynamic behavior by nonlinear dynamics tools. The created model is well enough to fulfill a number of requirements, namely: T -periodic steady-state movement after passing the transient process; the appropriate value of mold oscillations amplitude; the satisfactory value of the asymmetry coefficient that is the ratio of lower acceleration to the upper acceleration. We believe that the created model meets all the necessary requirements.

Table 2. Fig. 6. Ref. 13

УДК 539.3

Баженів В.А., Погорелова О.С., Постнікова Т.Г. Створення математичної моделі ударно-вібраційного майданчика для ущільнення та формування бетонних виробів // Опір матеріалів і теорія споруд: наук.-тех. збірн. – К.: КНУБА, 2020. – Вип. 104. – С. 103-116. – Англ.

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Табл 2. Рис. 6. Бібліогр. 13.

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