earth; 3) flat-radial conductive heating; 4) spherical-radial conductive heating of anisotropic rocks.

Originality. The problem of conductive heating of homogeneous and anisotropic-heterogeneous rocks with the specification of vertical, flat-radial and spherical-radial flows (the last one combines the first two) including the natural thermal flow from the entrails of the earth.

Practical value. The application of the heterogeneous rock conductive heating method during planning the influence on the formation bottom zone and the wells, which produce paraffinaceous oils, may allow increasing their productive rates.

Keywords: *conductive heat treatment of a well, considering natural heat flow from the bowels of the Earth, spherically radial heat flow*

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DYNAMICS OF VIBRATING MACHINES STARTING WITH UNBALANCED DRIVE IN CASE OF BEARING BODY FLAT VIBRATIONS

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ДИНАМІКА ПУСКУ ВІБРАЦІЙНИХ МАШИН З ДЕБАЛАНСНИМ ПРИВОДОМ У ВИПАДКУ ПЛОСКИХ КОЛИВАНЬ НЕСУЧОГО ТІЛА

Purpose. Improvement of dynamic and power descriptions of the vibration machines with inertia vibration exciters.

Methodology. The vibration mechanics approaches and motions direct division method were used for analytical researches. The vibromachines running processes were designed using the numeral integration of mechanical vibration system motion equalizations and the electromagnetic transients equalizations in asynchronous electric motors in the Maple software environment.

Findings. Expressions for vibration moments (additional dynamic loading caused by the bearing body's vibrations) during the resonant zone passage by vibration machines with the bearing body flat vibrations both with one arbitrarily located vibration exciter and with two self-synchronization vibration exciters for the different starting modes have been received in an analytical form. The running process improvement possibilities of vibration machines with unbalanced vibration exciters were demonstrated by using the "double" (in case of one vibration exciter) and "separate" (in case of two vibration exciters) electric motors starting methods. The first method bases on using semi-slow vibrations arising in the resonant zone, and that in the engine-off case in this zone, the vibratory torque acting on its rotor becomes rotating. The conditions when the separate starting is effective were shown. The conclusions and practical recommendations that allow facilitating vibration machines with an unbalanced drive starting were drawn.

Originality. Theoretical positions of vibration machines with unbalanced vibration exciters running dynamics that are operated by the asynchronous electric motors with limited power received further development.

Practical value. The results of the research allow us to decrease the working body resonant vibrations of vibration machines, dynamic loading on the electric motor rotor and machine construction elements, electric drive necessary power, and currents inrush.

Keywords*: vibration machine, unbalanced vibration exciter, resonant zone, double starting, separate starting, vibration moment, self-synchronization*

Posing the problem and its connection with the main scientific tasks. The problems solutions of vibration systems with inertial drive run-up and run-down is of considerable interest for vibration technical devices. At inertial vibro-exciter passage through the natural frequencies zone, the resonances vibrations may occur which cause both a dynamic loads sufficient rise on the electric motor rotor, on machine bearing construction elements and additional power losses in the system. So, the vibration machine with unbalanced drive start it is necessary the electric drive power sufficiently exceeding the power necessary for operating in stationary mode (2–5 times as large by some data). In addition to that, in case of large machines with the drive from asynchronous type electric motors the inrush starting current influences negatively on the feeding electrical grid.

In order to vibrations level decline while passing the resonance zone the various methods and means are used – from vibro-exciters with automatically regulated unbalance static moment to algorithms with feedback. No doubt, that for successful realization of the last ones it is important to have more thorough conception about the occurring processes dynamics.

The analysis of latest investigations. The survey of investigations, concerning the inertial vibro-exciter resonance zone passing may be found in [1–3]. In the last years, a number of tasks have been solved based on vibration mechanics approaches, in particular, by using the method of motions direct separation. In [3] the example of the simplest system with bearing body linear vibrations and one unbalanced exciter it is shown that such approach important advantage is its results comparative simplicity and physical integration.

In work [1] attention is paid to the system motion peculiarity nearby the resonance – the availability of the so-called inner pendulum and its "semi-slow" motions, which are physical based efficiency of starting control some methods of the vibration machines with inertial vibration exciting.

A great number of works are dedicated to the use of the self-synchronization phenomenon in vibration machines and devices, data are shown in [1, 3], and the latest ones in [4–6]. However, no attention was paid to the dynamics of such vibrations machines starting. The presented paper is dedicated to generalization and development of these works results [2, 7, 8].

