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RESEARCH OF THREE-MASS RESONANCE EQUIPMENT WITH AERO-INERTIAL DRIVE

The efficiency of resonance three-mass vibratory machine with aero-inertial drive is considered. Previously established principles of design and parameters, taking in account Zommerfelds effect in resonance operating mode, are used. The main properties of existing pneumatic vibrators are examined and the possibility of their use according to proposed approach is established. The approach in choosing of the driver is proposed.

Keywords: resonance vibratory machine, aero-inertial drive, Zommerfelds effect, pneumatic vibrators.

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ДОСЛІДЖЕННЯ ТРИМАСОВОЇ РЕЗОНАНСНОЇ ВІБРАЦІЙНОЇ МАШИНИ З АЕРОІНЕРЦІЙНИМ ПРИВОДОМ

Розглянута ефективність резонансних вібраційних машин з аероінерційним приводом. На основі попередніх досліджень представлені основні положення проектування таких машин та вибору їх параметрів з урахуванням явищ, пов'язаних з ефектом Зоммерфельда. Досліджені властивості існуючих пневматичних віброзбуджувачів та можливість їх застосування виходячи з запропонованого підходу розрахунку. Запропонована розрахункова методика вибору приводу системи.

Ключові слова: резонансні вібраційні машини, аероінерційний привід, ефект Зоммерфельда, пневматичні віброзбуджувачі.

Introduction. Vibratory equipment is widespread in the different fields of industry. Nowadays pneumatic vibrators acquired in various kinds of vibratory machines. The most efficient mode of work of such machines is resonance, which allows to reach the higher amplitude under the same generating force. It is a problem while operating in near-resonance region because of freezing of the rotational frequency due to Zommerfelds effect. Use of pneumatic vibrators has some advantages in compare with another kinds of such drives and it make possible to apply earlier developed approach [1, 2, 3] when the particular kind of the structure and mode of operation is used.

The development of vibratory equipment directed to improve the efficiency of such machines. The most optimal operation mode is resonance due to it's high efficiency. But for providing of resonance mode it is needed the use of control devices, which increases the cost of the vibratory equipment.

It was developed the approach to create near resonance aero-inertial vibrating machine. Here Zommerfeld effect (hanging of the vibration frequency) is used to set the near resonance operating mode.

Research and publications analysis. The expensive control systems are used to make the constant operating mode in vibratory equipment and to keep the near resonance frequency which allows to increase the efficiency of vibratory equipment due to high dynamic factor. And also the problem of freezing of frequency due to Zommerfelds effect in near resonance region can have a bad influence on electromechanical properties of the engine.

Research purpose. Development of approach in creation of near-resonance three-mass equipment with aero-inertial drive is presented. The approach in selection options of vibrator and driver is needed for use of proposed algorithm of design to create the most efficient vibratory machines in practice.

The main material. Three-mass vibrating machine with aero-inertial drive is formed from three oscillating masses and unbalanced drive. The disturbance of forced oscillations in three-mass vibratory machine with aero-inertial drive (Fig. 1) is provided through the mass m_3, the inertial value of which is less than of other masses. The oscillations of this mass occur by the independent linear coordinate x_3. The dissipation of energy in resonance elastic structure with stiffness c_{23} is reflected by the damper with the resistant factor μ_{23}.

From the previous investigations the three-mass structure can be described by the next system of equations [1, 2, 3]:

$$\begin{cases} m_1 \ddot{x}_1 \cdot + c_{12} \cdot (x_1 - x_2) + c_{ins} \cdot x_1 + \mu_1 \dot{x}_1 + \mu_{12} \cdot (\dot{x}_1 - \dot{x}_2) = 0; \\ m_2 \cdot \ddot{x}_2 + c_{12} \cdot (x_2 - x_1) + c_{23} \cdot (x_2 - x_3) + \mu_{12} \cdot (\dot{x}_2 - \dot{x}_1) + \mu_{23} \cdot (\dot{x}_2 - \dot{x}_3) = 0; \\ m_3 \cdot \ddot{x}_3 + c_{23} \cdot (x_3 - x_2) + \mu_{23} \cdot (\dot{x}_3 - \dot{x}_2) = m_{un} \cdot r \cdot \omega^2 \cdot \sin(\omega \cdot t). \end{cases} \quad (1)$$

