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ABOUT ONE AUTO-RESONANCE VIBRO-IMPACT DEVICE

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Abstract. *The paper deals with the problems of modeling vibro-impact autoresonance machines. The methods and capabilities of modern mechatronics allow you to create autoresonance systems that allow you to balance the work of friction forces, as well as even the forces themselves. This article briefly describes the theory of such systems, and then considers an example of calculating a device that allows you to reproduce the necessary periodic effects. Models of dynamic devices with arbitrary dynamic matching operators are considered. The analysis of the organization of autoresonance technological machines of vibro-impact action, based on the creation of a feedback scheme, including the measurement of impulses of the working body, which is the determining parameter of movement in a conservative model of process, is carried out. Examples and calculation formulas are given.*
Keywords: *vibro-impact system, dynamic compliance, autoresonance, feedback, energy balance, impact impulse.*

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ОБ ОДНОМ АВТОРЕЗОНАНСНОМ ВИБРОУДАРНОМ УСТРОЙСТВЕ

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Аннотация. *В статье рассматриваются проблемы моделирования виброударных авторезонансных машин. Методы и возможности современной мехатроники позволяют создавать авторезонансные системы, которые позволяют сбалансировать работу сил трения, а также даже сами силы. В этой статье кратко описывается теория таких систем, а затем рассматривается пример расчета устройства, позволяющего воспроизводить необходимые периодические режимы движения. Рассмотрены модели динамических устройств с произвольными операторами динамической податливости. Выполнен анализ организации авторезонансных технологических машин виброударного действия, основанный на создании схемы обратной связи, включающей измерение импульсов рабочего органа, являющегося определяющим параметром движения в консервативной модели процесса. Приведены примеры и формулы расчета.*
Ключевые слова: *виброударная система, динамическая податливость, авторезонанс, обратная связь, энергетический баланс, ударный импульс.*

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1. Introduction

Vibration machines and devices are widely distributed and are used in agricultural and industrial production, mining, construction, and in many other industries. Among vibration machines, as a rule, machines of vibro-impact action are distinguished by the highest efficiency, since the technological processes they implement are associated with systematic shocks of solids [1-4].

Obtaining stable and predictable results of processing workpieces or media with such machines is possible only under the condition that they implement the most effective resonant processes under conditions of constantly changing technological load. Obtaining stable and predictable results of processing workpieces or media with such machines is possible only under the condition that they implement the most effective resonant processes under conditions of constantly changing technological load.

It turns out that these difficulties can be avoided if you create a self-oscillating excitation system. Such a system is implemented by introducing a special feedback loop, which generates an exciting force through a non-linear transformation of the signal proportional to certain parameters of the movement of the working body (tool, impacter, etc.).

When tuning, which is called autoresonance, with any changes in the parameters of the oscillating system and technological load, the resonant state is automatically maintained in the system [5].

Let the system established a periodic process. During the period of tool movement, the work of the excitation and dissipation forces must be balanced. That is, there should be a balance of works of non-conservative forces. In this paper, we consider the problem of modeling the device, which at each moment of time provides the balance of the forces themselves, that is, the dissipation forces are compensated not on average, but uniformly [7].

2. General view system

Consider the scheme of the vibro-impact device of a fairly general form. Using the methods described in [2, 6], we represent the object under study as a linear system with an arbitrary number of degrees of freedom (Fig. 1) containing a working body (impacter, etc.) and control subsystem. The impacter is modeled by a point solid of mass *m*. Since the workflow is vibro-impact, the system presented is strongly non-linear. Additional nonlinearities determine the feedback elements included in the control subsystem.

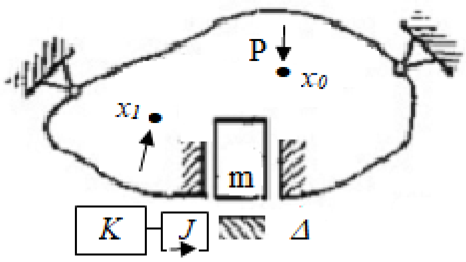

Fig. 1. General view of the studied mechatronic vibro-impact system

Suppose that as a result of constructing a correct mathematical model or as a result of carrying out full-scale measurements, a system of operators of dynamic compliance is specified [6, 8]. Such a system completely describes the linear part of the object under consideration. The working process consists in the organization of periodic collisions between the impacter and the fixed obstacle, for example, the workpiece surface.