Statement of the task. The majority of vibration machines with unbalanced drive may be idealized in the form of a system, consisting of a single bearing solid body, which may execute plane-parallel motion and is connected with stationary base with elastic and damping elements (fig. 1). As bearing body vibrations exciters, the unbalanced vibro-exciters (misbalanced rotors) driven by the asynchronous type electric motors are mostly used. Motion equations of such system may be written down in the following form (see, for instance, [1–3])

$$M\ddot{x} + \beta_x \dot{x} + c_x x = m_i \varepsilon_i \sum_{i=1}^{s} (\ddot{\varphi}_i \sin\varphi_i + \dot{\varphi}_i^2 \cos\varphi_i) ;$$

$$M\ddot{y} + \beta_y \dot{y} + c_y y = m_i \varepsilon_i \sum_{i=1}^{s} (\ddot{\varphi}_i \cos\varphi_i - \dot{\varphi}_i^2 \sin\varphi_i),$$

$$J\ddot{\varphi} + \beta_\varphi \dot{\varphi} + c_\varphi \varphi =$$

$$= \sum_{i=1}^{s} m_i \varepsilon_i r_i (\ddot{\varphi}_i \cos(\varphi_i + \delta_i) - \dot{\varphi}_i^2 \sin(\varphi_i + \delta_i)), \quad s = 1 \ldots n \tag{1}$$

$$I_i \ddot{\varphi}_i = L_i(\dot{\varphi}_i) - R_i(\dot{\varphi}_i) +$$
$$+ m_i \varepsilon_i \left[\ddot{x} \sin\varphi_i + \ddot{y} \cos\varphi_i + r_i \ddot{\varphi} \cos(\varphi_i + \delta_i) + g \cos\varphi_i \right], \tag{2}$$

where M, J – are correspondingly, mass and moment of the bearing body inertia on the axis which passes through its gravity center; x, y, φ – are coordinates, determining the bearing body's position; φ_i – are the vibro-exciters rotation angles; r_i and δ_i – are polar coordinates of vibro-exciters axes; m, ε – are, correspondingly, the exciter's mass and eccentricity; I_i – the drive rotating parts inertia moment applied to the vibro-exciter shaft; c_x c_y, c_φ – are the elastic elements horizontal, vertical and rotational rigidities; β_x, β_y, β_φ – are the viscous resistance coefficients; $L_i(\dot{\varphi}_i)$, $R_i(\dot{\varphi}_i)$ – is the electric motor torque and forces moment of resistance to rotation; g – is a free-fall acceleration.

The basic material exposition. To solve the set of equations (1), (2) we use the method of motions direct separation of [1, 3]. As a zero-order approximation let us accept $\varphi_i = \omega t$, $q_i = P_i \sin\omega t + Q_i \cos\omega t$ where $\omega = \omega(t)$ – are slow and $q_i = x, y, \varphi$ – are fast changing time functions. Then it is not difficult to come from the original equations system of vibro-exciters rotors motion (2) to their rotation equations in the resonance zone in the form, obtained in [3]

$$I_i \dot{\omega} = L_i(\omega) - R_i(\omega) + V_i(\omega), \tag{3}$$

where

$$V_i(\omega) = m_i \varepsilon_i \left\langle \ddot{x} \sin\varphi_i + \ddot{y} \cos\varphi_i + r_i \ddot{\varphi} \cos(\varphi_i + \delta_i) \right\rangle.$$

The angle brackets in (3) point out at averaging for the $T = 2\pi$ by fast time $\tau = \omega t$.

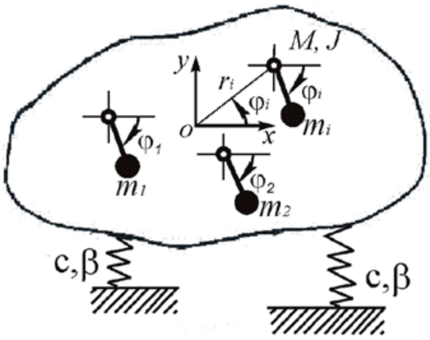


Fig. 1. General diagram of vibration system with unbalanced vibro-exciters

It should be noted that equation (3) differs from classic equation of machinery by presence of addend $V_i(\omega)$ – vibration moment which defines the vibration system conduct peculiarity. Presence of vibration moment explains both Sommerfeld's effect and vibro-exciters self-synchronization. The vibration moment determination is of main interest.

