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COMPARATIVE ANALYSIS OF METHODS FOR DETERMINING THE PARAMETERS OF SMALL-SIZED VIBRATION INSTALLATIONS

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The article discusses modern approaches to determining the main parameters of small-sized vibrating pads, which are widely used in the construction industry for compaction of concrete mixtures and the formation of products with specified physical and mechanical properties. The relevance of the study is due to the need to increase the efficiency of vibration equipment while reducing energy consumption, material consumption and dimensions of installations.

The main attention is paid to the analysis of existing methods for calculating the dynamic characteristics of vibration sites, in particular, the determination of the amplitude of vibrations, the frequency of excitation, accelerations of the working platform and the parameters of vibration exciters. Analytical, empirical and numerical approaches to modeling oscillatory processes, including the application of methods of the theory of oscillations and finite elements, are considered. It has been established that traditional analytical dependencies do not always take into account the influence of design features and real operating conditions, which can lead to deviations between the calculated and experimental data.

The paper conducts a comparative analysis of various calculation methods, identifies their advantages and limitations in terms of accuracy, labor intensity and the possibility of practical application. Particular attention is paid to the influence of the payload mass, the stiffness of elastic elements, and the excitation force parameters on the efficiency of the vibrating platform. It is shown that the most promising are combined approaches that combine analytical calculations with numerical modeling and experimental verification of results.

The results obtained can be used in the design and modernization of small-sized vibration units, as well as for optimizing their operating modes, taking into account specific technological conditions. The practical significance of the study is to improve the quality of compaction of concrete mixtures, reduce energy costs and ensure the reliability of equipment operation.

Thus, the analysis allows us to form reasonable recommendations for the choice of effective methods for calculating the parameters of small-sized vibration platforms and determines the directions of further scientific research in this area.

Keywords: vibration platform, vibration equipment, dynamic characteristics, oscillation amplitude, vibration frequency, vibration exciter, concrete mixture compaction, finite element method, mathematical modeling, parameter optimization.

Entry. Design schemes of small-sized vibrating pads, as a rule, are represented by single-mass systems (Fig. 1.) [7, 12].

Despite the well-known equation (1), its solution for the steady state (3), a number of problems arise in determining the mass, elastic and dissipative characteristics included in equation (1). Therefore, we will consider the existing methods for determining these characteristics and assess their impact.

Forced oscillations of the system (Fig. 1) under the conditions of the action of viscous resistance forces are described by a non-homogeneous differential equation of the second order [8]:

$$m\ddot{y}+b\dot{y}+cy=F_0\sin(\omega t+\varphi_0), \tag{1}$$

where F_0 is the forcing force of the imbalance, t is time; φ_0 - the initial phase of oscillations.

When calculating structures using the finite element method, a large.

The amplitude of the forcing force is determined by the well-known formula for an unbalanced vibrator:

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$$F_0 = m_0 r_0 \omega^2, \tag{2}$$

where m_0 is the unbalanced mass of the imbalance, r_0 is the eccentricity of the imbalance.

In a constant mode, that is, after a sufficiently large period of time after the start of movement, the resulting movement of the system consists only of forced oscillations, since free oscillations are attenuated [8].

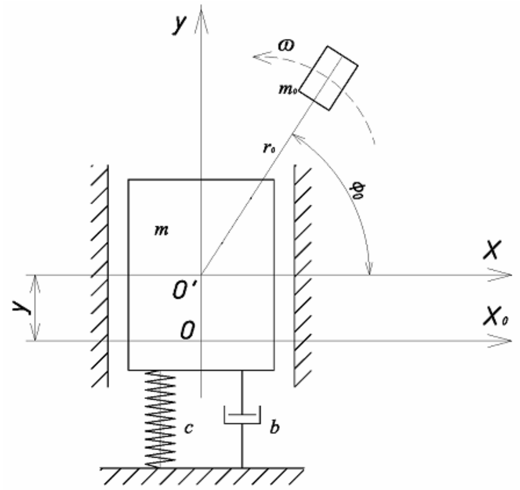

Fig. 1. Design scheme of a single-mass vibrating platform: *X* is the horizontal coordinate; *Y* is the vertical coordinate; *O, O′* is the center of the system in static and displacement, respectively; φ_0 is the phase angle; *m, c, b* – mass, coefficients of elasticity and resistance, respectively; m_0, r_0 are the mass and eccentricity of the imbalance, respectively; *y, ω* are the current amplitude of displacement and frequency of oscillations, respectively

