Istraživanje efikasnijih vibracionih mašina s obzirom na naponsko-deformaciono stanje tehnološkog okruženja

Ivan Nazarenko^{1*}, Oleg Dedov¹, Anatoly Svidersky¹, Nicolay Ruchinsky¹
¹ Državni univerzitet za građevinarstvo i arhitekturu u Kijevu, Kijev (Ukrajina)

U radu je razmatrana analitička metoda za određivanje uticaja materijala koji se obrađuju na dinamiku radnih delova vibracionih mašina. Predstavljeni su rezultati istraživanja koji omogućavaju identifikovanje zona efikasnog dejstva parametara i karakteristika sistema „mašina - okolina“ a kako bi se osigurao visok kvalitet procesa zbijanja, koji se zasniva na ideji svrsishodne upotrebe elastičnih karakteristika tipičnih vibracionih sistema.

Iznesena je naučna ideja koja se sastoji u činjenici da tokom proučavanja određenog procesa treba utvrditi matematički model vibracionog sistema „mašina-okruženje“ na osnovu uzimanja u obzir unutrašnje strukture podsistema kao jedinstvenog sistema, uprkos njihovoj različitoj fizičkoj prirodi i strukturi.

U radu je data teorijska potvrda mogućnosti stvaranja vibracionih mašina za rasute kalupe sa visoko efikasnim parametrima potrošnje energije i materijala za različite uslove formiranja betonskih i armiranobetonskih proizvoda. Razlog je stvaranje i racionalno korišćenje zakonitosti promena unutrašnjih (elastično-inercijalnih i disipativnih) karakteristika sistema „mašina-okruženje“ u režimu koji je približan režimu njegovih slobodnih oscilacija..

Ključne reči: Vibracioni sistem, Vibrator, Frekvencija vibracija, Amplituda vibracija, Dinamičko opterećenje, Efikasnost vibracione mašine

1. UVOD

Vibracione mašine se široko koriste u građevinskoj industriji za dobijanje kompaktne strukture betonskih smeša i tla. Najefikasnije od njih su vibro-udarne mašine. To je zbog činjenice da tokom udara u radnu smešu nastaju velika naprezanja (usled velikih ubrzanja), koja uzrokuju deformacije i, shodno tome, povećavaju gustinu smeše. Međutim, stepen efikasnosti ovih mašina u velikoj meri je određen veličinom troškova energije za savladavanje neophodnih radnih opterećenja. Zbog toga je rešavanje problema smanjenja potrošnje energije vibracionih mašina uz očuvanje njihove visoke tehnološke efikasnosti važan naučni i praktični zadatak.

Vibracione mašine se široko koriste u raznim granama ekonomije a za izvođenje širokog spektra tehnoloških procesa. Uobičajena karakteristika ove klase mašina je vibracioni efekat na materijal koji se obrađuje, usled čega materijal menja svoja svojstva, obezbeđujući tako odgovarajući tehnološki proces.

Veliki broj istraživanja posvećeno je proučavanju sistema pod uticajem vibracija, tj. analizi i sintezi vibracija mehaničkih sistema.

Što se tiče radova posvećenih proučavanju vibracija mašina za betonske smeše, moguće je primetiti da se u njima na osnovu određenih pretpostavki određuju parametri kretanja takvih sistema. Najopštija je pretpostavka sistema „vibracione mašine – radno okruženje“ u obliku diskretnog modela. Radovi [1,2] razjašnjavaju ovaj model i predlažu metod prelaska sa diskretnog beskonačnog sistema (diskretni - mašina, kontinuum - okruženje) do čisto diskretnog uzimajući u obzir talasne pojave u betonskoj smeši.

Ovaj pristup omogućava značajno pojednostavljenje šeme proračuna. Princip prelaska najrealnije šeme u izračunatu (diskretnu) dat je u [1] (slika 1).

Jedan od glavnih kriterijuma takvih sistema je energija udara u periodičnom kretanju, koja određuje efikasnost režima. Svrha rada je proučavanje i uspostavljanje efikasnih zona stabilnosti i procena energetskih karakteristika sistema „vibraciona mašina – radno okruženje“ za razvoj novih progresivnih mašina, uzimajući u obzir naponsko-deformaciono stanje radne sredine.