It should be noted that equation (3) keeps its form, obtained for the system with the bearing body linear vibrations [3] and for the examined more general case as well. Only expression for vibration moment has more complicated structure, and its obtaining algorithm remains unchanged, only computing difficulties grow up.

Vibration systems with one vibro-exciter Sommerfeld's effect. The certain part of present time machines operating has one unbalanced vibro-exciter. Using the considered approach it is not difficult to obtain the vibration moment expressions in the resonance zone for the case of vibro-exciter, placed arbitrarily as to centre of masses of the plane vibrating bearing body in the form

$$V(\omega) = -\frac{(m\varepsilon\omega)^2}{M} \left[\frac{n_x}{B_x^2} + \frac{n_y}{B_y^2} + \frac{Mr^2}{J} \frac{n_\varphi}{B_\varphi^2} \right]; \tag{4}$$

$$B_q = \sqrt{\left(1 - \lambda_q^2\right)^2 + 4n_q^2}\ ; \quad \lambda_q = \frac{p_q}{\omega}; \quad n_q = \frac{\beta_q}{2M_q \omega},$$

where p_q – are the frequencies of the natural system vibration.

Here, if $q = x, y$, then $M_q = M$, if $q = \varphi$, then $M_q = M\dfrac{\rho^2}{h^2}$; in addition to that, $n_\varphi = \dfrac{\beta_\varphi}{2J\omega}$.

As we can see, all addends in formula (4) are negative. Hence, vibration moment is always braking one, that is, it is an additional dynamic load upon the engine rotor, its dependence from frequency is of resonance character and, therefore, an essential braking effect is represented in comparatively natural frequencies narrow range. In addition to that, the value $V(\omega)$ rapid growth at approaching to resonance just explains the possible frequency "sticking" in the process of starting (Sommerfeld's effect) and, as consequence, the necessity of overrated (from starting conditions) post resonance vibromachines drive power. Such conclusion follows from diagramed presentation of dependences $L(\omega)$ and $M_{sum} = R(\omega) - V(\omega)$ (fig. 2), which intersection points abscissas correspond to possible stationary modes (curves L describe electric engines statical characteristics). Stability of motions is easily determined geometrically by the slope angles tangent signs and values to curves $L(\omega)$ and M_{sum}. It is evident that the resonance curve right slopes cannot be realized. According to the figure, the presence of several resonance peaks of the vibration moment curve may lead to the appearance of additional (as compared with the system of bearing body linear vibration) curves intersection points. Therefore, there exists a possibility of several stationary motion modes, having different angular velocities (up to seven, four of which may be stable). However, there are only two, principally different motion modes: "sticking" (curves 1) of the system with deficient power engine in the resonance zone (the motor coming in the process of running to this mode, would not be able to overcome the resonance peak) and far postresonance mode with frequency, which is close to the electric motor nominal frequency. If the motor power is sufficient, then, as a rule, after some breaking in the resonance zone, the system rapidly (upsetting) passes to far postresonance motion mode (curves 2).

To reach by the exciter the working frequency, the motor moment should overcome vibration moment $V(\omega)$ during its running. According to (4), maximal (peak) moment value $V(\omega)$ is as larger as less a damping n_q and higher than the own system vibrations frequency p_q. Hence, it is important not to overrate the rigidity value of elastic elements; the use of elastic suspension with a controlled rigidity change may be effective; it is possible to lower the resonance vibration amplitudes as well as the drive power by installation of maximal vibrations dampers. Expression (3) may be presented in the form of the sum of "partial" vibration moments v_q, which characterize the vibrations impact, corresponding to each of the generalized coordinates

$$V(\omega) = \sum_{q=x,y,\varphi} v_q ,$$

where

$$v_q = \frac{1}{2} F a_q \sin\gamma_q ; \; \sin\gamma_q = -\frac{2n_q}{B_q} ; \; a_q = \frac{m\varepsilon}{M_q B_q} ; \; F = m\varepsilon\omega^2 .$$

It is natural that maximal breaking effect is performed by "partial" vibration moment which corresponds to the highest natural vibration frequency p_q, so it is often enough to use just such vibration damper.