Denote the x coordinate of the impacter with mass m. Suppose that the impacter performs one-dimensional oscillations along a certain axis. Value Δ>0 - there is a gap. Case Δ<0 corresponds to preload. The preload can be provided by the system design, as well as in the case if at the point x_0, a constant force P is applied, which presses the working member to the surface to be treated.
At point x_1, a control force K is applied, the form of which we are going to determine, based on the characteristics of the auto-oscillatory system. This force is obviously formed by the organization of the feedback loop. Near the point of contact of the impacter with the treated surface is placed a sensor that measures some parameters of its movement, for example, the impact impulse J. The control action is formed in accordance with the sensor signal.
We will designate the operators of dynamic compliance, which correspond to the force acting at the point x_n of moving the point x_k, as $L(x_n, x_k; p)$; p – an operator of differentiation. Given the scheme given in Fig.1, for the localization point of the workflow x you can write the operator equation

$$x(t) = L(x_0, x; p)P + L(x_1, x; p)K - L(x, x; p)\Phi(x, \dot{x}), \qquad (1)$$

Where the expression $\Phi(x, \dot{x})$ denotes the force of the impact interaction [9,10]. We will assume that the impact is modeled using the Newton hypothesis, that is, it is momentary, and we also assume that it is elastic.
Energy losses in the implementation of the workflow can be considered, for example, by introducing the corresponding components into the representation for the operator [2, 5], as well as by using special techniques for modeling impact using more realistic hypotheses [6].
Suppose, finally, that all converters and sensors included in the system are inertia-free.
In the absence of friction and control action, the equation of motion of the conservative system corresponding to (1) takes the form:

$$x(t) = L(x_0, x; 0)P - L(x, x; p)\Phi(x, \dot{x}), \qquad (2)$$

where it is considered that the force P = const. And, since here we neglect friction, it is assumed that for all operators $L(x, x; p) : \mathrm{Im}[L(x, x; i\omega)] = 0$.
Let $u \equiv L(x_0, x; 0)P$. Periodic mode with one impact interaction for the period of movement of the conservative system (T_0) under the assumption that the origin of time coincides with the impact can be represented as [6]:

$$x(t) = u - J\chi(t, \omega_0);\ \omega_0 = 2\pi T_0^{-1}, \qquad (3)$$

where the periodic Green function (PFG) corresponding to the operator $L(x, x; p)$ is given by a Fourier series of the form:

$$\chi(t, \omega_0) = \sum_{k=-\infty}^{\infty} L(x, x; ik\omega_0)\exp(ik\omega_0 t). \qquad (4)$$

The impact impulse in this case of an elastic impact will be given by the ratio $J = 2m|\dot{x}(-0)| \geq 0$ and be based on the impact condition. From here it is easy to get:

$$J = \frac{[u - \Delta]}{\chi(0, \omega_0)}. \qquad (5)$$

In a conservative vibroimpact system, the impact impulse turns out to be the integral of motion mutually uniquely associated with the total energy E.
The considered auto-oscillatory system will be organized in such a way that the feedback should be constructed as a result of measuring the values of any of the integrals of motion, for example, a

shock impulse. It will be shown that the result will be a convenient way for organizations of autoresonance vibration-impact machines.

The frequency ranges for the existence of the solution (3), (5) are determined in specific cases from the condition $x(t) \le \Delta$. In practice, the condition $J \ge 0$ is checked, which is equivalent in most cases. Relation (5) defines the equation of the skeleton curve ($J = J(\omega_0)$).

We rewrite the equation of motion (1) using the dynamic stiffness operator L^{-1}(p) [4, 6]. We have then:

$$L^{-1}(x,x;p)x = P_x + F(p,x) - \Phi(x,\dot{x}).\,, \qquad (6)$$

Here is designated: $P_x = L^{-1}(x,x;0)L(x_0,x;p)P; \quad F(p,x) = L^{-1}(x,x;0)L(x_1,x;p)K.$

We will form the control action $F(p,x)$ in such a way that in the initial system (1) it is possible to implement a periodic self-oscillations, which would retain the shape of the motion mode of the conservative system (3).