Razmatranjem uslova postojanja stabilnosti režima sistema sa odvajanjem od elastičnog ograničenja oscilacija (slika 1, c) na granicama linearnih preseka u skladu sa radom [2]:

$$\frac{\varepsilon^2}{|1-\varepsilon^2|} \cdot \frac{\sin \varphi + \sin(\tau_x + \varphi)}{2} \leq q \leq \frac{\varepsilon^2}{1-\varepsilon^2}; \quad (1)$$

gde je

$$\varepsilon^2 = \frac{c}{m\omega^2}; \varphi = \frac{\pi + \tau_x}{2}; \left(\varphi = \frac{3\pi + \tau_x}{2} \right)$$

*Kontakt adresa autora: Povitroflotsky Avenue 31, Kyiv (Ukrajina), email: i_nazar@i.ua

* Članak izložen na skupu u vidu usmenog saopštenja, HM2017, Zlatibor

Slika 1: Proračunski modeli vibracionih:
a – diskretno-kontinualni, b – diskretni (dve mase), c – diskretni (jedna masa)

U izrazu (1) usvojeni su sledeći simboli: c - krutost ograničivača; $q = \frac{Q}{F_0}$ - odnos težine vibracione mašine ($Q = mg$) i amplituda sile poremećaja F_0 ; ω - frekvencija sile poremećaja; τ_x - trajanje kretanja vibratora u dodiru sa elastičnim graničnikom oscilacija: $\tau_x = \frac{\pi}{\varepsilon}$; φ - fazni ugao koji u proračunima pretpostavlja se da daje pozitivnu vrednost na granicama $c = \infty$ ($\varepsilon = \infty, \tau_x = 0$), pa u tom slučaju (1) daje vrednost:

$$q = 1 \quad (2)$$

Iz (2) sledi da se kombinuju gornja i donja granica regiona stabilnih režima. Međutim, stabilnost periodičnih režima pod apsolutno krutim ograničenjima oscilacija određuje koeficijent:

$$1 \leq q \leq \infty \quad (3)$$

Kada se upoređuju odnosi (2) i (3), sledi da postojanje stabilnih periodičnih načina kretanja sistema sa elastičnim ograničenjima (sa krutošću) pri graničnoj vrednosti ne prelazi u stanje stabilnosti pri kretanju odgovarajućih sistema uz pružanje apsolutno strogih ograničenja. Da bi se eliminisao ovaj nesklad, razmatranje interakcije udara je zamena elastičnosti i rasipanje ograničenja udarnim parom, uzimajući u obzir dužinu udara. Takav pristup omogućava uzimanje u obzir interakcije udara u vibracionom sistemu pomoću pulsne teoreme i intervala smirivanja brzine udara.

$$q_{onm} = \frac{1+R}{1-R} \cdot \frac{\sqrt{2(1+\cos\tau)}}{(2\pi \cdot i - \tau_x)^2} \left(2\pi \cdot i - \tau_x + 2tg \frac{\tau_x}{2} \right) \quad (4)$$

Maksimalna udarna brzina biće definisana sledećom zavisnošću:

$$\dot{y}_{0 \max} = q_{onm} \frac{2\pi \cdot i - \tau_x}{1+R} \quad (5)$$

Razmotriće se kretanje vibratora u intervalu $0 \leq \tau \leq \tau_x$.

U bez dimenzionim veličinama imamo:

$$\ddot{y}_1 + 2n\dot{y}_1 + \varepsilon^2 y_1 = \cos(\tau + \varphi) + mg \quad (6)$$

$$\text{gde je } \varepsilon^2 = \frac{c_0}{m\omega^2}; 2n = \frac{n'}{m\omega}, n' - \text{nivo disipacije.}$$

$$y_1(0) = y(2\pi \cdot i) = 0; \dot{y}_1(0) = \dot{y}(2\pi \cdot i) = \dot{y}_0 \quad (7)$$

$$\begin{aligned} y_1 &= \dot{y}_0 a_1 + b_1 \sin \varphi + c_1 \cos \varphi + d_1 q, \\ \dot{y}_1 &= \dot{y}_0 \dot{a}_1 + \dot{b}_1 \sin \varphi + \dot{c}_1 \cos \varphi + \dot{d}_1 q. \end{aligned} \quad (8)$$

