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Российский фонд фундаментальных исследований

ВИБРАЦИОННЫЕ ТЕХНОЛОГИИ, МЕХАТРОНИКА И УПРАВЛЯЕМЫЕ МАШИНЫ

СБОРНИК НАУЧНЫХ СТАТЕЙ

**по материалам XI Международной
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«Вибрация – 2014»**

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Теоретические исследования и практическое использование вибрационных процессов в технике и технологиях являются актуальной сферой деятельности и научных интересов многих ученых в России и за рубежом.

Результаты этих работ нашли свое отражение в программе XI Международной научно-технической конференции «Вибрация-2014. Вибрационные технологии, мехатроника и управляемые машины».

Проведение конференции в Юго-Западном государственном университете стало уже хорошей традицией и закономерностью, подтверждающей успешное развитие научной школы по вибрационной механике, заложенной заслуженным деятелем науки и техники России, профессором П.М. Алабужевым. В сборнике, публикуемом по итогам ее работы, представлены результаты исследований ведущих ученых России, Германии, Франции, Непала, Латвии, Украины, Белоруссии.

Тематика представленных на конференции научных работ весьма широка и многогранна: история развития механики, динамика конструкций и машин, моделирование динамических процессов, виброзащита, волновые процессы и случайная вибрация, мехатроника, робототехника и биомеханика. Особое внимание уделено системам, имеющим автоматическое управление и регулирование.

Сборник будет полезен научным работникам, инженерно-техническим специалистам, аспирантам и студентам, занимающимся проблемами исследований в области динамики машин и разработкой современной вибрационной техники и вибрационных технологий.

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DAMAGE VIBRODIAGNOSTICS IN INITIALLY SYMMETRIC STRUCTURES

This paper considers new damage detection methods, based on utilization of specific vibration effects caused by the distortions of system's initial symmetry due to appearance of faults. Dynamic behavior of damaged structure under the test harmonic excitation is analyzed using two methods: analogue-digital simulation on specialized computer system and numerical solution with program ANSYS. It is shown that specific distortions of flexural modes and vibration spectrum can be used as diagnostic signs of defects. The main advantage of this approach lies in the more high detection sensitivity in comparison with traditional resonant frequency methods. Further rise of detection sensitivity can be achieved by insertion of additional nonlinear element into the structure of testing object.

Initially symmetric structural elements or units are widely used in machines and engineering objects (reinforced members of thin-walled structures, sections of shafts or pipelines, etc.). Damages, which can appear in such structures, disturb their initial symmetry, and that can be the reason for appearance in the system of specific vibration effects.

At present, to detect damages in engineering structures, vibration methods of nondestructive testing are widely used [1]. The majority of existing vibration techniques is based on the monitoring of changes in resonant frequencies [2] or in damping factors [3]. But these vibration procedures do not always come up to practical requirements because of their inherently low sensitivity to defects. For example [4], the presence of a crack, which makes up about 10~20% of the undamaged cross-sectional area, reduces the natural frequencies of a component only by 0.6~1.9%. Besides, natural frequencies and damping factors of real structures can change their values not only through the generation of cracks, but also due to relaxation, wear, etc., and this latter change of parameters may be highly significant (up to 10~15%). Other demerit of existing vibration methods lies in the necessity to determine natural frequencies and damping characteristics of the structure in its undamaged condition and in the further monitoring of changes of these parameters while structure is in use. Such a procedure is very laborious and requires many working hours of trouble-shooting.

This paper considers possibilities to increase damage detection sensitivity by the utilization of specific vibration effects caused by the distortions of system's initial symmetry due to appearance of defects. Besides, in accordance with findings of works [1,5], it has been found advantageous to insert an additional nonlinear elastic element into the structure of testing object in order to increase the detection sensitivity.

Dynamic model of testing object. Object of the study is a uniform viscoelastic fixed beam (e.g. span of a shaft) which is initially symmetric relative to the central section. In order to find possible defects, forced vibrations of the beam are excited with the aid of test harmonic force $P\sin\omega t$ applied in the middle section (Fig. 1).

