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Forced vibrations of high-rise buildings

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Abstract. Studies were conducted to analyze the behavior of a high-rise structure on various kinematic effects, taking into account the real geometry, dissipative and nonlinear properties of the structure material. A generalized approach was developed for the dynamic calculation of high-rise structures, and the frequency response characteristics at various points of the structure were built. It was found that nonlinear properties are manifested when the impact can cause significant strain in the structure. This applies not only to the magnitude of the impact intensity, but also to its frequency content. If a non-linearly elastic strain of the material is manifested in the structure, then this leads to a decrease in the amplitudes of points displacements and to an increase in oscillation period compared to a linearly elastic structure under similar kinematic effects.

1. Introduction

In the dynamics of structures, studies of dynamic characteristics (i.e., natural frequencies, forms and decrements of vibrations) of a structure occupy a special place. The dynamic characteristics are a passport of the structure and allow us to judge the dynamic properties of the structure as a whole, without even examining its behavior under various influences. Determination of natural frequencies and vibration modes for elastic structures is an independent and rather difficult task. The determination of structure dynamic characteristics is complicated by an order of magnitude when dissipative properties of the material are taken into account.

High-rise structures include high-rise monolithic reinforced concrete structures, high-rise chimney stacks and ventilation pipes of thermoelectric and nuclear power plants, cooling towers of thermoelectric power plants and nuclear power plants and protective shells of nuclear power plants . Due to their design features and geometric dimensions, they are unique structures. Today, a large number of different high-rise structures are operated and built all over the world, including high-rise chimney stacks; the height of some of them reaches 150 m - 600 m [1-2]. If for the pipes of a height of 50 m the ratio of wall thickness δ to the radius R of its middle surface at the base is $\delta/R=1/5\div1/7$, then for the pipes of a height of 250 - 300 m it is $\delta/R = 1/12\div1/15$, and for the pipes of a height of H=420 m, the ratio δ/R is $\delta/R\approx1/23$. With increase in height H and radius R, the wall thickness of the pipe δ grows slowly [1-2]. Along with this, the pipe radius, thickness and slope of the cone change along its height, gradually moving from a conical section to a cylindrical one.

The design and construction of various high-rise structures require ensuring their reliable operation under various dynamic influences. This, in turn, dictates the conditions for limiting and reducing the

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level of structure vibrations from harmful dynamic effects. To date, various methods and means of dealing with unacceptable vibrations of structures are known, in particular, the change in rigidity and inertial parameters of structures in order to detune from resonances, the increase in damping properties by using materials and structures with high absorption capacity, for example, special coatings, the use of vibration isolation and various types of vibration dampers [8-10]. Each of the mentioned methods has its own rational range of application.

In acting building codes, the dynamic calculation model of axisymmetric structures (high-rise pipes, cooling towers, etc.) is adopted in the form of an elastic cantilever with distributed or concentrated mass, which does not account for a number of factors — real geometry, design features, spatial nature of the structure, dissipative properties of the structure material and others that directly affect the sought for values of dynamic characteristics and the stress-strain state of structures under various impacts. As a result, the found values of the sought for quantities noticeably differ from the real ones.

Studies of the natural vibrations of high axisymmetric structures (smokestacks) showed that despite the differences in nature of the higher modes of vibrations of the structures under consideration, the fundamental modes (bending, longitudinal and torsional) and the corresponding eigenfrequencies are independent of the dimension of the selected calculation model. Therefore, the studies of structure dynamic behavior in an unsteady-state mode, taking into account elastic, viscoelastic and nonlinear strain of the material, can be carried out in the framework of a one-dimensional model. The simplicity of the calculation model allows us to study in detail the effect of various laws of change under the impact and various laws of material strain on the dynamic behavior of structures. Dissipative processes in the materials are taken into account by the Voigt, Maxwell, and Kelvin models, but they do not always agree with experimental data, therefore, to eliminate this, the models with hysteresis absorption or hereditary viscoelastic Boltzmann-Volterra models are used, although their implementation is rather difficult and available experimental data are scarce [3–7].

