# On numerical simulation of vibrations in radio-electronic structures

Cite as: AIP Conference Proceedings **2467**, 060038 (2022); https://doi.org/10.1063/5.0094082
Published Online: 22 June 2022

**G. G. Yunusov, N. Q. Esanov, Sh. N. Almuratov, et al.**

# On Numerical Simulation of Vibrations in Radio-Electronic Structures

G. G. Yunusov[1,a)], N. Q. Esanov[2,b)], Sh. N. Almuratov[3,c)], Sh. Ablokulov[3,d)] and R. Sobirov[1,e)]

[1] *Bukhara engineering technological institute, Bukhara, Uzbekistan*
[2] *Bukhara State University, Bukhara, Uzbekistan*
[3] *Tashkent Institute of Chemical Technology, Tashkent, Uzbekistan*

[a)] Corresponding author: ganish2011@gmail.com
[b)] esanov-7373@mail.ru
[c)] al_shavkat@mail.ru
[d)] sh.ablokulov@bk.ru
[e)] rano.sobirova.1988@mail.ru

**Abstract.** In this article, from a scientific point of view, the issues of resistance to mechanical loads acting on on-board radio-electronic devices equipment (R.E.E.) are considered, where the necessary reduction of the impact of vibration loads is proposed to ensure resistance to mechanical influences. It is also said that in general, there are various means of protection against vibrations of the R.E.E., which include dampers, springs, gaskets, shock absorbers. To achieve this goal, you need to solve the following tasks: analyze the vibration of the R.E.E. unit (with and without attached masses) from vibration effects; to construct a mathematical model of the influence of the R.E.E.) parameters on the resonant state of the mechanical system, as well as on the magnitude of the oscillation amplitude. The aim of the work is to reduce the vibration of the radio-electronic unit and to develop a methodology for the influence of the attached mass, as well as changes in various geometric parameters on the resonant frequency and the attenuation decrements of the mechanical system. In solving the problem of R.E.E. from external vibration exposure at resonant frequencies, finite difference methods and the Godunov orthogonal run method were used. An algorithm for determining the resonant frequency and the displacement amplitude of the mechanical system under consideration is proposed. The application of the proposed mathematical model allows to reduce the total vibration loads of the R.E.E.to 25%.

## INTRODUCTION

To reduce the impact of vibration loads, there are various means of protection against R.E.E. vibrations: dampers, springs, gaskets, shock absorbers. The most effective among them are active shock absorbers, in which, in addition to damping elements, there are elements with an additional source of energy, which allows you to change the stiffness of the suspensions, and, thereby, reduce the impact of vibration loads on the R.E.E. Such shock absorbers are designed to reduce the amplitude of vibrations not only at resonant frequencies, but also in the entire required frequency range, which entails the complication of structures due to the introduction of additional vibration measurement tools [1, 2]. Reducing the vibration levels of radio-electronic equipment R.E.E. is an urgent task in the aircraft industry [3,4]. For the calculation of mechanical processes occurring in radio-electronic structures, it is presented in the form of a certain model [5, 6]. When developing a computational model, it is necessary, if possible, to strive to avoid making any fundamental adjustments to the physical phenomena under consideration in order to avoid uncontrolled errors. Vibration loads experienced by devices and apparatuses cause mechanical stresses in their elements. To protect the equipment from dynamic impacts, various dampers and damping elements are widely used [7, 8]. According to the nature of the application of external loads, vibration isolation of R.E.E. is conditionally divided into active and passive [9, 10]. If the object itself is a source of vibration, then it is necessary to isolate it from the reference base [11, 12].

*Proceedings of the II International Conference on Advances in Materials, Systems and Technologies*
AIP Conf. Proc. 2467, 060038-1–060038-10; https://doi.org/10.1063/5.0094082
Published by AIP Publishing. 978-0-7354-4349-5/$30.00

Electronic equipment is often modeled as a monolithic block. These include blocks in which the gaps between the radio components are filled with compound, foam, rubber, etc. In such blocks there are no voids in the model – the grid for them is three-dimensional (Fig. 1) [13, 14]. In this paper, we consider the oscillations of a viscoelastic spatial block under the influence of periodic loads.