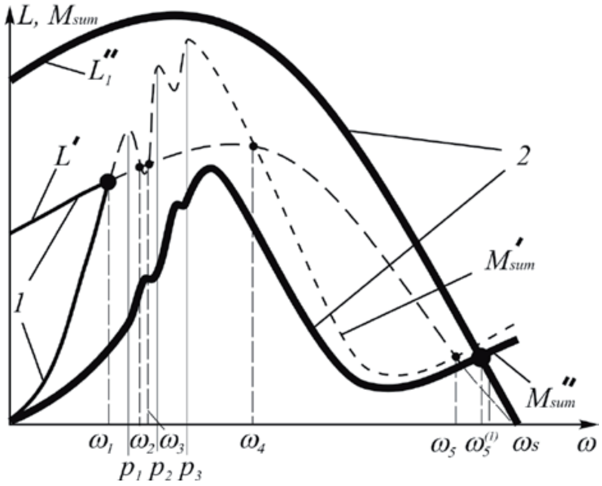


Fig. 2. Stationary modes of vibro-exciter rotation: 1 – "sticking" in resonance zone; 2 – far postresonance mode

It is clear from formula (4) that vibromachine starting at the working load absence is more difficult than at its presence; that to make the start easier it is advisable to install vibro-exciter in the system masses centre or as close to it, as possible. Therefore, the breaking vibration moment, resonance vibrations and, correspondingly, the necessary motor power are sufficiently less for centre-drilled system (fig.3, *a*) than, for instance, for the diagram shown in fig.3, *b* (in the first case the last component in vibration moment formula (4) disappears). It should also be noted that "rapid" (with frequency 2ω) vibration moment vibrations do not take place in such system in the steady mode that is favorable for the system durability.

On the other hand, taking into account the fact that the vibration moment value depends, first of all, on the vibro-exciter rotor running velocity, and to make starting easier, the engines with higher starting moment are recommended (it facilitates, also, the solution of the unbalance mass increase problem at first half-turn); at prescribed static moment unbalance mass should be designed with minimal inertia moment. So, the vibro-exciters constructions with laid on unbalanced mass are more preferable for the vibration amplitude changing, then those, having regulated static moment. In addition to this, it is recommended to exclude from the construction (if they are available) synchronizers, mechanical transmissions and so on, using the self-synchronization phenomenon, applying controlled electric drive.

Sommerfeld's effect representation during the vibromachine's run-up is visually demonstrated by the numerical modelling results, obtained for vibration system (fig. 3, *a*) with parameters $M = 330\,kg$; $J = 8{,}02\,kg \cdot m^2$; $c_y = c_x = 4{,}5 \cdot 10^5\,N/m$; $c_\varphi = 2{,}8 \cdot 10^4\,N \cdot m$; $m = 40\,kg$; $\varepsilon = 0{,}036\,m$, electric engine – asynchronous, with rotation frequency

$n_c = 1500\,\mathrm{rot/min}$, with power $P = 1{,}5\,kW$. According to fig.4, at passing the natural frequencies zone ($t = 0{,}15 - 0{,}42\ s$) the dynamic load upon the electric engine rotor grows sufficiently (curve 1); as we can see, the vibration moment value is several times larger than in stationary mode and its maximal vibrations are compatible with the engine starting moment.

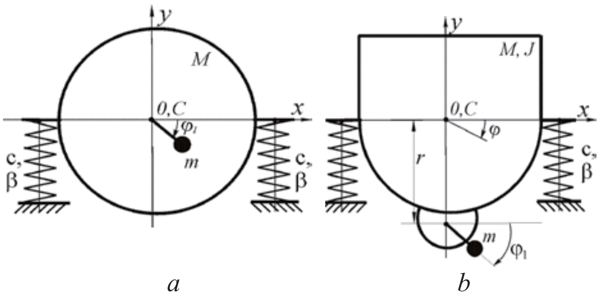


Fig. 3. Diagrams of vibromachines: a – with centrally installed vibro-exciter; b – with shifted vibro-exciter

Correspondingly, the exciter rotor running velocity slows down intensively up to short-term rotation frequency stabilization (curve 2), in addition to that, the bearing body maximal resonance vibration are being excited. Just after resonance passing, the vibration moment value decreases fast sufficiently and its vibrations cover positive zone (that is, it becomes rotating in some moments of time). Then their damping takes place as to low negative level (determined by resistance to the bearing body vibrations); the bearing body vibrations amplitude decreases fast as well and the engine rotating moment value changes from starting to nominal value (curve 3).