The practice of modern construction in seismic regions requires studying the dynamic behavior of structures and improving the methodology for their calculation, taking into account not only geometrical features of structures, but nonlinear and dissipative properties of the structure material as well. Practical methods of calculation are usually based on the dynamic analysis of structures as the linearly elastic systems. However, instrumental data and the results of engineering analysis on the nature of structures operation under strong earthquakes indicate that the structure rigidity does not always remain constant. Therefore, the parameters of actual reaction of the structures should be determined only in non-linear analysis, which allows developing more reasonable methods of design and construction, increase the efficiency of structures while maintaining the required level of reliability. To date, a number of publications are known devoted to the study of the dynamics of structures, in which calculation methods, research results are highlighted, and an attempt is made to consider the dissipative properties of the material.

In [15–17], the aerodynamic wind damping coefficients of modified square high-rise buildings were studied on the basis of estimated results. The effect of aerodynamically modified cross sections on the coefficient of aerodynamic damping under across-wind was investigated. The aerodynamic wind damping coefficients in high-rise buildings were determined using the eigensystem realization algorithm (ERA) method in combination with the random decrement technique (RDT).

Various basic and secondary modifications of the external forms of high-rise buildings and their advantages compared to conventional forms (square, triangular, round) were considered in [18–20]. The influence of wind-induced aerodynamic forces and moments was studied in buildings with a changed height; the space-time characteristics of vibrations were determined

In [21–25], the eigenfrequencies of a cylindrical shell under various boundary conditions were studied, the influence of uniform external pressure and symmetrical boundary conditions on the eigenfrequencies of homogeneous and multilayer isotropic cylindrical shells was studied, and the vibrational process of rigid composite cylindrical shells was investigated taking into account the bending behavior of the ribs.

In [26-27], the determination of the dynamic characteristics of high-rise monolithic reinforced concrete structures was investigated and the results obtained were recommended for certification of buildings.

Parametric vibrations of viscoelastic orthotropic plates of variable thickness were studied in [28–31], under external load, taking into account the influence of geometric nonlinearity, viscoelastic properties of the material, and other physicomechanical and geometrical design parameters on the dynamic instability region.

In [32–35], vibrations of high buildings caused by wind and tornado waves were studied to assess the aeroelastic effects of high buildings using the wind tunnel tests. The aerodynamic damping coefficient and aerodynamic stiffness were determined by analyzing the aeroelastic force acting on the oscillating model. For a 347-meter-high building, the effect of aeroelastic parameters on wind-induced responses and equivalent static wind loads was analyzed. The results showed that during a return period of 100 years, aerodynamic damping was positive and aerodynamic stiffness was negative.

The current stage in the development of the theory of seismic resistance involves an account for nonlinear behavior of the structure material under dynamic loads. Any real objects possess non-linear properties to a different extent but in some cases the influence of non-linearity is negligible, in such cases linear models and the corresponding linear theories are used. The question of estimating the nonlinearity for real structures remains open due to a number of mathematical problems that arise when solving the problem and in the absence of nonlinearity parameters of the material. This dictates the relevance of the studies presented in this paper, where the dynamic behavior of a high-rise structure is studied taking into account the linear, nonlinear strain and energy dissipation caused by internal friction in the building material under various kinematic effects.

2. Methods

Unsteady-state forced vibrations of a high axisymmetric structure are considered; the structure is represented by a one-dimensional model - a viscoelastic beam of annular cross section with a variable slope of the generatrix and a variable thickness. The lower end of the beam ($z = 0$) is rigidly fixed and the kinematic effect $w_0(t)$ is set on it; the upper end ($z = L$) is free. The beam material is a nonlinearly viscoelastic one. Bending unsteady-state forced vibrations of points located at different levels of a structure under set kinematic effect are to be determined.