## MATERIALS AND METHODS

### Problem Statement and Methods for Solving the Problem

The block and the radio-electronic equipment attached to its mass is a rectangular multilayer parallelepiped (or plate) with concentrated masses. The system of differential equations is obtained using the principle of possible displacements [15]:

$$\delta A_F + \delta A_I = 0. \tag{1}$$

Where $\delta A_F$ - variations of internal stresses

$$\delta A_F = -\delta \Pi = -\int_V \sigma_{ij} \delta \varepsilon_{ij} dV$$

$\Pi$ – potential energy; $\sigma_{ij}$ - components of the stress tensor, $\varepsilon_{ij}$ - components of the strain tensor, $V$ -volume.
The relationship between stress and strain satisfies the following relations

$$\sigma_{ij} = \tilde{\lambda} \varepsilon_{kk} \delta_{ij} + 2\tilde{\mu}\varepsilon_{ij} \quad (i, j, k = 1,2,3) \tag{2}$$

Where,

$$\tilde{\lambda} f(t) = \lambda_{01}\left[ f(t) - \int_{-\infty}^{t} R_\lambda (t-\tau) f(\tau) d\tau \right],$$
$$\tilde{\mu} f(t) = \mu_{01}\left[ f(t) - \int_{-\infty}^{t} R_\mu (t-\tau) f(\tau) d\tau \right].$$

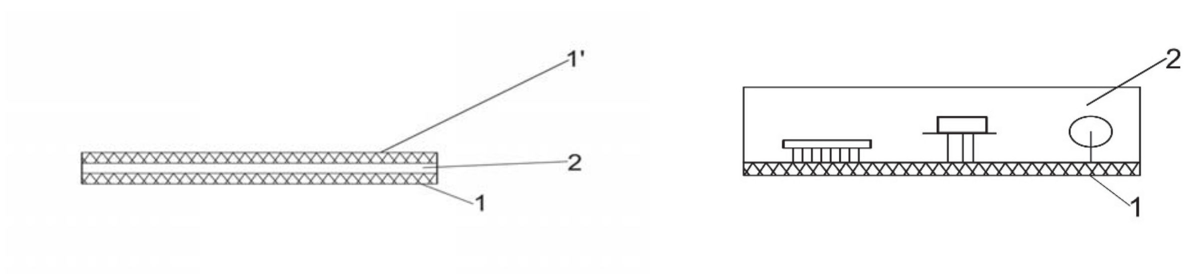
Where $f(t)$ – is an arbitrary function of time; $R_\lambda(t-\tau)$ and $R_\mu(t-\tau)$ the relaxation kernel; $\lambda_{01}, \mu_{01}$ –is the instantaneous modulus of elasticity. As an example of a viscoelastic material, we take the three-parameter relaxation kernel $R_n(t) = A_n e^{-\beta_n t} / t^{1-\alpha_{jn}}$ . Cauchy formulas, or the geometric relation:

$$\varepsilon_{ij}^{(k)} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{3}$$

The initial conditions for t=0:

$$u_{ki}(u_k, \vartheta_k, w_k) = 0, \frac{\partial u_{ki}}{\partial t} = 0 \tag{4}$$

Where $\sigma_{ij}^{(k)}$ - is the stress tensor of the k-th layer; $u_{ki}$ - the displacement vector of the k-th layer; $F_{ki}$ - the density dector of the mass forces of the k-th layer; $\rho_k$ - the density of the k-th cylindrical layer, t-time.




a) b)

**FIGURE 1.** Calculation scheme: a) a three-layer block without attached masses. b) a block with attached masses.

The system of differential equations (1) and relations (2) and (3) are closed with boundary conditions that are put in contact between the blocks $x = z_k$ :

$$\begin{aligned} &\sigma_z^k = \sigma_z^{k+1};\ u_k = u_{k+1};\\ &\tau_{zx}^k = \tau_{zx}^{k+1};\ \ \vartheta_k = \vartheta_{k+1};\\ &\tau_{xy}^k = \tau_{xy}^{k+1};\ w_k = w_{k+1}. \end{aligned} \tag{5}$$