Partial solution of equation (1):

$$y = A\sin(\omega t + \varphi_0 - \varphi),$$
$$\dot{y} = A\omega\cos(\omega t + \varphi_0 - \varphi),$$
$$\ddot{y} = -A\omega^2\sin(\omega t + \varphi_0 - \varphi). \tag{3}$$

Substituting (2) into (1) we get the following expressions to determine the amplitude of oscillations and the phase angle:

$$A = \frac{F_0}{\sqrt{(c - m\omega^2)^2 + (b\omega)^2}}, \tag{4}$$

$$\varphi = \mathrm{arctg}\frac{b\omega}{c - m\omega^2}. \tag{5}$$

The first term of equation (1) $m\ddot{y}$ reflects the action of inertial forces in a vibration system. As a rule, it consists of the mass of oscillating parts of the vibrating machine m_k , the mass of the shape m_f and the mass of the materials to be processed m_b [8, 10]:

$$m = m_k + m_f + m_b. \tag{6}$$

The most uncertain component (6) is the consideration of m_b .

In a number of works [1, 2, 8], it is recommended to take into account only a part of the mixture $m_б$, which has a significant impact on the parameters of the movement of the vibrating machine. This part of the material is called the attached mass and is calculated according to the formula [9]:

$$m_{pr} = K_{pr} \cdot m_b. \tag{7}$$

where K_{pr} is the mass addition coefficient.

The coefficient of connection for concrete mixtures is recommended to be taken within K_{pr} = 0.15 ... 0.4. Moreover, the lower limit corresponds to the formation of products from a plastic mixture with a little saturated reinforcement, and the upper limit - from rigid mixtures with a high saturation with reinforcement [9]. In the work [10] it is noted that the composition of the mixture significantly affects the coefficient of connection much less than the density of reinforcement. Therefore, for lightly reinforced products, K_{pr} = 0.2 ... 0.25, for medium-reinforced K_{pr} = 0.25 ... 0.3 and for densely reinforced K_{pr} = 0.3 ... 0,4.

The concept of attached (reduced) mass is introduced to adjust dynamic calculations based on the representation of a discrete model of the mixture [8], which simplifies the calculation, but does not reveal the physical essence of the process that occurs and leads to its incorrect interpretation. It is noted: "during vibration, a larger number of particles of the mixture are as if in a suspended state and do not affect the shape with their gravity".

Due to different values, there is a significant difference between the calculated and actual values of the amplitudes of oscillations. Thus, the deviation of the calculated values from the experimental ones is 120 – 190% at a frequency of oscillations of 1500 counts/min., 17 – 185% – at a frequency of 3000 counts/min. and 68 – 80% – at a frequency of 6000 counts/min. [2, 10]. Such a large difference between the calculated and actual values of the amplitudes of oscillations can be explained by insufficient consideration of the dynamic properties of the mixture.

One of the main reasons for the difference in the actual values of the amplitudes of oscillations, according to the author of the work [2, 10], is that equation (1) includes the entire mass of the concrete mixture. He points out: "In fact, the movement of particles of the concrete mixture during vibration is complex and cannot be described by the equation of harmonic oscillations. The practical result of this phenomenon is the fact that the value of the "reduced mass" involved in the vibrations together with the vibration pad differs from the value of the mass of the forming mixture.

In the paper [9] it is noted that some resistance coefficients turn out to be imaginary, and some are real. Imaginary coefficients mean that the resistance of the medium does not increase the oscillating mass, but decreases it, i.e. that the resistance is negative, which contradicts the existing laws of mechanics.

The second term of equation (1) $b\dot{y}$ expresses the effect of dissipative forces on the vibration machine in the form of a viscous resistance force [9]. Such a resistance force gives a differential function in time, and does not complicate the solution of the differential equation. According to some studies [10, 11], the hypothesis of viscous resistance is not confirmed. In this case, other models are used [8, 9] or the real dissipative forces are replaced by an equivalent viscous resistance. This resistance is determined under the condition that that the dissipation of energy in one cycle of oscillations is equal to the energy of the actual drag force [11].