The mathematical statement of the problem includes the variational equation of the principle of virtual displacements, according to which the sum of work of all active forces, including inertia forces, on a virtual displacement δw, satisfying geometrical boundary conditions is zero

$$\delta A_M + \delta A_u + \delta A_P = 0 \tag{1}$$

Here $\delta A_м$, $\delta A_и$, δA_P – are the virtual work of the bending moment, inertial forces and external forces, respectively, calculated by the formulas:

$$\delta A_M = -\int_0^L M(z)\delta\left(\frac{\partial^2 w}{\partial z^2}\right)dz\,,\ \delta A_u = -\rho\int_0^L F(z)\left(\frac{\partial^2 w}{\partial t^2}\right)\delta w dz\,,\ \delta A_P = \int_0^L P(z,t)\delta w dz \tag{2}$$

where ρ - is the beam material density, L – the beam length, $w(z)$ - the beam deflection, $M(z)$ - the bending moment; $F(z)$ - the cross-sectional area; $P(z, t)$ – the external dynamic forces.

The kinematic boundary condition at the base is

$$z = 0 : w(z,t) = w_0(t) \tag{3}$$

where $w_0(t)$ - is the known time function.

Initial conditions are

$$w(z,0) = u_0, \frac{\partial w(z,0)}{\partial t} = \dot{u}_0 \tag{4}$$

where u_0, $\dot{u}_0$ are the given constants.

To describe the relationship between the stress σ_z and the strain ε_z the nonlinear theory of viscoelasticity [4] is used, which has the form

$$\sigma_z = E\left\{\left[\varepsilon_z(t) - \int_0^t R_1(t-\tau)\varepsilon_z(\tau)d\tau\right] - \gamma\left[\varepsilon_z^3(t) - \int_0^t R_2(t-\tau)\varepsilon_z^3(\tau)d\tau\right]\right\} \quad (5)$$

where E - is the instantaneous modulus of elasticity of the material; R_1, R_2 are the relaxation kernels; γ=const>0 is the non-linearity coefficient, depending on the material of the beam.

The dependence between the deflection *w* and the strain ε_z is taken in the form

$$\varepsilon_z = -x\frac{\partial^2 w}{\partial z^2} \ , \quad (6)$$

and the relationship between bending moment M_z and stress σ_z is

$$M_z = \int_F x\sigma_z dF \quad (7)$$

The problem of unsteady-state nonlinear forced vibrations of a beam consists in the following: for a given function $w_0(t)$ under initial conditions $u_0, \dot{u}_0$ - to find the deflection *w(z, t),* strain $\varepsilon_z(z, t)$,), stress $\sigma_z(z, t)$ and bending moment $M_z(z, t)$, satisfying equations (1), (2), (5) - (7) and conditions (3), (4) for any possible δw.

To reduce the variational problem posed above to a system of resolving equations, the finite element method is used [11], where a one-dimensional element is selected as the finite element, taken in the form of a truncated cone that works on bending with four degrees of freedom (figure 1).

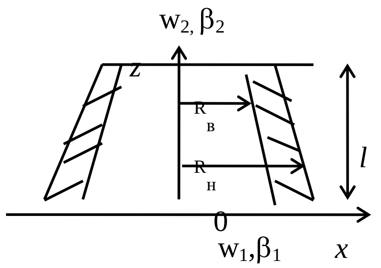

Figure 1. A finite element used

For the displacement function *w* inside the *e*-th element, the cubic approximation is used:

$$w = \alpha_1 + \alpha_2 z + \alpha_3 z^2 + \alpha_4 z^3 \quad (8)$$

then for the first and second derivatives the following expressions are obtained

$$\frac{\partial w}{\partial z} = \alpha_2 + 2\alpha_3 z + 3\alpha_4 z^2 ; \ \frac{\partial^2 w}{\partial z^2} = 2\alpha_3 + 6\alpha_4 z \quad (9)$$

The dependence of the nodal displacements and rotation angles of the *e*-th finite element $\{w_i\}$ on the vector of arbitrary constants $\{\alpha_i\}$ in matrix form is written as

$$\{w_i\} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 1 & l & l^2 & l^3 \\ 0 & 1 & 2l & 3l^2 \end{bmatrix} \{\alpha_i\} \quad (10)$$

where $\{w_i\}^T = \{w_i, \beta_1, w_2, \beta_2\}$, $\beta_1 = \frac{\partial w}{\partial z}\Big|_{z=0}$, $\beta_2 = \frac{\partial w}{\partial z}\Big|_{z=l}$.