Let the dynamic stiffness operator be represented as follows: $L^{-1}(x,x;p) = W_1(x,x;p) + W_2(x,x;p)$, and $\mathrm{Im}W_1(x,x;p) = \mathrm{Re}W_2(x,x;p)$. Solution (3), (5) was constructed under the assumption that $W_2(x,x;p)$ $\equiv 0$. We substitute the representation (3) into the relation (6) at a certain value of frequency ω_0 satisfying the condition $x(t) \le \Delta$ and we find as a result

$$W_2(x,x;p)[u - J\chi(t,\omega_0)] = F[p, u - J\chi(t,\omega_0)]. \qquad (7)$$

We will further look for the function $F(p,x)$ in the subclass of functions with the structure $\{\beta(J)W(x,x;p)\}$, where a certain function $\beta(J)$ is differentiable on any finite interval, and $W(x,x;p)$ is a meromorphic function [6] of a complex variable *p*. Taking into account that $W_2(x,x;p)u = 0$, from the relation (7) we find: $W_2(x,x;p)[J\chi(t,\omega_0)] = \beta(J)W(x,x;p)[J\chi(t,\omega_0)]$. Therefore, it is necessary to fulfill the conditions

$$W_2(x,x;p) = W(x,x;p),\ \beta(J) = 1. \qquad (8)$$

Thus, the conditions determining the structure of the control force $F(p,x)$ are obtained. From the second relation (8), one can find the stationary values of the pulse J^0, and from the inversion of the skeletal curve $J(\omega)$, the frequencies of the self-oscillations.

The choice of function $\beta(J)$, along with the structure of the operator of the linear part of the system, determines the nature and efficiency of the control system. From what follows, it will be possible to see that it is very convenient to choose a monotonically decreasing function.

Let's choose, for definiteness, $\beta(J) = \beta_0 J^{-1},\ \beta_0 = \text{const}>0$. Then the stationary value of the impact impulse $J^0 = \beta_0$ and the frequency of self-oscillations are determined from the relation $\beta_0 = J(\omega)$. Since the value of the integral of motion (impact pulse) lies exactly on the skeletal curve corresponding to the realization of nonlinear resonances, the conditions found ensure the realization of autoresonance motions of the type described.

3. Sustainability research

Found solutions need to be investigated for sustainability. In engineering calculations, the so-called energy condition is often used [2, 3, 5]. Strictly speaking, the energy condition is only necessary. The efficiency of using this condition in applied problems is widely known [2–5].

We write the function $E(J)$ that corresponds to the balance of work of non-conservative forces for the period of movement *T*:

$$E(J) = -J^2 \int_0^T [W_2(x,x;p)\chi(t,\omega) - \beta(J)W_2(x,x;p)\chi(t,\omega)]p\chi(t,\omega_0)dt \quad (9)$$

Here the first term on the right side corresponds to the work of the dissipation forces, and the second to the work of the forces of an external energy source.

After some transformations we get $E(J) = -J^2\lambda[1-\beta(J)]$; $\lambda = \text{const}>0$

For asymptotically stable periodic regimes with a stationary value of the integral of motion (in this case, an impulse of impact), the dissipation forces stabilize the system and therefore, in accordance with the energy condition $\frac{dE(J)}{dJ} < 0$ (при $J = J^0$). Given the calculations done, we get

$$E_J \equiv \frac{dE(J)}{dJ} = J^2\lambda \frac{d\beta}{dJ}. \quad (10)$$

At E_J <0, self-oscillations are asymptotically stable. By choosing the function - monotonously decreasing, we ensured the stability of the system. In the case when $\beta(J) = \beta_0 J^{-1}$, $\beta_0 = \text{const}>0$: $E_J = -\beta_0\lambda < 0$. Thus, in accordance with the energy condition, the stationary mode of motion $J^0 = \beta_0$ is asymptotically stable. More precisely stability analysis can be performed using other modern methods [9–12].

From a technical point of view, the organization of such a regime allows, with a minimum of energy costs, to achieve maximum efficiency of the technological process. Questions about the features of the practical implementation of the corresponding device are an independent problem and are not discussed in this article.

We can specify several ways to solve it. Important recommendations can be found in [9].

In conclusion of the theoretical part of the paper, we note that the notion of autoresonance of the vibration system was introduced in the book [13]. This concept has proven to be very useful in practical engineering.

The principles of autoresonance are used in the design of many vibration, vibro-impact and ultrasonic machines and devices [3, 4, 5, 8, 9, 16]. These principles underlie the special types of cyclic robots [14, 15, 17], as well as many other important technical objects.

4. An example of a system with one degree of freedom

As an interesting example, consider a system with one degree of freedom [6, 7] (Fig.2.).