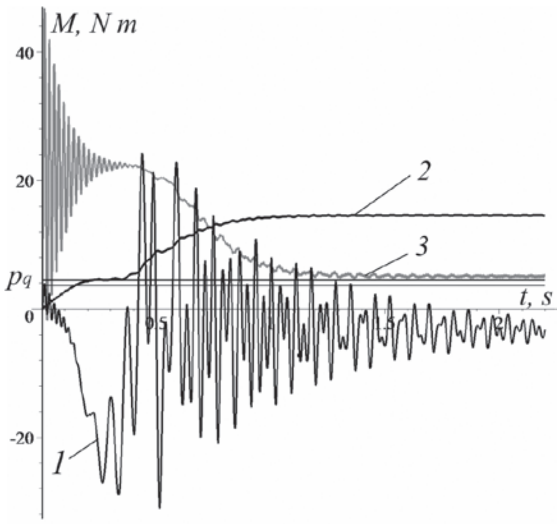


Fig. 4. Changing in time: 1 – of vibrational moment; 2 – of vibroexciter velocity; 3 – of engine moment

As it follows from the vibro-exciter rotation velocity diagrams for the cases of driving electric motor with different powers (fig.5) at motor replacing with power $P = 1{,}5\,kW$ into motor with power $P = 2{,}2\,kW$, the exciter velocity slowing down in resonance zone is practically absent (curve 3); while for the motor with power $P = 1{,}1\,kW$ its steady postresonance operating mode becomes impossible (curve 2 – the angular velocity “sticking” in postresonance zone).

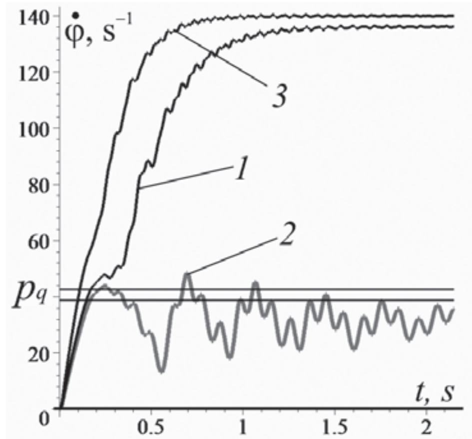


Fig. 5. The vibroexciter velocity changing in time: 1 – $P = 1{,}5\,kW$; 2 – $P = 1{,}1\,kW$ (“sticking” of velocity); 3 – $P = 2{,}2\,kW$

The vibration machines with unbalanced drive double start of. In such machines practical use the so called method of “double starting” is applied for vibrations level decreasing during the resonance frequencies passing. Its technical realization is rather simple. Method consists in switching-off and subsequent switching-on the electric motor in the resonance zone in predetermined moments of time. This method theoretical grounding with account of vibration mechanics standpoints facilitates its wider use. The method basis lies in two existing system motion regularities close to the Sommerfeld’s effect representation area: the first one – at motor switching-off in the resonance zone the vibration moment effecting the vibro-exciter rotor becomes positive, that is, rotating (it follows from the basic vibrational mechanics equation (3), written down for the stationary mode case); the second one – availability of so called inner pendulum and its “semi-slow” motions. So, using the motion direct separation method and accepting as the first approximation $\varphi_1 = \varphi_1^{(1)} = \omega t + \psi$; $q = q^{(0)} + q^{(1)}$, for general system (fig. 1) in case of one vibro-exciter it is not difficult to obtain the “semi-slow” vibrations equation in the form [2]

$$\ddot{\Psi} + 2n_1\dot{\Psi} + B\sin\Psi - \mathrm{P}\sin^2\frac{\Psi}{2} = 0\,, \qquad (5)$$

for the system under consideration

$$B = \sum_{q=x,y,\varphi} b_q\,;$$

$$b_q = \frac{(m\varepsilon\omega^2)^2}{2MI}\,\frac{p_q^2 - \omega^2}{(p_q^2 - \omega^2)^2 + 4n_q^2\omega^4}\,;\ P = \sum_{q=x,y,\varphi} \rho_q^2\,;$$

$$\rho_q = \frac{(m\varepsilon\omega^2)^2}{MI}\,\frac{2n_q\omega^2}{(p_q^2 - \omega^2)^2 + 4n_q^2\omega^4}\,;\ 2n_1 = k / I\,,$$

k – is the damping coefficient.

The value $q=\sqrt{|B|}$ is frequency of the inner pendulum small free vibrations provided that the rotor ω rotation frequency changing slowly [2]. The semi-slow vibrations occurrence effect in the resonance zone may be observed in the fig. 5–7. In addition to that, according to fig. 6, a (curve 1) semi-slow (with frequency $2q$) vibration moment vibrations take place after motor switching-off with regard to the shifted to the positive side level.