Hereinafter, the following notation is used: {} - vector, [] - matrix, ^{T} - transposition operation.

The transformation inverse to (10), i.e. the matrix dependence of $\{\alpha_i\}$ on $\{w_i\}$ is expressed as

$$\{\alpha_i\}=\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ -\frac{3}{l^2} & -\frac{2}{l} & \frac{3}{l^2} & -\frac{1}{l} \\ \frac{2}{l^3} & \frac{1}{l^2} & -\frac{2}{l^3} & \frac{1}{l^2} \end{bmatrix}\{w_i\}, \text{ i.e. } \{\alpha_i\}=[A]\{w_i\} \tag{11}$$

Using the indicated transformations (11), we express the displacement function (8) and its derivatives in a matrix form in terms of nodal displacements $\{w_i\}$

$$w=\left[1\cdot\cdot z\cdot\cdot z^2\cdot\cdot z^3\right]\{\alpha_i\}=\left[1\cdot\cdot z\cdot\cdot z^2\cdot\cdot z^3\right][A]\{w_i\} \tag{12}$$

$$\frac{\partial w}{\partial z}=\left[0\cdot\cdot 1\cdot\cdot 2z\cdot\cdot 3z^2\right][A]\{w_i\} \tag{13}$$

$$\frac{\partial^2 w}{\partial z^2}=[0\cdot\cdot 0\cdot\cdot 2\cdot\cdot 6z][A]\{w_i\}= \tag{14}$$

We introduce the matrix [B]

$$[B]=[0\cdot\cdot 0\cdot\cdot 2\cdot\cdot 6z][A][\ -\frac{6}{l^2}+\frac{12z}{l^3};-\frac{4}{l}+\frac{6z}{l^2};\frac{6}{l^2}-\frac{12z}{l^3};-\frac{2}{l}+\frac{6z}{l^2}\ ],$$

then

$$\frac{\partial^2 w}{\partial z^2}=[B]\{w_i\} \tag{15}$$

Substituting expression (5-7) in (2), we obtain the virtual work of the bending moment for the *e*-th element

$$\delta A_M^e=\int_0^l\left[\frac{\partial^2 w}{\partial z^2}\delta\left(\frac{\partial^2 w}{\partial z^2}\right)\int_F x^2 dF\right]dz-E\int_0^t\left\{R_1(t-\tau)\int_0^l\left[\frac{\partial^2 w}{\partial z^2}\delta\left(\frac{\partial^2 w}{\partial z}\right)\int_F x^2 dF\right]dz\right\}d\tau\ -$$

$$-E\gamma\int_0^l\left[\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z^2}\right)\int_F x^4 dF\right]dz+E\gamma\int_0^t\left\{R_2(t-\tau)\int_0^l\left[\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z}\right)\int_F x^4 dF\right]dz\right\}d\tau \tag{16}$$

Substitution of (15) into (16) and integration over the cross-sectional area leads each term of expression (16) to the following form:

the first term

$$\delta\{w_i\}^T E\int_0^l J^e(z)[B]^T[B]dz\{w_i\}=\delta\{w_i\}^T\left[K^e\right]\{w_i\} \tag{17}$$

where

$J^e(z)=\frac{\pi}{4}\left[(R_H(z))^4-(R_B(z))^4\right]$ - is the moment of inertia of the cross section of the *e*-th element; R_H, R_B are the outer and inner radii of the element, respectively;

$\left[K^e\right]=E\int_0^l J^e(z)[B]^T[B]dz$ - is the stiffness matrix of the *e*-th element.

the second term

$$E\int_0^t\left\{R_1(t-\tau)\int_0^l\left[\frac{\partial^2 w}{\partial z^2}\delta\left(\frac{\partial^2 w}{\partial z^2}\right)\int_F x^2 dF\right]dz\right\}d\tau=\delta\{w_i\}^T E\int_0^t\{R_1(t-\tau)\int_0^l J^e(z)[B]^T[B]dz\}d\tau\{w_i\}=$$

$$=\delta\{w_i\}^T\int_0^t R_1(t-\tau)\left[K^e\right]\{w_i\}d\tau \qquad (18)$$

the third term

$$E\gamma\int_0^l\left[\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z^2}\right)\int_F x^4 dF\right]dz=\delta\{w_i\}^T E\gamma\int_0^l J_1^e(z)[B]^T[B]\{w_i\}[B]\{w_i\}[B]\{w_i\}dz \qquad (19)$$

Here $J^e(z)=\frac{\pi}{8}\left[(R_H(z))^6-(R_B(z))^6\right]$.