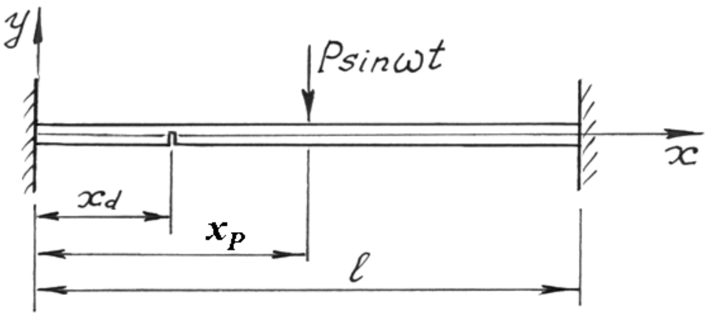

Fig. 1. Dynamic model of testing object

The differential equation of forced bending vibrations of the beam can be stated as follows:

$$EI\frac{\partial^4 y}{\partial x^4}+\mu\frac{\partial^2 y}{\partial t^2}+bEI\frac{\partial^5 y}{\partial t\cdot\partial x^4}=P\sin\omega t\cdot\delta(x-x_P)\,, \tag{1}$$

where EI and μ are the flexural rigidity in bending and the distributed mass of the beam; y is the lateral displacement of the beam cross-section with co-ordinate x; b is the coefficient of internal friction; P and ω are the amplitude and the frequency of the external harmonic excitation; $\delta(x-x_P)$ is a Dirac delta function; $x_P = 0.5l$ is the co-ordinate of the cross-section to which the external harmonic force is applied; l is a length of the beam.

Damage is simulated as a local reduction of beam's flexural rigidity EI in the cross-section with co-ordinate $x = x_d$. In this case the change of beam's flexural rigidity EI versus co-ordinate x can be mathematically described by the following expression:

$$EI(x) = EI_0[1 - q \cdot \delta(x - x_d)], \tag{2}$$

where $q = (1 - I_d/I_0)$ is a measure of the relative damage value; $I_\mathbf{d}$ is the second moment of the damaged cross-section; I_0 is the second moment of the undamaged cross-section; $\delta(x - x_\mathrm{d})$ is a Dirac delta function.

In the case of a fixed beam, the boundary conditions are as follows

$$y(0,t) = 0,\ \frac{\partial y}{\partial x}(0,t) = 0,\ y(l,t) = 0,\ \frac{\partial y}{\partial x}(l,t) = 0. \tag{3}$$

Dynamics of the system has been analyzed using two different methods: modeling on the specialized analogue-digital computer system developed in Riga Technical University [6, 7] and numerical simulation with program ANSYS.

The mathematical simulation has been carried out assuming the parameters of equations (1) – (3) according to the following conditions: $b\omega_1 = 0.00785$; $\omega_1 l^2\sqrt{\mu/(EI)} = 7.1$; $Pl^2/(EI) = 1.5$; $x_d/l = 0.25$; $x_\mathbf{P}/l = 0.5$. Here ω_1 is the first natural flexural frequency of the beam in the undamaged state. Other parameters of equations (1) – (3), expressed in dimensionless form ($p = Pl^2/(EI)$; $\nu = \omega/\omega_1$; q), have been varied within the limits of $p = 0.5 \div 2.0$, $v = 0.25 \div 3.50$ and $q = 0 \div 0.35$. Under such conditions the analysis of resonant vibrations corresponding to the two first flexural modes has became possible.

Vibration methods for damage detection. The dynamic behavior of the system under study is illustrated by the amplitude-frequency characteristic (AFC), which graphically represents mutual connections between the driving frequency ν and the half-swing of oscillations $u_0 = y_0/l$ in the beam's cross-section with co-ordinate $z = x/l = 0.4$ (Fig. 2). The dotted line corresponds to the case of undamaged beam ($q = 0$), but the full line – for the beam with damage ($q = 0.2$).