$$a_1 = e^{-n\tau} \varepsilon_1^{-1} \sin \varepsilon_1 \tau; d_1 = (1 - \dot{a}_1 - na_1) \varepsilon^{-2}; \quad (9)$$

$$\begin{aligned} b_1 &= [(a_1 - \sin \tau) \cos \psi + (\cos \tau - d_1 - na_1) \sin \psi] \cdot \left[(1 - \varepsilon^2)^2 + 4n^2 \right]^{-\frac{1}{2}}; \\ c_1 &= [(\cos \tau - d_1 - na_1) \cos \psi - (a_1 - \sin \tau) \sin \psi] \cdot \left[(\varepsilon^2 - 1)^2 + 4n^2 \right]^{-\frac{1}{2}}; \\ \varepsilon_1 &= \sqrt{\varepsilon^2 - n^2}; tg \psi = \frac{2n}{\varepsilon^2 - 1}. \end{aligned}$$

$$y_1(\tau_x) = y(\tau_x) = 0; \dot{y}_1(\tau_x) = \dot{y}(\tau_x) = -R\dot{y}_0 \quad (10)$$

Zamenom uslova (10) u jednačini (8), kao i predstavljanjem $\sin \varphi$ i $\cos \varphi$ preko q_{onm} i $\dot{y}_{0 \max}$ u skladu sa formulama (8) i (10), mogu se dobiti zavisnosti za određivanje R i τ_x :

$$R = \frac{\delta \left(\dot{c}_1 + \dot{b}_1 ctg \frac{\tau_x}{2} \right) + 2\lambda_1 (\dot{d}_1 + \delta \dot{a}_1)}{\delta \left(\dot{c}_1 + \dot{b}_1 ctg \frac{\tau_x}{2} \right) - 2\lambda_1 (\delta + \dot{d}_1)}, \quad (11)$$

$$2\lambda_1 \left[\dot{d}_1(d_1 - 1) - a_1(\delta + \dot{d}_1) \right] + \left(c_1 + b_1 \operatorname{ctg} \frac{\tau_x}{2} \right)$$

$$\left[2\dot{d}_1 + \delta(1 + d_1) \right] - \left(\dot{c}_1 + \dot{b}_1 \operatorname{ctg} \frac{\tau_x}{2} \right) (2d_1 + \delta a_1) = 0$$

gde je $\lambda_I = \frac{2}{\delta} + \operatorname{ctg} \frac{\tau_x}{2}$; $\delta = 2\pi \cdot i - \tau_x$.

Saglasno[1]:

$$\tau_x = \frac{\pi}{\sqrt{\varepsilon^2 - n^2}}; R = e^{-n\tau_x}. \quad (12)$$

Sada je neophodno odrediti uslove postojanja i stabilnosti periodičnih režima. Obim stvarnih vrednosti ograničen je odnosom:

$$q \leq \frac{\lambda \sqrt{\varepsilon_I^2 + \varepsilon_2^2}}{f_I}. \quad (13)$$

Za stvarnu pojavu periodičnih režima potrebno je obezbediti određena ograničenja sistemskih parametara i poremećaja određenih uslovima stabilnosti ovih režima. Za sistem (slika 1, c), granica regiona stabilnosti periodičnih režima određena je jednačinom:

$$\pm \dot{y}_0(1 \pm R)^2 + (2\pi \cdot i - \tau_x)[q(1 + R) + \cos(\tau_x + \varphi) + R \cos \varphi] = 0$$

Analiza rezultata ukazuje na postojanje nekoliko zona stabilnosti (slika 2), što je važno za određivanje parametara uticaja vibracija na sistem, a za koje je poznato da su svedeni na dve najvažnije [2]:

$$\varepsilon = \sqrt{\frac{c}{m\omega^2}} \quad q = \frac{mg}{F_0} \quad (14)$$

Slika 2: Mapa stabilnosti vibro-udarnih mašina

Za proučavanje intenziteta dinamičkog uticaja u radu koristi se metoda dinamičke petlje histereze. Dobijeni su izrazi za područja petlje histereze sa različitim zakonima opterećenja. U nesimetričnom zakonu u stacionarnom režimu imamo:

$$\Delta W = \frac{\sigma_{cm}^2 \gamma \pi}{2E_0(1 + \gamma^2)} \left[1 + \frac{(1 - K_\tau)}{K_\tau^2} \right], \quad (15)$$