Expanding the expression under the integral sign in (19):

$$\left\{V^e\right\}=J_1^e(z)[\ -\frac{6}{l^2}+\frac{12z}{l^3};-\frac{4}{l}+\frac{6z}{l^2};\frac{6}{l^2}-\frac{12z}{l^3};-\frac{2}{l}+\frac{6z}{l^2}\ ]^T[\ (-\frac{6}{l^2}+\frac{12z}{l^3})w_1+(-\frac{4}{l}+\frac{6z}{l^2})\varphi_1+$$
$$+(\frac{6}{l^2}-\frac{12z}{l^3})w_2+(-\frac{2}{l}+\frac{6z}{l^2})\varphi_2\ ]^3$$

(20)

we see that it is a vector whose coordinates are cubic polynomials from nodal displacements.

As a result of integration over the length of the element, the third term (19) is

$$E\gamma\int_0^l\left[\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z^2}\right)\int_F x^4 dF\right]dz=\delta\{w_i\}^T E\gamma\left\{V^e\right\} \qquad (21)$$

where the index "e" indicates that the vector {Ve} is defined for the *e*-th element.

The fourth term

$$E\gamma\int_0^t\left\{R_2(t-\tau)\int_0^l\left[\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z}\right)\int_F x^4 dF\right]dz\right\}d\tau=\delta\{w_i\}^T E\gamma\int_0^t R_2(t-\tau)\int_0^l J_1^e(z)[B]^T[B]\{w_i\}[B]\{w_i\}dzd\tau$$

(22)

Considering (20), we get

$$E\gamma\int_F x^5\int_0^l R_2(t-\tau)\left(\frac{\partial^2 w}{\partial z^2}\right)^3\delta\left(\frac{\partial^2 w}{\partial z^2}\right)dzdF=\delta\{w_i\}^T E\gamma\int_0^t R_2(t-\tau)d\tau\left\{V^e\right\} \qquad (23)$$

The use of the finite element method procedure leads the variational problem (1) and (3) to a nonlinear system of integro-differential equations, which has the following matrix form:

$$[M]\{\ddot{w}(t)\}+[K]\{w(t)\}=\{P(t)\}-\int_0^t R_1(t-\tau)[K]\{w(t)d\tau+E\gamma\{V(t)\}\}-E\gamma\int_0^t R_2(t-\tau)d\tau\{V(t)\}$$

(24)

Here [M], [K] are the matrices of mass and rigidity of the entire structure; {w} is the displacement vector of all the nodal points of the structure; {V} is a vector whose coordinates are determined by cubic polynomials of system displacements, {P} is a vector of external influences.

This equation is solved by the Newmark method [12]. Equation (24) at given initial conditions (4) is solved by direct integration using a numerical step-by-step procedure. We used the Newmark method to solve the system of equations (24), based on independent expansions of $w(t_i+\tau)$ and its derivative into the series in powers τ, while holding the terms containing the third derivative w_i. The coefficients for the residual terms α and β are selected from the condition for ensuring the unconditional convergence of the integration process:

$$w(t_i+\tau)=w_i+\tau\dot{w}_i+\frac{\tau^2}{2}\ddot{w}_i+\alpha\tau^3\dddot{w}_i$$

$$\dot{w}(t_i+\tau)=\dot{w}_i+\tau\ddot{w}_i+\beta\tau^2\dddot{w}_i \tag{25}$$

Substituting $\dddot{w}_i=\frac{\ddot{w}_{i+1}-\ddot{w}_i}{\tau}$, expressions for displacements and velocities (25) are written as

$$w_{i+1}=w_i+\tau\dot{w}_i+\frac{\tau^2}{2}\ddot{w}_i+\alpha\tau^3(\ddot{w}_{i+1}-\ddot{w}_i) \tag{26}$$

$$\dot{w}_{i+1}=w_i+\tau\ddot{w}_i+\beta\tau^2(\ddot{w}_{i+1}-\ddot{w}_i) \tag{27}$$