A pulse load of the following type is placed on the outer surface of the plate.

$$z = b:\ \sigma_{zz}^{(k)} = P(x, y, z)\ ;\tau_{xy}^{(k)} = 0\ ;\tau_{zx}^{(k)} = 0. \tag{6}$$

The pressure in the case of a local award is represented by the dependence on coordinates and time in the form.

$$P(b, x, z) = P_b e^{-i\nu t}\xi_p\ , \qquad 0 \le t \le T_0\,, 0 \le z \le l \tag{7}$$

Where $P_b$- is the amplitude value of the external vibration pressure, $T_0$ - is the period of pressure action; $l$ - is the length of the cylinder; $c$- is the length of the cylinder affected by the pulse load and $\xi_p$ is a constant value. The desired functions in the equations of the system (1) – (6) depend on three spatial variables and time.

For proper (or steady - state forced) oscillations of the variation equation in symbolic form, it can be represented as.

$$\delta G(U_{nj}^0(\overline{x}), \omega^2) = 0 \tag{8}$$

We write out a concrete representation of the functional G, for example, for a package of rectangular plates with point constraints:

$$G\left[W_n^0(x,y),\omega^2\right] = -\frac{1}{2}\sum_{n=1}^{N}\bar{D}_n\int_0^{a_n}\int_0^{b_n}\left[(\frac{\partial^2 W_n^0}{\partial x^2}+\frac{\partial^2 W_n^0}{\partial y^2})^2 - 2(1-\nu_n)(\frac{\partial^2 W_n^0}{\partial x^2}\frac{\partial^2 W_n^0}{\partial y^2} - (\frac{\partial^2 W_n^0}{\partial x\partial y})^2)\right]dxdy -$$

$$-\frac{1}{2}\sum_{n=1}^{N-1}\sum_{l=1}^{L_n}\bar{D}_n\left[W_n^0(x_n^l,y_n^l) - W_{n+1}^0(x_n^l,y_n^l)\right]^2 -$$

$$-\frac{1}{2}\sum_{n=1}^{N}\sum_{l'=1}^{L'_n}C_{l'n}(W_n^0)^2(x_n^{l'},y_n^{l'}) + \frac{\omega^2}{2}\sum_{n=1}^{N}\rho_n h_n\int_0^{a_n}\int_0^{b_n}(W_n^0)^2 dxdy +$$

$$+\frac{\omega^2}{2}\sum_{n=1}^{N}\sum_{q=1}^{Q_n}M_{qn}(W_n^0)^2(x_n^q,y_n^q) \quad ,$$

Where $\omega$- is the complex frequency, $h_n, a_n, b_n$ -is the thickness and linear dimensions of the n - the plate, $x_n^q, y_n^q$ - is the coordinates of the *n* - the concentrated mass, $x_n^l, y_n^l$ -is the coordinates of the *l* – the spring (shock absorber), - coordinates of the elastic (viscoelastic) support. If the n - the plate, *l*- the $l'$- the spring, and the y- the support are viscoelastic, then they are represented $\bar{D}_n, \bar{C}_{ln}, \bar{C}_{l'n}$ by the following formulas:

$$\bar{D}_n = D_n f_n(\omega_R), \bar{C}_{l\text{n}} = C_{l\text{n}} f_{l\text{n}}(\omega_R), \bar{C}_{l'n} = C_{l'n} f_{l'\text{n}}(\omega_R)$$