This approach simplifies mathematical calculations, but does not reflect the physical essence of the processes occurring under the influence of a complex resistance force that occur during the operation of vibration machines.

The elastic forces included in equation (1) are usually taken into account by the linear stiffness of the elastic suspension of the vibrating machine, which in some cases requires clarification [9, 10].

The argument of the trigonometric function of the forcing force , included in the right side of equation (1.4), indicates the circulation of the force in time and has an additional constant shift along the angle of rotation of the φ, which is called the phase angle [9-11], the essence of which is interpreted differently in the literature [7-11].

The need to introduce this angle into the argument of the function is due to finding the exact mathematical solution of equation (1).

The numerical values of the phase angle vary quite widely [9, 11].

Thus, in the work [9] it is noted that the angle of shift of the phase of the shape displacement from the phase of the forcing force ranges from 165°≥φ≥155°, according to other data [9] - 18... 20°, [11] - 30… 40°.

Thus, the numerical values of the phase angle φ lie within a fairly wide range of 18 ... 165°.

Methods for measuring the phase angle during the operation of technological vibration machines are not given in the literature, with the exception of works [9]. This is probably due to the complexity of measuring this parameter of oscillations, namely: the lack of generally accepted methods and schemes for measuring the phase angle.

Equation (1) gives solutions in which kinematic relationships are established between the quantities that characterize the oscillations of the system, such as the amplitude, frequency and phase of oscillations. The solution does not give a direct answer to the questions of the force interaction between the vibrating machine and the processed medium and does not reflect the energy costs, which consist of frictional resistances in the machine drive and the total resistances of system vibrations [9]. The first of them do not affect the behavior of the oscillating system and determine approximately a constant part of the total energy consumption. The second ones change depending on the displacement of the system elements and the parameters of the compacted medium. They have a significant effect on the amplitude and energy of oscillations.

A number of works are devoted to the assessment of concrete compaction energy [1, 3, 4, 5, 6, 9, 11, 13].

In the work [10], the value of this energy is determined by the formula:

$$P = b_{ud} S A^2 \omega^2, \tag{8}$$

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where b_{ud} - specific coefficient of resistance, attributed to the unit of active area of the vibration compactor housing; S - active area of the vibration compactor housing; A, ω – respectively, the amplitude and frequency of oscillations.

Power in work [8] is found by the formula:

$$P=\alpha_M m'\bar{N}, \tag{9}$$

where α_M - some decreasing imperial coefficient; m' - Concrete weight is given; $\bar{N}$ – specific power of oscillations.

In the work [1], power is found based on the hypothesis of viscous resistance:

$$P=C\frac{A^2\omega^2}{2}, \tag{10}$$

where C is the coefficient of resistance of the medium to vibrations.

In the paper [10, 11], the power of the wave energy flow transferred to the concrete medium during vibration forming is theoretically determined:

$$P=2\pi^2\rho SA^2 f^2, \tag{11}$$

where ρ is the density of the medium; f is the frequency of oscillations in Hz.

From the formula it can be seen that the power is proportional to the cross-sectional area. From this, the author concludes that it is necessary to transmit concrete vibrations through the largest possible area of the working body.

In the work [9], the specific power for compaction of the mixture is found according to its initial state according to the formula:

$$P=W/t, \tag{12}$$

where W is the energy determined by the area of the stress-strain curve.

Useful power in robots [1, 8, 11] is determined by the maximum forcing force and phase angle .

$$P=\frac{1}{2}F_0 A\omega\sin\varphi. \tag{13}$$

Analyzing the results of the work of various authors, we can conclude that they are all based on the presentation of the "concrete mix-environment" system in the form of one or another simplified model.

More fully dynamic properties of the concrete mixture are expressed by a system with distributed parameters. This circumstance is noted in the works [8, 11].

In the work [10], a formula is obtained for determining the amplitude of vibrations of the vibration platform when modeling a concrete mixture as an elastic medium:

$$A=\frac{F_0}{c_0-m_0\omega^2-m_b\omega^2\dfrac{\mathrm{tg}(\omega h/c)}{(\omega h/c)}}, \tag{14}$$

where h is the height of the mixture column; c is the speed of wave propagation in the medium; m_b is the mass of concrete.