Date su zavisnosti (15) za stacionarni (uspostavljeni) način kretanja, odnosno energija ΔW koja ide na zgušnjavanje jedinične zapremine smeše je konstantna. Da bi se procenila energija potrebna za zbijanje smeše od početne vrednosti gustine do potrebne, izraz (15) je predstavljen kao:

$$E_n = \frac{t_n}{T} \Delta W_0 v \frac{K_s - 1}{K_s \ln K_s} \quad (16)$$

gde su t_n - dužina tehnološkog procesa; ΔW_0 - površina petlje histereze na početku sabijanja smeše; K_s - koeficijent promena u površini petlje histereze u procesu oscilacija.

Na osnovu izraza (15) i (16) formula za specifičnu snagu i koeficijent efikasnosti, koji ima najveću vrednost $K_\tau = 1/\sqrt{2}$. Opšti kriterijum za procenu efikasnosti vibracionih efekata predstavljen je, koji uz kontaktno naprezanje i brzinu deformacije σ , uzima u obzir nagib profila talasa opterećenja:

$$\frac{\sigma_{cm} \cdot v_{\varepsilon_1} \cdot \Gamma\left(\frac{n_1+1}{2}\right)}{\Gamma\left(\frac{n_1+1}{2}\right)} = \frac{\sigma_{cm} \cdot v_{\varepsilon_2} \cdot \Gamma\left(\frac{n_2+1}{2}\right)}{\Gamma\left(\frac{n_2+1}{2}\right)} \quad (17)$$

gde su $\Gamma(x)$ - gamma funkcija X ; v_{ε_1} - nivo deformacije;
 n - koeficijent nagiba prednjeg dela.

Sistem u kojem će ocena (17) biti viša biće efikasniji. Prema rezultatima istraživanja, razvijeni su principi stvaranja novih progresivnih vibracionih mašina za građevinsku industriju u formiranju više šupljih ploča i temeljnih blokova.

2. ZAKLJUČAK

1. Predložena je teorija o interakciji između radnih delova vibracionih mašina i medija koji se obrađuju, što na osnovu procene naponskog stanja medija u kontaktnoj zoni omogućava utvrđivanje efekta medijum koji se obrađuje na dinamiku radnih delova mašine.

2. Predloženi su principi stvaranja zapreminskih mašina za oblikovanje sa visoko efikasnim parametrima potrošnje energije i materijala.

3. Pronađene su efikasne zone parametara i karakteristika sistema mašina-okolina kako bi se osigurao visok kvalitet postupka sabijanja, koji se zasniva na ideji o svrsishodnoj

upotrebi elastičnih karakteristika opšteg sistema vibracija, što pod svim istim uslovima rezultira i smanjenjem troškova energije za neke mašine do 50%.

4. Analitičke zavisnosti su dobijene za procenu energije interakcije radnog organa i materijala na osnovu upotrebe metode dinamičkih petlji histereze.

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Research of Energy-Saving Vibration Machines with Account of the Stress-Strain State of Technological Environment

Ivan Nazarenko^{1*}, Oleg Dedov¹, Anatoly Svidersky¹, Nicolay Ruchinsky¹

¹ Kyiv National University of Construction and Architecture, Kiev

An analytical method for determining the influence of a process medium on the dynamics of the working bodies of vibrating machines is considered. The results of the research are presented, which made it possible to identify the zones of effective action of parameters and characteristics of the "machine - environment" system to ensure the high quality of the compaction process, which is based on the idea of purposeful use of the elastic characteristics of a common vibration system.

The scientific idea is put forward, which consists in the fact that during the study of a particular process the mathematical model of the vibration system "machine-environment" should be determined on the basis of taking into account the internal structure of these subsystems as a single one, despite their different physical nature and structure.