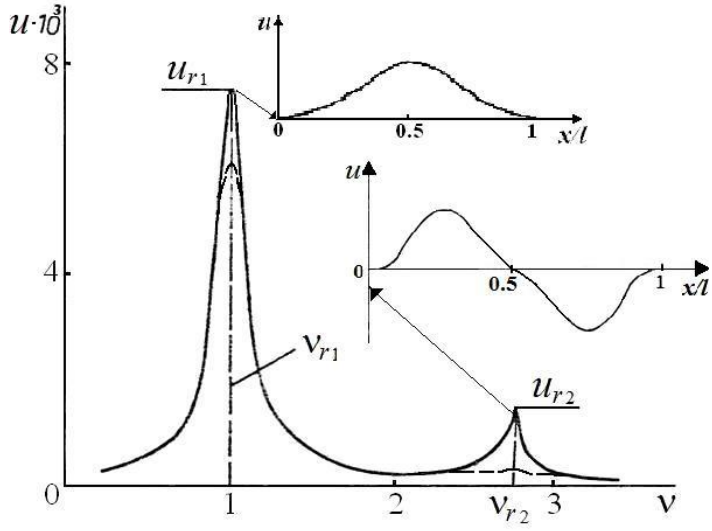

Fig. 2. Amplitude-frequency characteristic of the beam's flexural vibrations

Peak points of the AFC correspond to the first (frequency ν_{r1}) and the second (frequency ν_{r2}) modes of beam's resonant flexural vibrations. In the case of undamaged beam (dotted line on the AFC), the crest of the first mode and the nodal point of the second mode coincide with the co-ordinate $x_{\mathbf{P}}/l = 0.5$ of section where external harmonic force $P\sin\omega t$ is applied. Therefore excitation force $P\sin\omega t$ has a sufficient effect only on the first resonant regime ν_{r1} and practically does not influence the resonant oscillations of the second flexural mode.

After appearance of a defect, initial symmetry of the system is disturbed. As the result, the crest point of the first mode and the node of the second mode are shifted from the middle section of the beam in the direction of damage location. And the value of this shift is increased with the rise of defect. Therefore harmonic excitation force, applied in section $x_{\mathbf{P}}/l = 0.5$, begins active action on the beam not only at the first resonant mode, but also at the second one. In consequence of this, beam's flexural vibrations by the second resonant mode (frequency ν_{r2}) are sufficiently intensified, and the corresponding change of the AFC occurs (full line on the AFC).

Similar results have been received by numerical calculation with ANSYS (see Fig. 3). Modeling was performed using elements BEAM188.

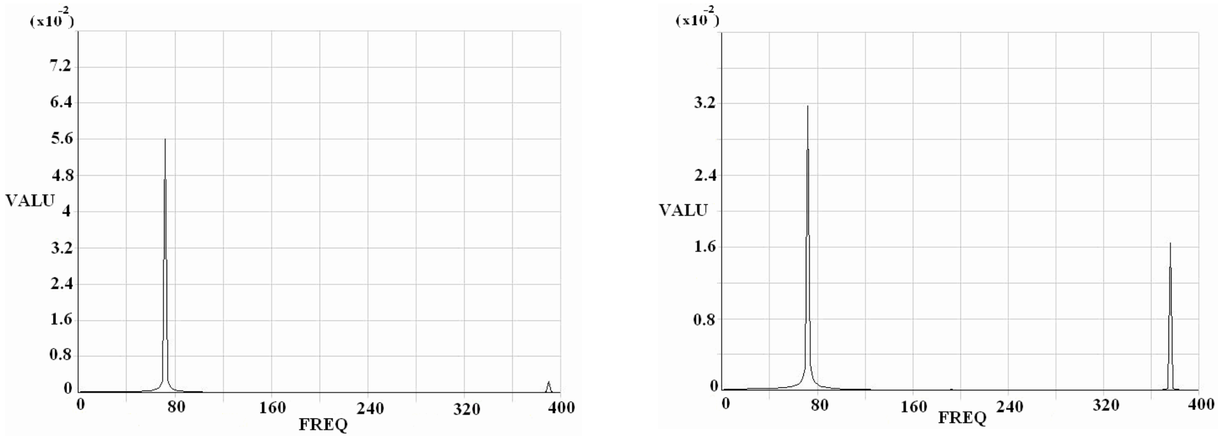

Fig. 3. Amplitude-frequency characteristics: left – for the beam with no damage; right – for the beam with damage

Distortions of flexural modes due to appearance of fault can be used as diagnostic signs of defects. Graphs of the first and the second modes of bending vibrations corresponding to the beam's undamaged (q = 0; full lines) and damaged (q = 0.2; dotted lines) states are presented in Fig. 4.