Here is the expression in the denominator:

$$m_{pr}=m\frac{\mathrm{tg}(\omega h/c)}{(\omega h/c)} \tag{15}$$

determines the elastic and inertial properties of the concrete mixture, which affect the amplitude of vibrations.

However, the accepted simplification excludes energy dissipation and leads to significant discrepancies between the calculated and actual values of the amplitude of oscillations in the zones close to resonance, and this factor limits the use of this formula.

Obviously, at $(\omega h/c)=(\pi/2)+\kappa\pi$ values, where k = 0.1, 2, ... n gives a value $m_{pr}=\infty$, and the amplitude of oscillations is directed to 0.

The model with distributed parameters is used in the work [8] when describing the process of compaction of a concrete mixture by the vibrating piston method.

In these calculations, the effect of concrete on the oscillations of the system is taken into account on the basis of the viscous friction hypothesis.

The hypothesis of viscous friction meets with objections in a number of works [5-11], in which it is proved that it is impossible to use this hypothesis for vibration compaction of concrete.

In the work [1, 10], according to the hypothesis of the dependence of resistance forces on the amplitude of stress, an expression was obtained to determine the amplitude of piston oscillations:

$$A=\frac{F_0\left[1+\left(\beta(c/\omega)\right)^2\right]}{\omega\rho cF\sqrt{\left(\left((c/\omega)-\omega M\right)\left[1+\beta^2(c/\omega)^2\right]-a_1\right)^2+\left(z_p\dfrac{1+\left(\beta(c/\omega)\right)^2}{\rho cS}+a_2\right)^2}}, \tag{16}$$

where

$$a_1=\frac{\beta(c/\omega)\ \mathrm{sh}2\pi d_0+\sin 2\pi\beta_0}{\mathrm{ch}2\pi d_0+\cos 2\pi\beta_0};\ a_2=\frac{\mathrm{sh}2\pi d_0\cdot\beta(c/\omega)\sin 2\pi d_0}{\mathrm{ch}2\pi d_0+\cos 2\pi\beta_0},$$

ρ is the density of the mixture; c is the speed of wave propagation in the medium; β is the attenuation coefficient; S is the area of support of compacted concrete on the vibration piston; M is the mass of the piston; Z_p is the resistance of the vibration piston to vibrations.

To determine the amplitude of vibrations of the vibration pad in the work [7]:

$$A=\frac{F_0}{\sqrt{\dfrac{\left[\left(k_0-M_0\omega^2\right)\left(1+(M_3/m)a_2\right)-M_3\omega^2\left(1+(m/M_3)b_1\right)\right]^2+\left[\left(k_0-M_0\omega^2\right)(M_3/m)b_2-m\omega^2a\right]^2}{\left(1+(M_3/m)a_2\right)^2+\left((M_3/m)b_2\right)^2}}}, \tag{17}$$

where a_1, a_2, b_1, b_2 are the coefficients that characterize the resistance forces of the concrete mixture.

As follows from (16), the effect of vibrating concrete on the amplitude of oscillations is determined by the numerical values of the parameters ρ, β, c and S.

Undoubtedly, the analytical description of the process based on the model with distributed parameters most fully reflects the physical content of the behavior of the system during the vibration formation of concrete. However, the practical use of the above formulas encounters serious difficulties due to different values of attenuation coefficients (Fig. 2) and significant differences in the values of the propagation rates of oscillations, without numerical values of which calculations cannot be performed (Table 1).

Table 1

Numerical values of the propagation rate of vibrations and moduli of elasticity of the compacted concrete mixture

Speed of propagation of elastic waves v, m/s	Modulus of elasticity, $E\cdot10^4$ N/m2	Frequency f, Hz	Source of information
10… 21	–	25	[8]
31… 37	–	50	
168… 178	–	200	
20… 25	94,0… 150,0	25	
35… 55	287,9… 710,8	50	
80… 120	1504,0… 3384,0	100	[10]
160… 180	6016,0… 7614,0	200	
35… 120	–	25… 50	
48… 57	–	50	[9]
45… 48	423,3… 495	50	[11]
–	245… 882	25… 100	[7]