And after converting into matrix and some designations the expressions for determination of the amplitude were gotten X_1, X_2 and X_3 accordingly of the masses m_1, m_2 and m_3 of three-mass vibratory machine

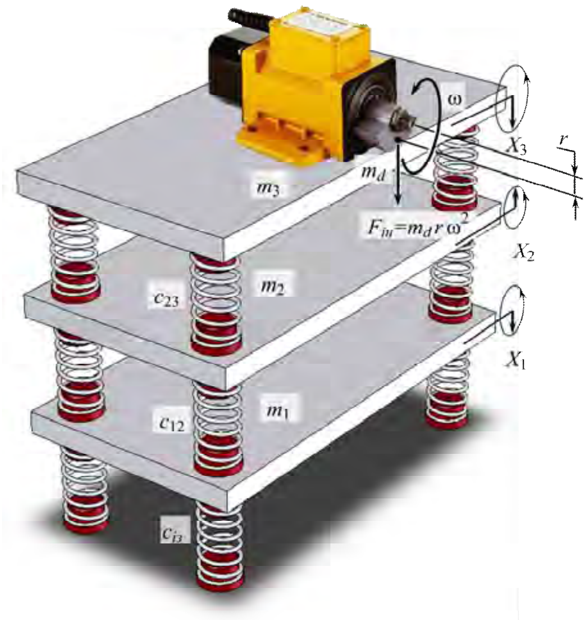

Fig. 1. Principle scheme of three-mass vibratory mace with inertial drive

$$X_1(\omega) = \frac{-F_{ін} \cdot k_{12}(\omega) k_{23}(\omega)}{\Delta}; \quad X_2(\omega) = \frac{F_{ін} \cdot k_{11}(\omega) k_{23}(\omega)}{\Delta};$$

$$X_3(\omega) = \frac{F_{ін} \cdot (k_{12}(\omega) k_{21}(\omega) - k_{11}(\omega) k_{22}(\omega))}{\Delta}, \quad (2)$$

Where $\Delta = k_{12}(\omega) k_{21}(\omega) k_{33}(\omega) - k_{11}(\omega) k_{22}(\omega) k_{33}(\omega) + k_{11}(\omega) k_{23}(\omega) k_{32}(\omega)$ – determinant of the matrix coefficients of the unknowns in the system of equations and

$$k_{11}(\omega) = c_{12} + c_{iз} - m_1 \omega^2 + i(\mu_1 + \mu_{12})\omega \; ; \; k_{12}(\omega) = k_{21}(\omega) = -c_{12} - i\mu_{12}\omega \; ;$$

$$k_{13}(\omega) = k_{31}(\omega) = 0 \; ; \; k_{22}(\omega) = c_{12} + c_{23} - m_2 \omega^2 + i(\mu_{12} + \mu_{23})\omega \; ;$$

$$k_{23}(\omega) = k_{32}(\omega) = -c_{23} - i\mu_{23}\omega \; ; \; k_{33}(\omega) = c_{23} - m_3 \omega^2 + i\mu_{23}\omega \; .$$

For establishing of the stiffness values c_{12} and c_{23} it is needed to take in account that three mass structure with an arbitrary selected options has two own system resonant peaks $\Omega_{в1}$ and $\Omega_{в2}$. In such occasion choosing acceptable for design two natural frequencies $\Omega_{в1}$ and $\Omega_{в2}$, with determined masses m_1, m_2 and m_3 and make the determinant equal zero, neglecting the resistant ratio and vibroinsulators stiffness, the stiffness c_{12} through the first natural frequency $\Omega_{в1}$ and c_{23} through the second- $\Omega_{в1}$ can be calculated. The values of stiffness will provide designed oscillating system

$$c_{12} = \frac{m_1 \cdot \omega_{b1}^2 \cdot \left[m_2 \cdot m_3 \cdot \Omega_{b1}^2 - c_{23} \cdot (m_2 + m_3)\right]}{m_3 \cdot \Omega_{b1}^2 \cdot (m_1 + m_2) - c_{23} \cdot (m_1 + m_2 + m_3)};$$
$$c_{23} = \frac{m_3 \cdot \Omega_{b2}^2 \cdot \left[m_1 \cdot m_2 \cdot \Omega_{b2}^2 - c_{12} \cdot (m_1 + m_2)\right]}{m_1 \cdot \Omega_{b2}^2 \cdot (m_2 + m_3) - c_{12} \cdot (m_1 + m_2 + m_3)}. \tag{3}$$