Specified distortions of flexural modes due to appearance of fault can be used in non-destructive testing as information signs of defect. In this case, fault can be detected by the recording of parameters which are definitively related to distortions of flexural modes: 1) a shift Δ_x of nodal (or antinodal) point of beam's flexural mode; 2) a difference in amplitudes Δu between two points of the beam equally distant from its middle (e.g. $\Delta u = u_1 - u_2$); 3) a ratio of amplitudes u_1/u_2 in two points of the beam equally distant from its middle.

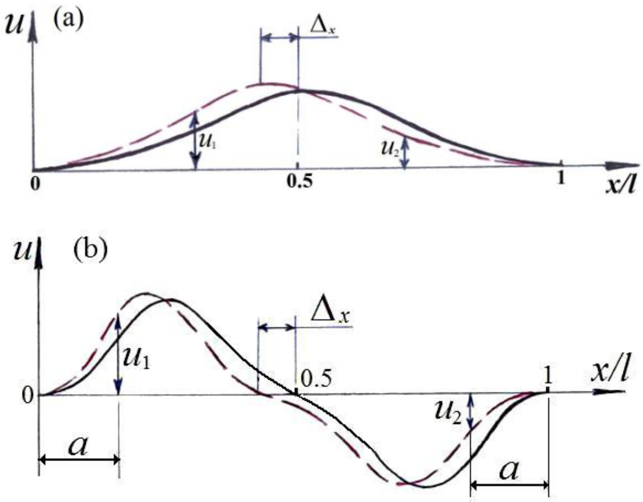

Fig. 4. Modes of beam resonant flexural vibrations: (a) the first resonant mode; (b) the second resonant mode

The most preferable diagnostic sign has been selected taking account of its sensitivity to defect, convenience of measuring, and simplicity of hardware implementation. On the first stage parameter Δ_x was excluded due to its inconvenience for measuring and hardware implementation. Parameters Δu and u_1/u_2 have been compared by their detection sensitivity [1]

$$\eta = \frac{\Delta\Pi/\Pi}{\Delta\lambda/\lambda}, \tag{4}$$

where Π is numerical value of diagnostic sign in initial (undamaged) condition; λ is numerical value of testing parameter (e.g. stiffness k) in initial condition; $\Delta\Pi$ is absolute change of diagnostic sign which conforms to absolute change of parameter λ on value $\Delta\lambda$.

As the result of research, it has been finally proposed that the vibration testing should be carried out by recording the amplitudes ratio, u_1/u_2, measured on the second flexural mode of beam's resonant vibrations. In this case detection sensitivity is about $\eta = 0.9 - 1.0$, that is sufficiently higher in comparison with the traditional resonant frequency methods [2].

Damage detection by the proposed linear vibration method can be carried out in the following operational procedure. At first, the second mode of beam's resonant flexural vibrations is excited. After that, the ratio of amplitudes u_1/u_2 in two points of the beam equally distant from its middle is recorded. The presence and the value of defect are evaluated by the measured ratio of amplitudes u_1/u_2 with the aid of calibration curve $q = f(u_1/u_2)$ preliminary constructed.

Insertion of additional nonlinear element into the structure of testing object. For the first time the idea to increase the detection sensitivity by the special transformation of initial linear mathematical model of testing object into the nonlinear one was formulated by S.Tsyfansky [1,5]. In the case considered here, such transformation is made by the insertion of additional nonlinear elastic support into the structure of the system (Fig. 5).