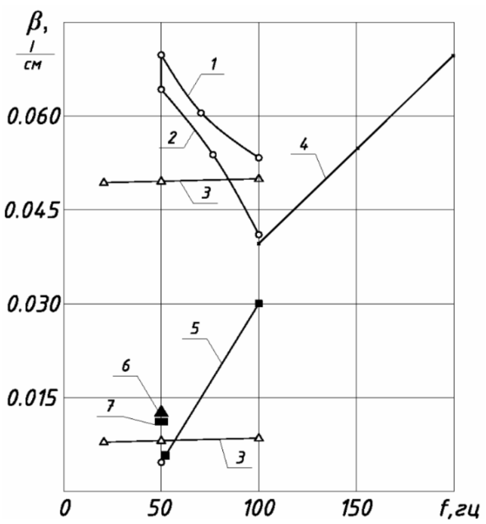

Fig. 2. Numerical values of the attenuation coefficient for different frequencies of oscillation according to the authors' work [8]

Conclusion. All the considered methods refer to the stable mode of motion, however, the question arises of assessing the movement of vibrating machines in the transient (accelerating) mode of operation, since such a mode takes place for a single-mass vibration system.

Oscillations under the influence of energy sources with limited power, systems change their properties, that is, the properties of oscillating systems become dependent on the properties of the energy source.

There is a well-known work [8] on the study of forced oscillations of a linear system with one degree of freedom during the passage of a resonance zone, where a formula is obtained for determining the frequency at which the maximum amplitudes of oscillations are reached based on the hypothesis of viscous friction. The author notes the effect of beating, which is the result of the imposition of forced oscillations with the frequency of the forcing force and attenuation of free oscillations that occur during the passage of resonance.

A number of works [6-13] have been carried out, in which this problem is investigated, a well-grounded review of which is given in the monograph [14, 15]. Since in this work the main goal is to find stabilization methods to describe the behavior of vibrating machines in the transient mode of operation, Lagrange equations of the second kind are used, which are based on the energy description of the movement of the system, and well reflect the energy exchange between the rotating and oscillating parts of the vibrating machine, which allows you to study the processes of interaction of a vibrating machine with an inertial vibration drive and solve the research problem.

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ПОРІВНЯЛЬНИЙ АНАЛІЗ СПОСОБІВ ВИЗНАЧЕННЯ ПАРАМЕТРІВ МАЛОГАБАРИТНИХ ВІБРАЦІЙНИХ УСТАНОВОК

У статті розглянуто сучасні підходи до визначення основних параметрів малогабаритних вібромайданчиків, що широко застосовуються у будівельній галузі для ущільнення бетонних сумішей та формування виробів із заданими фізико-механічними властивостями. Актуальність дослідження обумовлена необхідністю підвищення ефективності роботи вібраційного обладнання при одночасному зменшенні енергоспоживання, матеріаломісткості та габаритів установок.

Основну увагу приділено аналізу існуючих методів розрахунку динамічних характеристик вібромайданчиків, зокрема визначенню амплітуди коливань, частоти збудження, прискорень робочої платформи та параметрів вібробуджувачів. Розглянуто аналітичні, емпіричні та чисельні підходи до моделювання коливальних процесів, включаючи застосування методів теорії коливань і скінченних елементів. Встановлено, що традиційні аналітичні залежності не завжди враховують вплив конструктивних особливостей і реальних умов експлуатації, що може призводити до відхилень між розрахунковими та експериментальними даними.

У роботі проведено порівняльний аналіз різних методик розрахунку, визначено їх переваги та обмеження щодо точності, трудомісткості та можливості практичного застосування. Особливу увагу приділено впливу маси завантаження, жорсткості пружних елементів та параметрів збуджувальної сили на ефективність роботи вібромайданчика. Показано, що найбільш перспективними є комбіновані підходи, які поєднують аналітичні розрахунки з чисельним моделюванням та експериментальною верифікацією результатів.

Отримані результати можуть бути використані при проєктуванні та модернізації малогабаритних вібраційних установок, а також для оптимізації їх режимів роботи з урахуванням конкретних технологічних умов. Практичне значення дослідження полягає у підвищенні якості ущільнення бетонних сумішей, зниженні витрат енергії та забезпеченні надійності роботи обладнання.