Then the acceleration obtained form (26)

$$\ddot{w}_{i+1}=\frac{1}{\alpha\tau^2}(w_{i+1}-w_i)-\frac{1}{\alpha\tau}\dot{w}_i+\left(1-\frac{1}{2\alpha}\right)\ddot{w}_i \tag{28}$$

is substituted into the velocity expression (27)

$$\dot{w}_{i+1}=\frac{\beta}{\alpha\tau}(w_{i+1}-w_i)+\left(1-\frac{\beta}{\alpha}\right)w_i+\frac{\tau}{2}\left(2-\frac{\beta}{\alpha}\right)\ddot{w}_i \tag{29}$$

To find a solution w_{i+1} for time t_{i+1}, the general equation of motion is written as follows:

$$[M]\ddot{w}_{i+1}+[C]\dot{w}_{i+1}+[K]w_{i+1}=\{P_{i+1}\} \tag{30}$$

After substituting expressions for accelerations (28) and velocity (29) into (30) an algebraic system of equations is obtained

$$[A]\{w_{i+1}\}=\{R_{i+1}\} \tag{31}$$

Where

$$[A]=[K]+\frac{1}{\alpha\tau^2}[M]$$

$$\{R_{i+1}\}=\{P_{i+1}\}+[M]\left(\frac{1}{\alpha\tau^2}\{w_i\}+\frac{1}{\alpha\tau}\{\dot{w}_i\}+\left(\frac{1}{2\alpha}-1\right)\{\ddot{w}_i\}\right)+\{W_i\} \tag{32}$$

where

$$\{W_i\}=\int_0^t R_1(t-\tau)[K]\{w_i\}d\tau+E\gamma\{V_i\}-E\gamma\int_0^t R_2(t-\tau)d\tau\{V_i\} \tag{33}$$

To solve the resulting system of equations (31), it is necessary to specify at the initial moment the values of displacements $\{w_0\}$, velocity $\{\dot{w}_0\}$ and accelerations $\{\ddot{w}_0\}$. Usually $\{\ddot{w}_0\}=0$ is taken. The Newmark method is unconditionally stable if $\beta\geq 0.5$, $\alpha\geq 0{,}25(\beta+0{,}5)^2$.

3. Results and discussion

The dynamic behavior of a high-rise smokestack of the Novo-Angren hydro-electric power plant is considered; its actual dimensions are presented in [14], the smokestack is rigidly pinched at the base. The problem of forced unsteady-state vibrations of this structure is solved under various laws of kinematic disturbance of the lower base. The material of the structure is assumed to be a nonlinearly elastic one.

In this case, equation (24) takes the following form

$$[M]\{\ddot{w}(t)\}+[K]\{w(t)\}=\{P(t)\}+E\gamma\{V(t)\} \quad (34)$$

under corresponding initial conditions (4). For the problems considered in this section, the initial conditions are assumed to be homogeneous, i.e.

$$w(z,0)=0, \qquad \frac{\partial w(z,0)}{\partial t}=0. \quad (35)$$

The task is to determine the displacements of the points of the structure at different points in time. The obtained linear system of differential equations (34) with initial conditions (35) is solved by the Newmark method. Using the developed techniques and composite programs for an PC-IBM, some problems were solved and the dynamic behavior of high-rise structures was studied taking into account the properties of the nonlinear elastic properties of the material. In all the values of the coefficient γ considered below, equal to 120,000.

3.1. Resonance mode.

The lower base of the pipe is subjected to a kinematic effect of the type

$$z=0: \qquad \ddot{u}_0=\sin(pt) \quad (36)$$

The frequency of impact p was chosen close to the eigenfrequency ω_0 of the pipe bending vibrations.