The linear oscillator vibrates, impacting with a fixed obstacle 2, on which impact impulses sensor is installed, which forming a signal proportional to their magnitude *J*. This signal is converted to some function $\beta(J)$. After multiplying with the signal coming from the speed sensor 1, the result is transmitted to the exciter 4. Assuming the impacter mass is equal to one, we write the equation of motion in the operator form

$$(p^2 + \Omega^2 + 2bp)x = -\Phi(x,\dot{x}) + \beta(J)px \quad (11)$$

where b>0 – coefficient of viscous friction. Because a system with a gap is considered here, the force P = 0. In the conservative case ($b = \beta(J) = 0$), solution (3), (5) has the form [6]:

$$x(t) = -J\chi(t);\ \chi(t) = [2\Omega \sin \tfrac{1}{2}\Omega T_0]^{-1} \cos\Omega(t - \tfrac{1}{2}T_0);\ J = -2\Omega \text{tg}(\pi\Omega\omega_0^{-1}). \quad (12)$$

The representation for the periodic Green function $\chi(t)$ in the second formula (12) takes place only for $t \in [0, T]$. Generally speaking for all $t \in (-\infty, \infty)$, here it is necessary to use the Fourier series (4), when $L(x, x; ik\omega_0) = (\Omega^2 - k^2\omega_0^2)^{-1}$. The set of natural frequencies of the conservative vibro-impact system $\Omega \leq \omega < 2\Omega$.

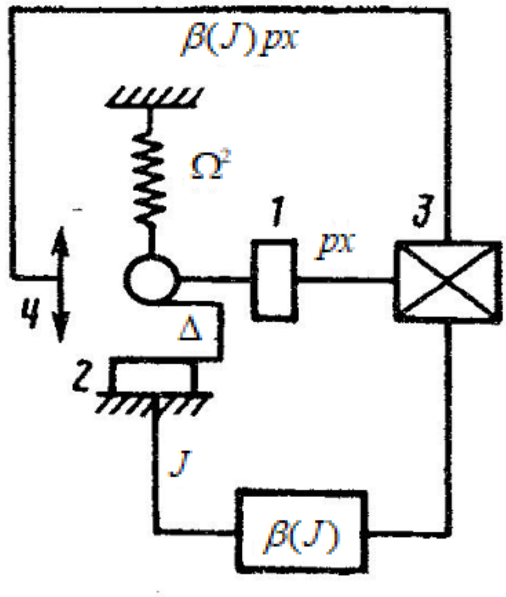

Fig. 1. Example of a system with one degree of freedom: 1 - speed sensor; 2 - impact impulses sensor; 3- signals multiplier; 4 – vibration exciter.

In this case $W_1(p)=p+\Omega^2$, $W_2(p)=2bp$. When $\beta(J) = \beta_0 J^{-1}$, the stationary value of the impact impulse turns out to be $J^0 = J^0 = 0{,}5\beta_0 b^{-1}$.

The solution has the form described by the first formula (12). From the third formula (12) we find the frequency of self-oscillations

$$\omega = \frac{\pi\Omega}{\pi - \mathrm{arctg}[\beta_0 / (4b\Omega\Delta)]}. \quad (13)$$

In accordance with the energy condition (10), the found mode is asymptotically stable. More accurate studies that can be performed numerically or using modern methods of nonlinear mechanics [12] confirm this conclusion.

Results and discussion

The main result of the article is to describe a specific class of autoresonance devices with a remarkable property: the impacter performs the movement characteristic of the conservative system. Thus, the considered systems can be attributed to the class of systems with full compensation of friction forces at each moment of time.

Final remarks.

1. It should be noted that the organization of the machines in question requires, obviously, to provide in the design a device intended for the rigid start of the vibro-impact mode of movement [4, 8], since at least one blow must occur for organizing the measurement and fixation of shock

impulses. Devices for the rigid start usually give the impacter additional energy, for example, the system is shaken.

2. For systems with a large number of degrees of freedom, the organization of an autoresonance scheme of the type in question may require the use of other integrals of motion
3. Accounting for the loss of energy during an impact using the recovery coefficient hypothesis, generally speaking, takes beyond the scope of this class of devices, since the rebound of the impacter simulated according to Newton cannot be accurately described using a conservative model. This circumstance is of a model nature and does not fundamentally affect the physical properties of the systems under consideration. Accounting for energy losses during an impact can be performed using an "equivalent correction" in the viscous friction coefficient [6]: $b = b_1 + r\pi^{-1}\Omega$, where b_1, is the "true" viscous friction coefficient, $r = 1 - R$, R is the coefficient recovery speed at impact ($0 < R \leq 1$).

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