The theoretical substantiation of the possibility of creating bulk molding Vibration Machines with highly effective parameters of energy intensity and material consumption for various conditions for the formation of concrete and reinforced concrete products is given in the paper. This is due to the creation and rational use of the regularities of changes in the internal (elastic-inertial and dissipative) properties of the "machine-environment" system in the regime, which is approximate to the mode of its free oscillations.

Keywords: Vibration system, Vibrator, Vibration frequency, Vibration amplitude, Dynamic loading, Efficiency of the vibrating machine

1. INTRODUCTION

Vibrating machines are widely used in the construction industry to seal concrete mixtures and soil. The most effective of these are vibro-impact machines. This is due to the fact that during the impact of the impact in the working mixture there are large compressive stress (due to large accelerations), which cause deformation and, consequently, increase the density of the mixture. However, the degree of efficiency of these machines is largely determined by the magnitude of energy costs for the implementation of necessary workloads. Therefore, solving the problem of reducing energy consumption of vibration damping machines while preserving their high technological efficiency is an important scientific and practical task.

Vibrating machines are widely used in various branches of the national economy to perform a significant range of technological processes. A common characteristic feature of this class of machines is the vibrational effect on the material being processed, as a result of which the material changes its properties, thus ensuring the corresponding technological process.

Many studies devoted to the study of the vibration damaging systems are presented, which are the analysis and synthesis of mechanical vibration damaging systems. Concerning works devoted to the study of vibration damping machines for sealing concrete mixtures, it is possible to note that in these works the parameters of motion of such systems are determined, based on certain assumptions. The most general is the assumption of a "vibrating machine - processing environment" system in the form of a discrete model.

The work of [1, 2] clarifies this model and proposes a method of transition from discrete continuum systems (discrete - machine, continuum - environment) to purely discrete ones taking into account wave phenomena in a concrete mixture. This approach makes it possible to significantly simplify the calculation scheme. The principle of transition of the most realistic scheme to the calculated (discrete) is given in [1] (Fig. 1).

One of the main criteria of such systems is the impact energy in periodic motion, which determines the effectiveness of the regime.

The purpose of the article is to study and establish effective zones of stability and assessment of the energy characteristics of the "vibrating machine - processing environment" system for the development of new progressive machines, taking into account the stress-strain state of the environment.

Consider the existence condition of the stability of modes of the system with a separation from the elastic limiter of oscillations (Fig. 1, c) on the boundaries of linear sections in accordance with the work [2]:

$$\frac{\varepsilon^2}{|1-\varepsilon^2|} \cdot \frac{\sin \varphi + \sin(\tau_x + \varphi)}{2} \leq q \leq \frac{\varepsilon^2}{1-\varepsilon^2}; \quad (1)$$

where

$$\varepsilon^2 = \frac{c}{m\omega^2}; \varphi = \frac{\pi + \tau_x}{2}; \left(\varphi = \frac{3\pi + \tau_x}{2} \right)$$

*Corresponding author: Povitroflotsky Avenue 31, Kyiv (Ukraine), email: i_nazar@i.ua

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Figure 1: Calculation models of vibration systems :
a – discrete-continuous, b – discrete (two mass), c – discrete (one mass)

In the dependencies (1) the following symbols are adopted: c - the stiffness of the restrictor; $q = \frac{Q}{F_0}$ - the ratio of the weight of the vibrating machine ($Q = mg$) to the amplitude of the force F_0 of perturbation; ω - frequency of perturbation force; τ_x - the duration of movement of the vibrator in contact with the elastic limiter of oscillations: $\tau_x = \frac{\pi}{\varepsilon}$; φ - the phase angle, which in the calculations is assumed to provide a positive value at the boundaries $c = \infty (\varepsilon = \infty, \tau_x = 0)$, condition (1) gives the value:

$$q = 1 \quad (2)$$

From (2) it follows that the upper and lower boundaries of regions of stable modes are combined.