The obtained horizontal displacements of various points of the pipe are shown in Fig. 2. Figure 2b shows the diagram of nonlinear deformation σ~ε for the upper part of the pipe resulting from this action. Here, for comparison, asterisks show the linear strain diagram σ~ε obtained by the same action in the elastic case.

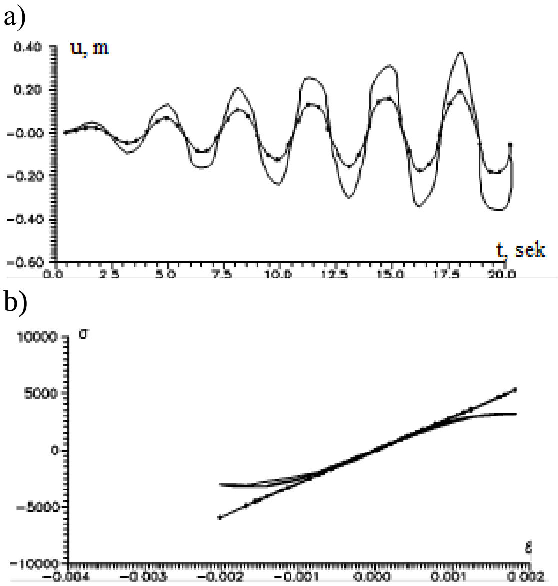

Figure 2. Forced vibrations of various points of the pipe taking into account the nonlinear elastic properties of the material at resonance: a) oscillations of the pipe points: -*-*-*-* - at a height of z = 200m; ____ - at a height of z = 325m. b) stress-deformation diagram: ______ nonlinearly elastic material; -*-*-*- linearly elastic material.

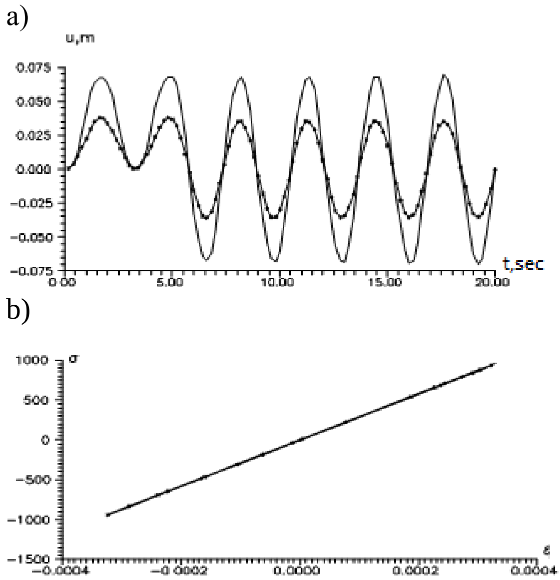

Figure 3. Forced vibrations of various points of the pipe taking into account the nonlinear elastic properties of the material at a long pulse (T=5 sec): a) oscillations of the pipe points: -*-*-*-* - at a height of z = 200m; ____ - at a height of z = 325m. b) stress-deformation diagram: ______ nonlinearly elastic material; -*-*-*- linearly elastic material.

As can be seen from the results (Fig.2.a), the displacements of the pipe points are oscillatory in nature with the amplitude increasing with time. The deformation diagram (Fig.2.b) in this case is pronounced non-linear.

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3.2. Long-term impulse effect.

The kinematic effect is set on the lower base of the pipe according to the law:

$$z=0: \qquad \ddot{u}_0 = \begin{cases} 1 & t \le 5sek \\ 0 & t > 5sek \end{cases} \tag{37}$$

Here, the pulse duration exceeds the main period of natural vibrations. The nature of the impact and the movements of various points of the pipe obtained using the developed program are shown in Fig.3a, the stress-deformation diagram in Fig.3b.

3.3.Sinusoidal - damping effect.