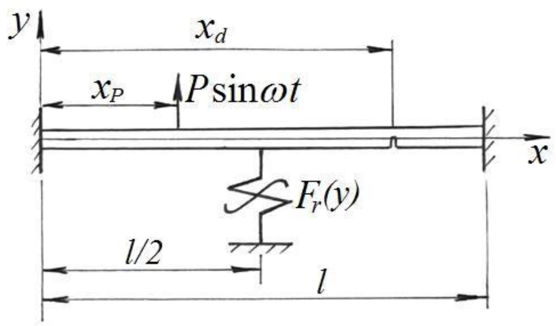

Fig. 5. Dynamic model of the beam interacting with nonlinear elastic support

Forced bending flexural vibrations of the damaged beam interacting with nonlinear elastic support $F_r(y)$ are mathematically described by the following differential equation:

$$EI\frac{\partial^4 y}{\partial x^4}+\mu\frac{\partial^2 y}{\partial t^2}+bEI\frac{\partial^5 y}{\partial t\cdot\partial x^4}=P\sin\omega t\cdot\delta(x-x_P)-F_r(y)\cdot\delta(x-l/2)\,, \qquad (5)$$

where $\delta(x-0.5l)$ is Dirac delta function, $F_r(y)$ is a nonlinear restoring force of the additional elastic support. The sense of other symbols is the same as in equation (1).

As before, damage which can appear in the beam is mathematically described by the equation (2), but boundary conditions – by the equation (3). The variant of nonlinear elastic support with characteristic $F_r(y)$ of preload type is considered:

$$F_r(y)=ky+F_0\,\mathrm{sign}\,y\,. \qquad (6)$$

The problem considered in this paper was solved assuming the parameters of equations (5) – (6) according to the following conditions: $b\omega_1$ = 0.00785; $\omega_1 l^2\sqrt{\mu/(EI)}$ = 7.1; $Pl^2/(EI)$ = 1.5; x_d/l = 0.75; $x_{\mathbf{P}}/l$ = 0.5; $F_0 l^2/(EI)$ = 1; $kl^3/(EI)$ = 20. Dynamics analysis has been made by modeling on the specialized analogue-digital computer system [6,7] as well by numerical simulation with program ANSYS. Modeling with ANSYS was performed using two types of elements: BEAM188 for beam and COMBIN39 for nonlinear elastic support.

Amplitude-frequency characteristics for the beam interacting with nonlinear support are shown in Fig. 6. These AFC are received by numerical simulation with ANSYS. As it is seen, damage in a beam has an influence on spectrum of forced vibrations. Specifically, additional low-frequency harmonic component (~190 Hz) appears in vibration spectrum.

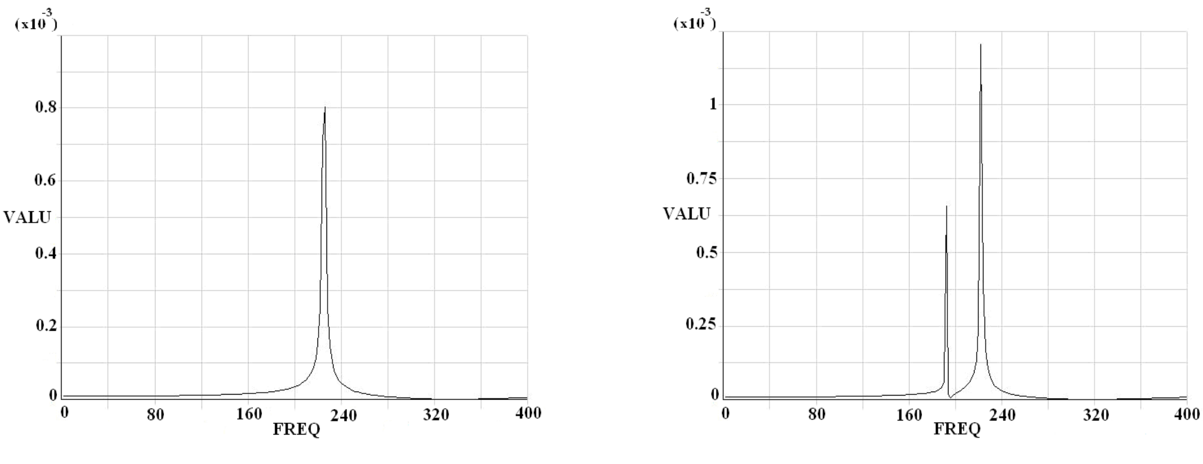

Fig. 6. Amplitude-frequency characteristic for the beam interacting with nonlinear support: left – beam without damage; right – beam with damage

Influence of additional elastic support on beam's resonant oscillations by the second flexural mode (frequency ν_{r2}) has been analyzed in more details. Possible distortions of the second flexural mode due to appearance of defect are shown in Fig. 7. As additional information, the time responses $u=f(\tau)$ and the spectrograms for points B and B' are presented.