Where $f(\omega_R) = 1 - \Gamma_c(\omega_R) - i\Gamma_s(\omega_R)$ - is a complex function whose numerical coefficients depend on the parameters of the relaxation core of the corresponding viscoelastic elements, $D_n = E_n h_n^3 / (12(1-\nu_n^2))$ $C_{l\text{n}}, C_{l'\text{n}}$- is the generalized instantaneous stiffness of the *n* – the plate, *l*- the $l'$-the shock absorber, and the support, respectively. In the elastic case, where is the generalized stiffness of the n-the plate, *l* - the spring, and the support, respectively. A similar functional can be written for a system of rotation shells. The components of the displacement vector $U_{nj}^0(\bar{x})$ are the desired functions of the vibrational equation (10) and must satisfy the boundary conditions on the surfaces $\Omega_n^{bo}$, i.e.

$$L_n U_{nj}^0(\bar{x}) = 0, \ \bar{x} \in \Omega_n^{bo} \tag{9}$$

Where $\bar{x}_n^s$ – are the coordinates of the s - the support of the n - the body. If a part of the supports is pinched, then the following conditions will be added.

$$\frac{\partial U_{nj}^0(\bar{x}_n^s)}{\partial \alpha_n^s} = 0, \quad (s = 1,....S_n^\alpha \ ; j = 1,....J) \tag{10}$$

Where $\alpha_n^s$ – is the direction of the unit vector along which the rigid pinching of the body is carried out at $x_n^s$ the point.

In the program implementing the algorithm, condition (11) taken into account only for the shells of rotation. The presence of rigid racks between the *n* and (*n+1*) - the body is taken $N \geq 2$ into account by the relations.

$$U_{nj}^0(\bar{x}_n^r) - U_{n+1,j}^0(\bar{x}_n^r) = 0 \ (r = 1,...R_n; j = 1,..J), \tag{11}$$

Where $\bar{x}^r$ - is the coordinate of the *r* - the rack, $R_n$ - is the number of racks between the *n*, (*n*+1) - the bodies. In the case of $N = 1$ condition (12), none. Thus, the displacement vector is additionally subject to restrictions of the type (10)-(12). The superimposition of point connections on the system taken into account using the Lagrange multiplier method. Then the vibrational equation (8) rewritten as.

$$\delta\left\{G(U_{nj}^{0}(\overline{x}),\omega^{2})+\sum_{n=1}^{N}\sum_{s=1}^{S_n}\sum_{j=1}^{J}\lambda_{nj}^{s}U_{nj}^{0}(\overline{x}_{n}^{s})+\sum_{n=1}^{N}\sum_{s=1}^{S_n^{\alpha}}\sum_{j=1}^{J}k_{nj}^{s}\frac{\partial U_{nj}^{0}(\overline{x}_{n}^{s})}{\partial\alpha_{n}^{s}}+\sum_{n=1}^{N-1}\sum_{r=1}^{R_n}\sum_{j=1}^{J}\mu_{nj}^{r}\left[U_{nj}^{0}(\overline{x}_{n}^{r})-U_{n+1,j}^{0}(\overline{x}_{n}^{r})\right]\right\}=0 \tag{12}$$

Where $\lambda_{nj}^{s}, k_{nj}^{s}, \mu_{nj}^{s}$ – are the Lagrange multipliers. It is necessary to find the spectrum of complex natural frequencies $\omega^{k}=\omega_{R}^{k}+i\omega_{I}^{k}$, where $\omega_{R}^{k}$ – are the frequencies, and $\omega_{I}^{k}$ – are the damping coefficients of the natural damped oscillations.

## RESULTS AND DISCUSSION

The approximate solution of the vibrational equation (13) is sought in the form of an approximating form composed of fundamental functions that satisfy both the equation and the given geometric boundary conditions on the surfaces of $\Omega_{n}^{fr}$ each body. It is assumed that the functions $\Phi_{nj}^{k}(\overline{x})$ for such bodies are known (for rectangular plates and circular cylindrical shells, this is the fundamental sequence of beam functions). Then the approximating forms can be constructed as a finite expansion over these known functions:

$$U_{nj}^{0}(\overline{x})=\sum_{k=1}^{K}\gamma_{nj}^{k}\Phi_{nj}^{k}(\overline{x}) \tag{13}$$

Where $\gamma_{nj}^{k}$ – are the desired complex coefficients.