Damping kinematic effects can occur when the structure is loaded with seismic, explosive and other long-term unsteady impacts. Therefore, the kinematic effect set on the lower base of the pipe is considered according to the law of a damping sinusoid:

$$z=0: \quad \ddot{u}_0 = A\sin(pt)\exp(-\gamma t) \tag{38}$$

Here A is the amplitude of the impact; $p \approx \omega_0$ - is the impact frequency equal to the first frequency of the bending eigenmodes of the pipe; γ =0.1 is the coefficient characterizing the impact attenuation over time. The amplitude of the impact A is chosen so that its maximum is 0.1. This is done to be able to compare the displacements obtained in all cases and to analyze the effect of impact nature, but not its intensity.

The amplitude of action thus obtained is equal to

$$A = \frac{\sqrt{p^2+\gamma^2}\exp\left(\frac{\gamma}{p} arctg\frac{p}{\gamma}\right)}{p} \tag{39}$$

The obtained horizontal displacements of the middle and upper points of the high-altitude pipe under kinematic effects are shown in Fig. 4a. Figure 4b compares the displacements of the point z = 325 m obtained with a linearly elastic solution and taking into account the nonlinear deformation of the material. Figure 4c shows the diagram of nonlinear deformation $\sigma \sim \varepsilon$ in the upper part of the pipe. Here, asterisks, as in the previous example, show a linear diagram obtained in the elastic case.

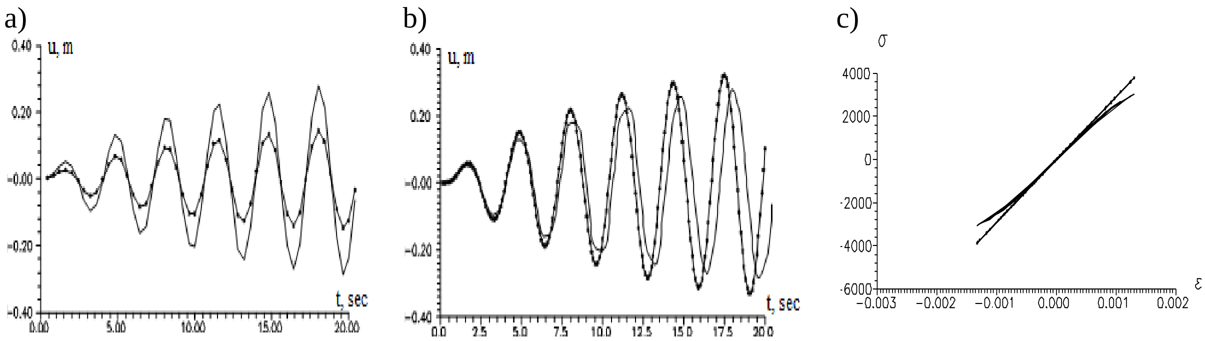

Figure 4. Forced vibrations of various points of the pipe taking into account the nonlinear elastic properties of the material under damping effect: a) oscillations of the pipe points: -*-*-*-* - at a height of *z* = 200m; ____ - at a height of *z* = 325m. b) oscillations of the point *z* = 325m:_____ nonlinear solution; -*-*-*- linear solution; c) stress-deformation diagram: _____ nonlinearly elastic material; -*-*-*- linearly elastic material.

As can be seen from Fig.3b, the displacements obtained in this case are also somewhat inferior to the displacements in the perfectly elastic case under the same action. The oscillation period in the nonlinear version increases. To a large extent, the noted differences are manifested with an increase in displacements and deformations over time. The deformation diagram, as in the previous example, is nonlinear.

4. Conclusions

Based on the above studies, the following conclusions can be drawn:

1.A generalized approach to solving problems of unsteady-state forced vibrations of a high-rise structure taking into account the real geometry, dissipative and nonlinear properties of the structure material was developed.

2.The results obtained in a linearly elastic statement show that an account for physical nonlinearity leads to a decrease not only in the amplitudes of displacements, but also to an increase in the period of oscillations of a nonlinear elastic structure.

3.It was determined that in order to obtain a nonlinear effect, not only the intensity is important, but also the frequency spectrum of the impact, causing sufficient strains to manifest the nonlinearity.

4.If nonlinear elastic strain of the material is manifested in the structure, this leads to a decrease in the amplitudes of point displacements and to an increase in the period of oscillations under different kinematic influences.

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