Таким чином, проведений аналіз дозволяє сформувати обґрунтовані рекомендації щодо вибору ефективних методів розрахунку параметрів малогабаритних вібромайданчиків і визначає напрямки подальших наукових досліджень у цій галузі.

Ключові слова: вібромайданчик, вібраційне обладнання, динамічні характеристики, амплітуда коливань, частота коливань, віброзбуджувач, ущільнення бетонної суміші, метод скінченних елементів, математичне моделювання, оптимізація параметрів.

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COMPARATIVE ANALYSIS OF METHODS FOR DETERMINING THE PARAMETERS OF SMALL-SIZED VIBRATION INSTALLATIONS

The article discusses modern approaches to determining the main parameters of small-sized vibrating pads, which are widely used in the construction industry for compaction of concrete mixtures and the formation of products with specified physical and mechanical properties. The relevance of the study is due to the need to increase the efficiency of vibration equipment while reducing energy consumption, material consumption and dimensions of installations.

The main attention is paid to the analysis of existing methods for calculating the dynamic characteristics of vibration sites, in particular, the determination of the amplitude of vibrations, the frequency of excitation, accelerations of the working platform and the parameters of vibration exciters. Analytical, empirical and numerical approaches to modeling oscillatory processes, including the application of methods of the theory of oscillations and finite elements, are considered. It has been established that traditional analytical dependencies do not always take into account the influence of design features and real operating conditions, which can lead to deviations between the calculated and experimental data.

The paper conducts a comparative analysis of various calculation methods, identifies their advantages and limitations in terms of accuracy, labor intensity and the possibility of practical application. Particular attention is paid to the influence of the loading mass, the rigidity of elastic elements and the parameters of the excitatory force on the efficiency of the vibrating platform. It is shown that the most promising are combined approaches that combine analytical calculations with numerical modeling and experimental verification of results.

The results obtained can be used in the design and modernization of small-sized vibration units, as well as for optimizing their operating modes, taking into account specific technological conditions. The practical significance of the study is to improve the quality of compaction of concrete mixtures, reduce energy costs and ensure the reliability of equipment operation.

Thus, the analysis allows us to form reasonable recommendations for the choice of effective methods for calculating the parameters of small-sized vibration platforms and determines the directions of further scientific research in this area.

Keywords: vibration platform, vibration equipment, dynamic characteristics, oscillation amplitude, vibration frequency, vibration exciter, concrete mixture compaction, finite element method, mathematical modeling, parameter optimization.

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Максим'юк Ю.В., Кузьмінець М.П., Онищенко А.М., Мартинюк І.Ю., Максим'юк О.В. **Порівняльний аналіз способів визначення параметрів малогабаритних вібраційних установок** // Опір матеріалів і теорія споруд: наук.-тех. збірн. – Київ: КНУБА, 2026. – Вип. 116. – С. 491-498.

У даній роботі розглянуто сучасні методи розрахунку параметрів малогабаритних вібромайданчиків і показано, що традиційні аналітичні підходи мають обмежену точність через неврахування реальних умов роботи. Найефективнішими є комбіновані методи, що поєднують аналітичні, чисельні та експериментальні підходи. Їх застосування дозволяє підвищити ефективність, надійність і енергоощадність обладнання, а також оптимізувати процес ущільнення бетонних сумішей.

Табл. 1. Іл.2. Бібліогр. 15назв.

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Maksymiuk Yu.V., Kuzminets M.P., Onishchenko A.M., Martinyuk I.Yu., Maksymiuk O.V. **Comparative analysis of methods for determining the parameters of small-sized vibration installations** // Strength of Materials and Theory of Structures: Scientificand-technical collected articles. – Kyiv: KNUBA, 2026. – Issue. 116. – P. 491-498.

This paper examines modern methods for calculating the parameters of small-sized vibration platforms and shows that traditional analytical approaches have limited accuracy due to the neglect of real operating conditions. The most effective are combined methods that integrate analytical, numerical, and experimental approaches. Their application makes it possible to improve the efficiency, reliability, and energy efficiency of the equipment, as well as to optimize the process of concrete mixture compaction.

Table. 1. Fig. 2. Refs. 15 titles.

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