You can $\Phi_{nj}^{k}(\overline{x})$ pre-normalize it. $\Omega_{n}^{fr}$ The sum (14) satisfies the boundary conditions automatically due to the choice of terms. By varying the generalized coordinates $\lambda_{nj}^{s}, k_{nj}^{s}, \mu_{nj}^{s}, \gamma_{nj}^{s}$ of equation (13), we obtain a homogeneous system of linear equations. The dimension of this system, where, $J\cdot N'\times J\cdot N'$, where, $N'=\sum_{n=1}^{N}(S_{n}+S_{n}^{\alpha}+R_{n})+N\cdot K$, J - is the number of components of the displacement vector $U_{nj}^{0}(\overline{x})$. Without giving specific calculations, we will write this system in matrix form:

$$(A+\sum_{n=1}^{N_n}f_{n}(\omega_{R})A_{n}^{n}+\sum_{n=1}^{N-1}\sum_{l=1}^{L_n}f_{ln}(\omega_{R})A_{ln}^{n}+\sum_{n=1}^{N}\sum_{l'=1}^{L_n}f_{l'n}(\omega_{R})A_{l'n}^{n}-\omega^{2}B)\overline{\xi}=0 \tag{14}$$

Where $\overline{\xi}$ – is the vector-column of generalized coordinates; $N_n$ - the number of viscoelastic bodies of the system; *B* – a symmetric, degenerate matrix of generalized masses of the system; $A_{n}^{n}, A_{ln}^{n}, A_{l'n}^{n}$ - square matrices $J\cdot N'\times J\cdot N'$ of dimension consisting of zeros , except for the sub matrices of instantaneous stiffness's of the n - the viscoelastic body of the *l*- the shock absorber and the *l*′- the viscoelastic support, respectively; A-a symmetric matrix, (its sub matrix $A^0$ of dimension $J\cdot K\times J\cdot K$) is the generalized total stiffness of the elastic elements of the system, and $A_{H}=A_{b}^{T}$ the sub matrices take into account the kinematic conditions of the rigid elements imposed on the system point connections. The most effective way to solve such equations is the Muller method, which is used here. Without revealing the frequency determinant, we calculate at each step only its value for a fixed value $\omega=\omega_{R}+i\omega_{I}$. Damping coefficients-allow us to judge $\omega_{I}$ the damping properties of the system under consideration. In engineering, the logarithmic decrement of vibration damping used to estimate the rate of damping of oscillatory processes. It related to the damping coefficient by the following formula:

$$\delta = \frac{2\pi\omega_I}{\omega_R}$$

A system of rectangular plates with point constraints is considered. The relaxation kernel for deformable elements (shock absorbers) is chosen in the form $R(t) = Ae^{-\beta t}t^{\alpha-1}$, where $A, \alpha, \beta$ - are the parameters of the kernel. The viscosity of the shock absorber is assumed to be such that its creep deformation during the quasi-static process is a small fraction of the total (~12%).

Table 1 shows comparisons of the first two natural frequencies $\omega_1, \omega_2$ of the elastic shell with the results of [15]. A twentyfold increase in the shell thickness (third row of the table) increases the second and first natural frequencies by 3.6 and 3.7 times, respectively.

**TABLE 1.** Calculation results for the two lowest frequencies.

| $h$ | $\omega_1$ | $\omega_R^1$ | $-\omega_I^1$ | $\omega_2$ | $\omega_R^2$ | $-\omega_I^2$ |
|---|---|---|---|---|---|---|
| 0.01 | 0.0987 | 0.0872 | $0.4 \cdot 10^{-4}$ | 0.142 | 0.135 | $0.94 \cdot 10^{-4}$ |
| 0.1 | 0.274 | 0.26 | $0.34 \cdot 10^{-3}$ | 0.281 | 0.27 | $0.44 \cdot 10^{-3}$ |
| 0.2 | 0.339 | 0.322 | $0.53 \cdot 10^{-3}$ | 0.502 | 0.48 | $0.11 \cdot 10^{-2}$ |

For this case, the kernel parameters are as follows: $A = 0.0070, \alpha = 0.1, \beta = 0.05$. The block is a rectangular parallelepiped (Fig. 1) made of Styrofoam ($\lambda = 3.5 \cdot 10^6 N/m^2$. $\mu = 1.5 \cdot 10^6 N/m^2$ $\rho = 0.15 \cdot 10^3 N/m^2$, $A = 0.048$; $\beta = 0.05$; $\alpha = 0.1$). Block dimensions 70·20·140 mm. Along the outer perimeter there is an aluminum belt, with the help of which the unit is attached to the body of the device.