However, the stability of periodic regimes under absolutely rigid constraints of oscillations is determined by the ratio:

$$1 \leq q \leq \infty \quad (3)$$

When comparing relations (2) and (3), it follows that the existence of stable periodic modes of motion of systems with elastic constraints (with stiffness) at the limit value does not pass into the condition of stability under the motion of the corresponding systems with the rendering of absolutely strict constraints. To eliminate this disparity, the consideration of shock interaction is the replacement of the elasticity and dissipation of the constraints by the shock pair, taking into account the length of the play. Such an approach allows to take into account the shock interaction in the vibration system by the pulse theorem and the recovery rate of the impact velocity.

$$q_{onm} = \frac{1+R}{1-R} \cdot \frac{\sqrt{2(1+\cos \tau)}}{(2\pi \cdot i - \tau_x)^2} \left(2\pi \cdot i - \tau_x + 2tg \frac{\tau_x}{2} \right) \quad (4)$$

Then the maximum impact velocity will be determined by the dependence:

$$\dot{y}_0 \max = q_{onm} \frac{2\pi \cdot i - \tau_x}{1+R} \quad (5)$$

Consider now the motion of the vibrator in the range $0 \leq \tau \leq \tau_x$.

In dimensionless variables we will have:

$$\ddot{y}_1 + 2n\dot{y}_1 + \varepsilon^2 y_1 = \cos(\tau + \varphi) + mg \quad (6)$$

where $\varepsilon^2 = \frac{c_0}{m\omega^2}$; $2n = \frac{n'}{m\omega}$, n' - dissipation rate.

$$y_1(0) = y(2\pi \cdot i) = 0; \quad \dot{y}_1(0) = \dot{y}(2\pi \cdot i) = \dot{y}_0 \quad (7)$$

$$\begin{aligned} y_1 &= \dot{y}_0 a_1 + b_1 \sin \varphi + c_1 \cos \varphi + d_1 q, \\ \dot{y}_1 &= \dot{y}_0 \dot{a}_1 + \dot{b}_1 \sin \varphi + \dot{c}_1 \cos \varphi + \dot{d}_1 q. \end{aligned} \quad (8)$$

$$\begin{aligned} a_1 &= e^{-n\tau} \varepsilon_1^{-1} \sin \varepsilon_1 \tau; d_1 = (1 - \dot{a}_1 - n a_1) \varepsilon^{-2}; \\ b_1 &= [(a_1 - \sin \tau) \cos \psi + (\cos \tau - d_1 - n a_1) \sin \psi] \cdot \left[(1 - \varepsilon^2)^2 + 4n^2 \right]^{\frac{1}{2}}; \\ c_1 &= [(\cos \tau - d_1 - n d_1) \cos \psi - (a_1 - \sin \tau) \sin \psi] \cdot \left[(\varepsilon^2 - 1)^2 + 4n^2 \right]^{\frac{1}{2}}; \\ \varepsilon_1 &= \sqrt{\varepsilon^2 - n^2}; tg \psi = \frac{2n}{\varepsilon^2 - 1}. \end{aligned} \quad (9)$$

$$y_1(\tau_x) = y(\tau_x) = 0; \quad \dot{y}_1(\tau_x) = \dot{y}(\tau_x) = -R \dot{y}_0 \quad (10)$$

By substituting the conditions (10) in equation (8), as well as the meaning $\sin \varphi$ and $\cos \varphi$ through q_{onm} and $\dot{y}_0 \max$ in accordance with formulas (8) and (10), we obtain dependences for determining R and τ_x :

$$R = \frac{\delta \left(\dot{c}_1 + \dot{b}_1 ctg \frac{\tau_x}{2} \right) + 2\lambda_1 (\dot{d}_1 + \delta \dot{a}_1)}{\delta \left(\dot{c}_1 + \dot{b}_1 ctg \frac{\tau_x}{2} \right) - 2\lambda_1 (\delta + \dot{d}_1)}, \quad (11)$$

$$2\lambda_1 \left[\dot{d}_1(d_1 - 1) - a_1(\delta + \dot{d}_1) \right] + \left(c_1 + b_1 c g \frac{\tau_x}{2} \right) \\ \left[2\dot{d}_1 + \delta(1 + d_1) \right] - \left(\dot{c}_1 + \dot{b}_1 c g \frac{\tau_x}{2} \right) (2d_1 + \delta a_1) = 0$$

where $\lambda_I = \frac{2}{\delta} + c g \frac{\tau_x}{2}$; $\delta = 2\pi \cdot i - \tau_x$.