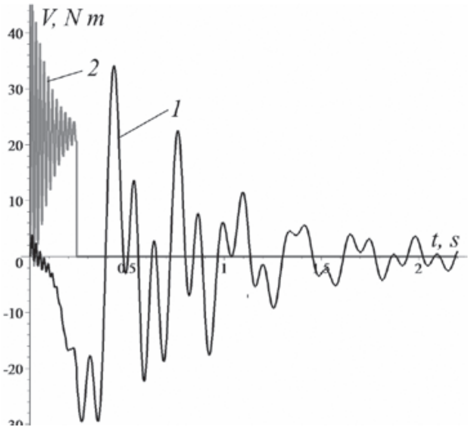
Fig.6, *b* and fig.7 demonstrate the running realization possibility and coming to the rotation mode with frequency, close to nominal of the "insufficient" power ($P=1{,}1\,kW$) motor with the help of "double starting" method. As one can see, the method successful use necessary condition is, first of all, the effect upon the vibro-exciter rotor in the moment of the motor repeated switching-on (in figures $\mathrm{t}_{\text{rep. sw.–on}}=0{,}48\ \mathrm{s}$) of rotating vibration moment commensurable with its starting moment. It is not difficult to realize the above-mentioned condition with the help of the electric motors control modern means. The applied recommendations are to switch off the motor in the moment of the bearing body intensive resonance vibrations growing and at once (in a period of time that equals to semi-slow vibrations $t=2/q$ semi-period) to switch it on again.

Vibration systems with two self-synchronizing exciters. Separate starting. Many modern vibration machines, in particular, screens and platforms with directed vertical (horizontal) vibrations are realized by the diagram, shown in fig. 8. For this vibration system, the vibration moments expressions influencing in resonance zone upon the exciters rotors rotating in opposite directions are presented in the form

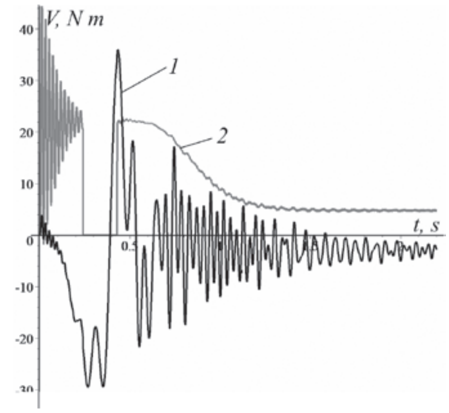
$$V_i(\omega)=-\frac{1}{2}\frac{(m\varepsilon\omega)^2}{M}\frac{n_y}{B_y^2}. \qquad (6)$$

Vibration machines with self-synchronizing exciters permit the electric motors separate (in turns) start possibility; however, it is not applied in practice. Using the approach under consideration, it is possible to demonstrate such start possible advantages.

It is not difficult to establish that vibration moment in case of only one vibro-exciter running will equal $V_{separ}(\omega)=\frac{1}{4}V(\omega)$, where $V(\omega)$ is determined by formula (4). It follows from analysis (4) that if the natural vibration system p_q frequencies are sufficiently different (it may always be reached by the choice of elastic elements), then with the frequency ω growing in the running processes each addend, except the one that corresponds to $\omega\approx p_q$, will be negligibly low. Taking into account the fact that ratio Mr^2/J for the considered dynamic system is sufficiently less than one it is possible to come to the vibration moment value following estimation, functioning in the resonance zone in case of its electric motors separate start: $V_{separ}(\omega)\approx\frac{1}{2}V_i(\omega)$. Therefore, by the system parameters corresponding choice at the motors separate start it is possible to attain the resonance vibration moments decrease and, as a result, to attain all connected with this vibromachine dynamic and power characteristics possible improvements.



a



b

Fig. 6. Changing in time: 1 – of vibration moment; 2 – of motor moment ($P=1{,}1\,kW$): a) switching-off of the motor in the resonance zone, $\mathrm{t}_{\text{sw.–off}}=0{,}3\ \mathrm{s}$*; b) the engine double starting,* $\mathrm{t}_{\text{sw.–off}}=0{,}3\ \mathrm{s}$, $\mathrm{t}_{\text{rep. sw.–on}}=$ $=0{,}48\ \mathrm{s}$