Inertial parameters m_1 and m_2 should be set by engineer [2, 3] and the value of m_3 could be established according to two known masses. With two given frequencies $\Omega_{в1}$ and $\Omega_{в2}$ the mass m_3 must have such value which would be satisfactory for the structure [1,2,3].

The values of the first and the second oscillating masses, which are structurally given, are being formed with the condition [1,2] $m_1 = (3\ldots5)m_2$.

The stiffness of vibroinsulators is established from the condition of that the frequency of the vibrating machine grounded on the vibroinsulators should have natural frequency three – five times lower than forced one. So from [1], [2].

$$c_{із} = m_c\left[\frac{\Omega}{(3\ldots5)}\right]^2. \tag{4}$$

The most effective operation mode for such machine is such where the forced vibrations will take place in near resonance region of the first peak of the structure.

Using technologically needed frequency X_1 and mass m_1 of the operating body and taking in account that the magnitude of the disturbing inertia force can be determined as $F_{in} = m_d\, r\, \Omega^2$, so the magnitude of unbalanced mass m_d can be determined.

With the aim to determine the disturbing moment the parameters and first and the second natural frequencies Ω_{b1} and Ω_{b2} of three-mass structure should be chosen.

Using the established value of the unbalanced mass and amplitudes of masses [1,2] the needed moment for disturbing oscillations of three-mass vibrating machine can be calculated

$$M = \frac{m_d^2 \cdot r^2 \cdot \omega^2 \cdot \lambda_3}{2 \cdot m_3} \cdot \left[\sin\left(\frac{\mu_{23} \cdot \lambda_3}{m_3 \cdot \Omega}\right) + \frac{1}{\sqrt{2}}\right] + \mu \cdot \Omega, \tag{5}$$

which is needed to give the motion to unbalanced air-screw (Fig. 2).

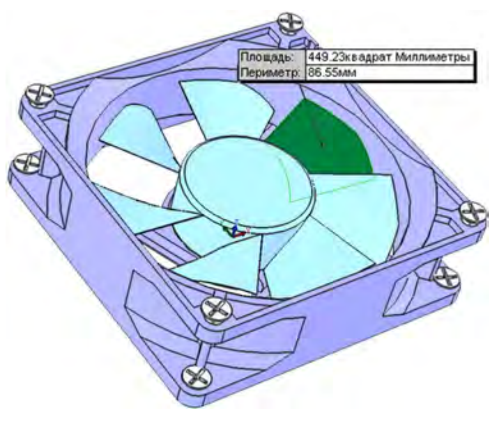

Fig. 2. The experimental vibrator for three-mass resonant aero-inertial machines

The use of such drive (Fig. 2) has a great disadvantage because of the big lost of air. So the efficiency in that occasion can't be high [6]. But the developed approach can be used in the vibratory equipment with pneumatic vibrators (Fig. 3).

There is no a big difference in disturbing of oscillations between proposed aero-inertial drive and ball or turbine pneumatic vibrators due to similar structure and principle of operating. Algorithm of calculating developed for aero-inertial drive of three-mass vibratory machine also can be applied for near-resonance three-mass vibratory machine with pneumatic vibrator.

For providing high efficiency of operating environment the needed oscillating amplitude should be laid.