According to the work [1]:

$$\tau_x = \frac{\pi}{\sqrt{\varepsilon^2 - n^2}}; R = e^{-n\tau_x}. \quad (12)$$

Determine now the condition of existence and stability of periodic regimes. The scope of actual values is limited by the ratio:

$$q \leq \frac{\lambda \sqrt{\varepsilon_I^2 + \varepsilon_2^2}}{f_I}. \quad (13)$$

For the real occurrence of periodic regimes, it is necessary to provide certain restrictions of system parameters and perturbations determined by the stability conditions of these regimes. For a system (Fig. 1, c), the boundary of stability regions of periodic regimes is determined by the equation:

$$\pm y_0(1 \pm R)^2 + (2\pi \cdot i - \tau_x)[q(1 + R) + \cos(\tau_x + \varphi) + R \cos \varphi] = 0$$

Analysis of the results indicates the existence of several zones of stability (Fig. 2), which is important for determining the parameters of the vibration damaging system, which are known to be reduced to two mains [2]:

$$\varepsilon = \sqrt{\frac{c}{m\omega^2}} \quad q = \frac{mg}{F_0} \quad (14)$$

Figure 2: Stability map of vibro-impact machines

To study the intensity of dynamic influence in work, the method of dynamic loop hysteresis is used. Expressions for areas of a hysteresis loop with various load laws are obtained. In the non-symmetric law in the stationary mode we have:

$$\Delta W = \frac{\sigma_{cm}^2 \gamma \pi}{2E_0(1 + \gamma^2)} \left[1 + \frac{(1 - K_\tau)}{K_\tau^2} \right], \quad (15)$$

Dependences (15) are given for the stationary (established) mode of motion, that is, the energy ΔW going on the densification of the unit volume of the mixture is constant. To estimate the energy required to compaction the mixture from the initial value of density to the required technology, expression (15) is represented as:

$$E_n = \frac{t}{T} \Delta W_0 v \frac{K_s - 1}{K_s \ln K_s} \quad (16)$$

where t_n - the length of the technological process; ΔW_0 - area of the hysteresis loop at the initial moment of sealing the mixture; K_s - coefficient characterizing changes in the area of the loop of hysteresis in the process of oscillations.

On the basis of formulas (15) and (16) expressions for specific power and the coefficient of efficiency, which has the greatest value $K_\tau = 1/\sqrt{2}$. A generalizing criterion

for assessing the effectiveness of vibrational effects is presented, which, along with the contact stress and deformation velocity σ , takes into account the slope of the load wave profile:

$$\frac{\sigma_{cm} \cdot v_{\varepsilon_1} \cdot \Gamma\left(\frac{n_1 + 1}{2}\right)}{\Gamma\left(\frac{n_1 + 1}{2}\right)} = \frac{\sigma_{cm} \cdot v_{\varepsilon_2} \cdot \Gamma\left(\frac{n_1 + 1}{2}\right)}{\Gamma\left(\frac{n_2 + 1}{2}\right)} \quad (17)$$

where $\Gamma(x)$ - gamma function X ; v_{ε_1} - strain rate; n - coefficient characterizing the steepness of the front.

The system in which the score (17) will be higher will be more effective.

According to research results, the principles of the creation of new progressive vibrating machines for the building industry in the formation of multi-hollow slabs and foundation blocks have been developed.

2. CONCLUSIONS

1. A theory is proposed for the interaction between the working parts of vibration machines and media to be processed, which, on the basis of an estimate of the stress state of the medium in the contact zone, makes it possible to determine the effect of the medium being processed on the dynamics of the machine working parts.

2. The principles of creation of volumetric forming machines with highly effective parameters for energy intensity and material consumption are proposed.

3. The zones of effective operation of the parameters and characteristics of the machine-environment system are found to ensure the high quality of the sealing process, which is based on the idea of the purposeful use of elastic characteristics of the general vibration system,

which, under all the same conditions, also results in the reduction of energy costs for some machines up to 50%.

4. The analytical dependences are obtained for estimating the energy of considering the interaction of the working organ and the medium on the basis of the use of the method of dynamic hysteresis loops.

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