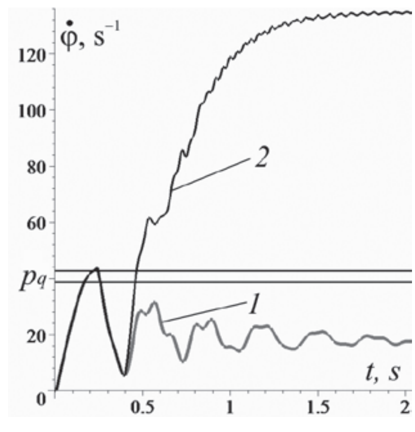


Fig. 7. The vibro-exciter velocity changing in time ($P=1{,}1\,kW$): 1 – switching-on the motor in the resonance zone, $\mathrm{t}_{\text{sw.–off}}=0{,}3\ \mathrm{s}$*; 2 – the motor double start,* $\mathrm{t}_{\text{sw.–off}}=0{,}3\ \mathrm{s}$, $\mathrm{t}_{\text{rep. sw.–on}}=0{,}48\ \mathrm{s}$

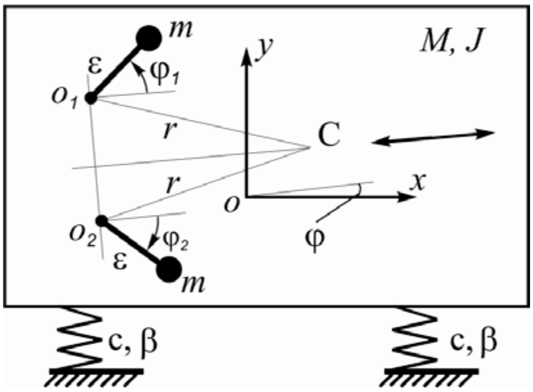


Fig. 8. Diagram of vibration machine with self-synchronizing exciters

In favor of vibration moments decrease at self-synchronizing vibro-exciters, the separate start provides the fact that, owing to the differences between their phases, some bearing body "collateral" vibrations occur and that it is necessary at more precise vibration moment determination that effects the *i-y* vibro-exciter, to take into account the other exciters' effect. That is, at the vibration moment calculation it should be presented in the form of two items sum, one of which (being determined above) represents additional load, caused by power losses at vibrations, and the second (as a rule, noticeably less in the resonance zone) is caused by the other vibro-exciters' influence. It should be noted, that the second item represents the power redistribution between the vibro-exciters. Formulas for its value determination for many vibration systems may be found in specialized literature [1,3]. The positive effect may be amplified by installation of vertical vibrations damper. Besides, in case of motors separate start using, the starting currents decrease (almost doubled) is rather important.

It should be noted that somewhat electric drive excessive power is recommended for easing the start in case of vibromachines with two selfsynchronizing exciters. In addition to that, the unbalanced exciter rotation vibration support effect should be used in steady state, working with one switched off motor. Especially the rotation vibration support mode is the most stable for the studied dynamic system. It follows from comparison (4) and (6) that dynamic load upon the electric motors rotors and, correspondingly, the necessary electric drive total power in case of the vibromachine working part vertical vibrations will be sufficiently less than at elliptic trajectory.

Results of simulation confirm the separate start advantages. Thus, for instance, according to fig. 9 ($Mr^2 / J = 0,52$), in such start case, the vibration resonance amplitudes of the bearing body masses centre are sufficiently less than those at the motors synchronous start. In addition to that, the horizontal and turning resonance vibrations amplitudes grow. However, their amplitudes are far from the vertical vibrations amplitudes maximal values.

Findings. Thus, the majority of inertial vibro-exciters behavior mechanisms and vibration system in general at the resonance zone passing may be explained on the vibration mechanics approaches basis. On this basis, the practical conclusions and recommendations improving the vibromachine with unbalanced drive dynamic and power characteristics may be obtained.

The methods of electric motors double and separated start are effective for easing the start of postresonance vibromachines with unbalanced drive.

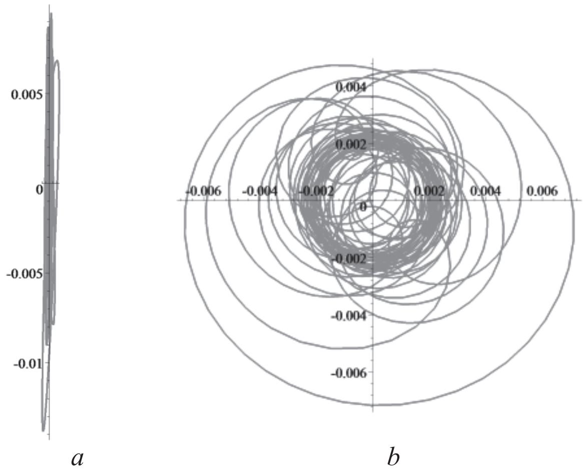


Fig. 9. The bearing body masses centre trajectories: a – synchronous (ordinary) start of motors; б – start of one motor (separate start)

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Ярошевич М.П. Про деякі особливості динаміки розбігу вібраційних машин зі збудниками, що самосинхронізуються / М.П. Ярошевич, А.В. Силивонюк // Науковий вісник НГУ. – 2013. – №4. – С. 37–45.