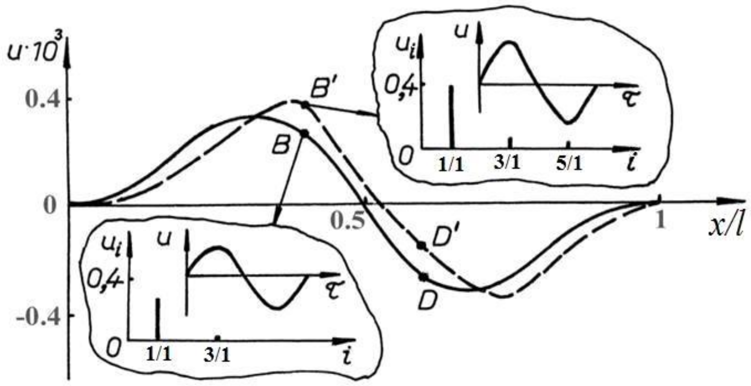

Fig. 7. Distortion of the second flexural mode due to appearance of fault:
full line – undamaged beam (q = 0); dotted line – beam with damage (q = 0.2)

If there is no damage in the beam, the nodal point of the second flexural mode coincide with the co-ordinate x/l = 0.5 of location of additional nonlinear elastic support. In this case, nonlinearity of elastic support practically does not influence the resonant oscillations of the second mode. Therefore bending oscillations of undamaged beam are close to monoharmonic ones (relative value of the 3rd harmonic component $u_{3/1}$ in Fourier spectrum makes up only 0.007).

With the appearance of damage, the original symmetry of beam's resonant mode is disturbed, and the nodal point is moved in the direction of defect. Therefore nonlinear elastic support, placed in section x/l = 0.5, begins active interactions with the vibrating beam not only at the first resonant mode, but also at the second one. In consequence of this, nonlinear properties of the system on the second resonant regime are sufficiently intensified, and as the result the corresponding change of vibration spectrum occurs. For example, at q = 0.2 (see Fig. 7) due to the influence of nonlinear elastic support spectral ratio $u_{3/1}/u_{1/1}$ is increased more than ten times (in comparison with the case of q = 0) and reaches $u_{3/1}/u_{1/1}$ = 0.08 (here $u_{1/1}$ is an amplitude of main harmonic component). Besides, the sensitivity η to defect of the proposed spectral diagnostic sign $u_{3/1}/u_{1/1}$ is about 1.4 – 1.5 times higher in comparison with the above considered linear approach.

Damage detection by the proposed nonlinear vibration method can be carried out in the following operational procedure. At first, the middle section of the testing beam must be connected with an additional nonlinear elastic support. After that, the second mode of beam's resonant flexural vibrations is excited, and spectral ratio $u_{3/1}/u_{1/1}$ of these vibrations is recorded. The presence and size of damage are evaluated by the measured value of vibration parameter $u_{3/1}/u_{1/1}$ with the aid of calibration curve $q = f(u_{3/1}/u_{1/1})$ preliminary constructed.

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ВИБРОСТЕНД ДЛЯ ИССЛЕДОВАНИЯ МИКРОТОКОВ МДП-ТРАНЗИСТОРОВ В НИЗКОЧАСТОТНОМ ДИАПАЗОНЕ

В работе представлена структурная схема разработанного вибростенда и приведены результаты исследования усилительных свойств МДП-транзисторов в режиме микротоков в низкочастотном диапазоне.

Для периметрального контроля трубопроводов можно использовать различные методы регистрации присутствия или движения, использующие охранные датчики на