Fig. 3. Examples of pneumatic turbine and ball vibrators

Needed power N for disturbing of one-mass oscillating system can be determined from the next equations:

$$N_{зб} = \frac{1}{T}\int_0^T P(t)\cdot \dot{x}(t)dt\,, \tag{6}$$

where $P(t)$ is the law of changing of disturbance force; $x(t)$ is the law of changing of velocity of oscillating mass; T is a period on which integration is held. It is known [5] that the law of changing of disturbance force through the dynamic factor is

$$P(t) = \frac{Xc}{\lambda}\sin(\omega t) = \frac{mX\omega^2}{\lambda}\sin(\omega t), \tag{7}$$

where X is the amplitude of oscillations of mass m; c is the stiffness of the elastic system; λ is dynamic factor of mechanic oscillating system. Motion of the mass m is offset according to the phase ε relatively to the disturbance moment. So the law of motion velocity of oscillating mass change is next

$$\dot{x}(t) = \frac{d}{dt}X\sin(\omega t - \varepsilon) = \omega X\cos(\omega t - \varepsilon). \tag{8}$$

We can get the expression for determining the power by integrating the expression (6) over the period $T = \frac{2\pi}{\omega}$ and taking in account expressions (7) and (8) and considering the case of phase offset $\varepsilon = \frac{\pi}{2}$, when consumption power reaches the maximum value. So

$$N_{зб} = \frac{\omega}{2\pi}\int_0^{\frac{2\pi}{\omega}} \frac{mX\omega^2}{\lambda}\sin(\omega t)\omega X\cos(\omega t - \varepsilon)dt = \frac{\sqrt{6}}{4}\frac{X^2\omega^3 m}{\lambda}. \tag{9}$$

The expression for determining of the disturbance power [5] is formed with using (9) for n-mass structure of oscillating structure (according to the principle of superposition) and taking fixed definite set frequency of the forced oscillations (denoting ω = Ω) introducing the efficiency of the vibrator it can be transformed next way

$$N_{зб} = \frac{\sqrt{6}}{4}\frac{\Omega^3}{\eta}\sum_{i=1}^{n}\frac{X_i^2 m_i}{\lambda_i}, \tag{10}$$

where X_i, λ_i is the amplitude of oscillations and dynamic ratio of mass m_i; η is the efficiency of the vibratory drive. Using (10) the power of needed drive can be determined [4, 5].

Knowing all needed parameters such as needed disturbing moment, power, oscillating amplitude and all the other technological parameters we can choose the kind of the drive. Now there is a large amount of manufactured pneumatic vibrators which can be chosen using handbooks due to established parameters by proposed algorithm of designing of vibrating equipment [1,2,3].

Turbine vibrators provide almost silent operation — just 65-75 dB throughout their entire service life, ball vibrators wear and the ball and ballrace become pitted, the noise they produce rises sharply and can reach up to 100 dB.

Turbine vibrators features sealed bearings that are prelubricated for life and do not require additional lubrication. In compare ball vibrators require airline lubrication. Ball vibrators draw up to 50% more air than comparable turbine vibrators.

Turbine vibrators feature high efficiency throughout their service life, while ball vibrator begins to lose speed and efficiency from the moment it is put into service because the ball and ball race become pitted with use.

So compare of ball and turbine pneumatic vibrators allows to determine the highest efficiency of turbine. Such turbine vibrators are powered by a turbine, with suitable lead counterweights, put on rotation by air. Turbine is supported by screened and pre-greased bearings, therefore lubricated air is not necessary; for this reason these models are suitable for environments were a high cleanliness is required (such as food or pharmaceutical industry). Anti-spark materials of vibrators such as aluminum and brass with its function, without any slithering between parts, allows to use them in fire hazard environments.