Мета. Покращення динамічних та енергетичних характеристик вібраційних машин з інерційними віброзбудниками.

Методика. Для аналітичних досліджень використаний підхід вібраційної механіки та метод прямого розділення рухів. Моделювання процесів розбігу вібромашин виконане за допомогою чисельного інтегрування рівнянь рухів механічної коливальної системи та рівнянь електромагнітних перехідних процесів в асинхронних електродвигунах у програмному середовищі Maple.

Результати. В аналітичній формі отримані вирази для вібраційних моментів (додаткового динамічного навантаження, викликаного коливаннями несучого тіла) під час проходження зони резонансу вібраційними машинами з пласкими коливаннями несучого тіла як з одним довільно розміщеним віброзбудником, так і з двома віброзбудниками, що самосинхронізуються, для різних режимів пуску. Демонструються можливості поліпшення процесу розбігу вібромашин з дебалансними віброзбудниками шляхом використання методів „подвійного" (у випадку одного віброзбудника) та „роздільного" (у випадку двох збудників) пуску електродвигунів. Показано, що в основу першого методу покладене використання напівповільних коливань, що виникають у зоні резонансу, а також те, що, у випадку вимкнення двигуна в цій зоні, вібраційний момент, який діє на його ротор, стає обертальним. Вказані умови, за яких є ефективним роздільний пуск. Наводяться висновки та практичні рекомендації, що дозволяють полегшити пуск вібраційних машин з дебалансним приводом.

Наукова новизна. Отримали подальшого розвитку теоретичні положення динаміки розбігу вібраційних машин з дебалансними віброзбудниками, що приводяться в рух асинхронними електродвигунами обмеженої потужності.

Практична значимість. Результати роботи дозволяють зменшувати резонансні коливання робочого органу вібраційних машин, динамічні навантаження на ротор електродвигуна та елементи конструкції машини, необхідну потужність електроприводу, а також величину пускових струмів.

Ключові слова: *вібраційна машина, дебалансний віброзбудник, резонансна зона, подвійний пуск, роздільний пуск, вібраційний момент, самосинхронізація*

Цель. Улучшение динамических и энергетических характеристик вибрационных машин с инерционными вибровозбудителями.

Методика. Для аналитических исследований использованы подходы вибрационной механики и метод прямого разделения движений. Моделирование процессов разбега вибромашин выполнено с помощью численного интегрирования уравнений движений механической колебательной системы и уравнений электромагнитных переходных процессов в асинхронных электродвигателях в программной среде Maple.

Результаты. В аналитической форме получены выражения для вибрационных моментов (добавочной динамической нагрузки, вызванной колебаниями несущего тела) при прохождении зоны резонанса вибрационными машинами с плоскими колебаниями несущего тела как с одним произвольно расположенным вибровозбудителем, так и с двумя самосинхронизирующимися вибровозбудителями для разных режимов пуска. Демонстрируются возможности улучшения процесса разбега вибрационных машин с дебалансным приводом путем использования методов „двойного" (в случае одного вибровозбудителя) и „раздельного" пуска электродвигателей (в случае двух возбудителей). Показано, что в основе первого метода лежит использование полумедленных колебаний, возникающих в зоне резонанса, а также то, что, в случае выключения двигателя, в этой зоне вибрационный момент, действующий на его ротор, становится вращающим. Указаны условия, при которых эффективен раздельный пуск. Приводятся выводы и практические рекомендации, которые позволяют облегчить пуск вибромашин с дебалансным приводом.

Научная новизна. Получили дальнейшее развитие теоретические положения динамики разбега вибрационных машин с дебалансными вибровозбудителями, которые приводятся в движение асинхронными электродвигателями ограниченной мощности.

Практическая значимость. Результаты работы позволяют уменьшать резонансные колебания рабочего органа вибрационных машин, динамические нагрузки на ротор электродвигателя и элементы конструкции машины, необходимую мощность электропривода, а также величину пусковых токов.

Ключевые слова: *вибрационная машина, дебалансный вибровозбудитель, резонансная зона, двойной пуск, раздельный пуск, вибрационный момент, самосинхронизация*

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