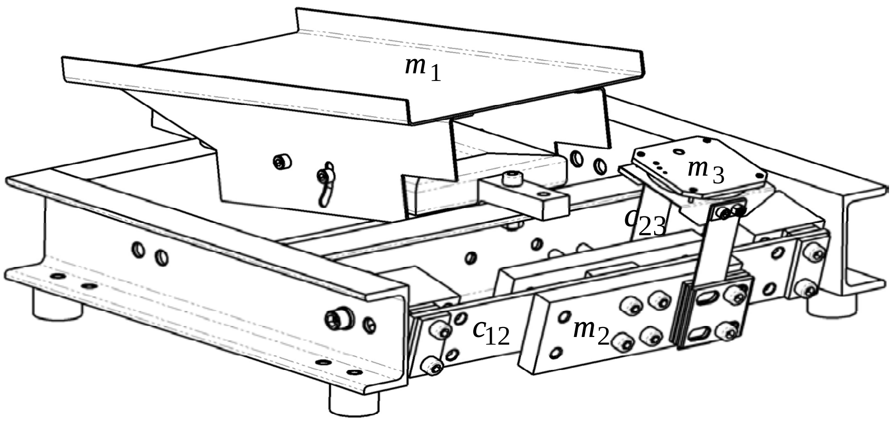

Fig. 4. Proposed structure of three-mass vibratory machine where the pneumatic vibrator can be used

Proposed structure of vibrating machine (Fig. 4) can be applied for equipment where the ball or turbine vibrators are used.

Use in complex proposed approach in designing of resonance three mass structure with a high dynamic factor and law cost pneumatic ball or turbine vibratory gives an opportunity to create an efficient vibratory equipment with a number of advantages and special fields where it can be applied.

Conclusion. In the previous research of aero-inertial resonance equipment the simple airscrew was used. Therefore it was a great consumption of air. But the use of simple airscrew allowed to simplify experimental investigation. Analyzing pneumatic vibrators it is recommended to use turbine vibrators. The turbine contour, the feeding ports shape and their position, are designed to obtain maximum performances; moreover the clearance between turbine and body is very small and allows the total exploitation of air push. Due to these characteristics, the efficiency is very high while air consumption and especially the noise, are very low. Efficiency, referred to all the models, shows the ratio between under load and no-load performances (force and frequency). On request special versions are available: completely proof, for aggressive environments, for high temperature, etc. The structural similarity of the experimental aero inertial drive (unbalanced air screw) and pneumatic vibrator gives opportunity to use the same calculations for more efficient vibratory equipment with such advantages as frequency adjustment and absence of reverse action on engine, low air consumption, easy installation of vibratory, cost efficient vibration.

The great advantage of such kind of the equipment with the pneumatic drive is that it can be used in potentially explosive environment, food and pharmaceutical industries.

1. Кузьо І. В. Вибір структури та обґрунтування силових і жорсткістних параметрів вібраційної машини з аероінерційним збуренням / І. В. Кузьо, О. В. Ланець, Я. В. Шпак // Серія: Галузеве машинобудування, будівництво: Збірн. наук. праць Полтавського нац. техн. ун-ту ім. Ю. Кондратюка. – 2012. – Вип. 2 (32). Т. 1. – С. 120–131. 2. Кузьо І. В. Встановлення конструктивно-силових параметрів та дослідження руху коливальних мас у вібраційній машині з аероінерційним збуренням / І. В. Кузьо, О. В. Ланець // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні: Український міжвідомчий наук.-техн. зб. – Львів: Вид-во Львівської політехніки, 2012. – Вип. 46. – С. 28–44. 3. Кузьо І. В. Встановлення оптимальних параметрів аероінерційної вібраційної машини з умови забезпечення мінімального збурювального момента / І. В. Кузьо, О. В. Ланець, В. І. Лозинський // Оптимізація виробничих процесів і технічний контроль у машинобудуванні: Вісник Нац. ун-ту “Львівська політехніка”. – Львів, 2012. – № 746. – С. 100–104. 4. Блехман И. И. Вибрационная механика / И. И. Блехман. М.: Изд-во “Фізматлит”, 1994. – 400 с. 5. Ланець О. В. Реалізація ефекту Зоммерфельда у вібраційному майданчику з інерційним приводом / О. В. Ланець, Я. В. Шпак, В. І. Лозинський, П. Ю. Леонович // Автоматизація виробничих процесів у машинобудуванні та приладобудуванні. – 2013. – Вип. 47. – С. 12–28. 6. Кузьо І. В. Експериментальне дослідження вібраційних машин з аероінерційним збуренням / І. В. Кузьо, О. В. Ланець, В. М. Гурський // Вібрації в техніці та технологіях. – 2012. – № 4 (68) – С. 48–56.