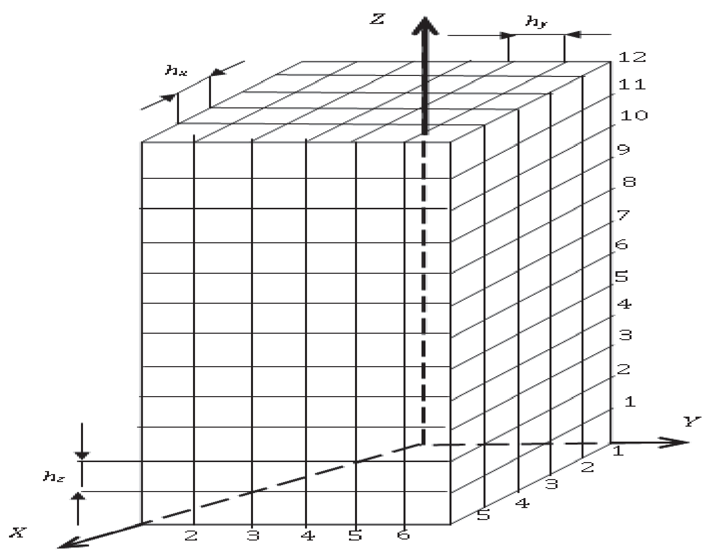



**FIGURE 2.** Design scheme of the monolithic block.

Since aluminum has a modulus of elasticity three orders of magnitude greater than foam, the deformation of aluminum in the direction of the axis is not taken into account. The foam contains small radio components, the influence of which on the amount of elastic deformations of the foam also not taken into account. The increase in the mass of the foam due to radio components taken into account using the mass coefficient *m*. When building a model, the block is divided into finite elements, the dimensions of which are equal (Fig. 2) $h_1 = 1 \cdot 10^{-3} m$; $h_2 = 0{,}20 \cdot 10^{-3} m$; $h_3 = 1 \cdot 10^{-3} m$.

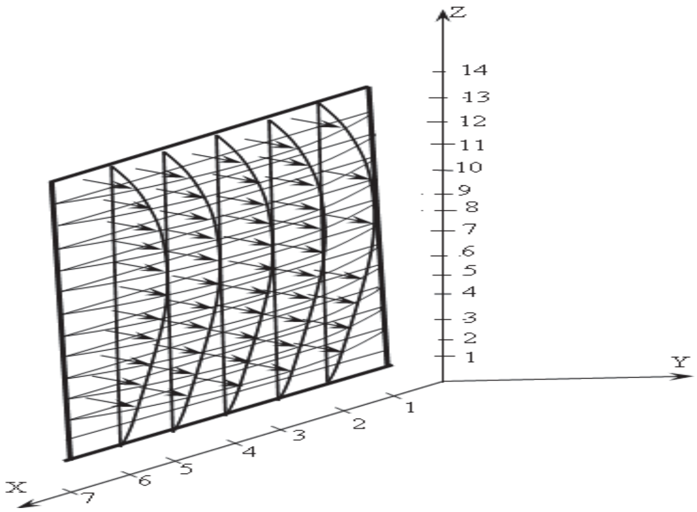



**FIGURE 3.** Direction of transverse movements.

Thus, 70, 80 and 140 grid steps are laid along the length, width and height of the block, respectively, which provides a relative error in calculating the oscillation amplitude with a minimum coefficient of the difference scheme $a_{11} = \frac{\mu r^2}{\rho} \quad h_1^2 = 0,05$ within 10%.

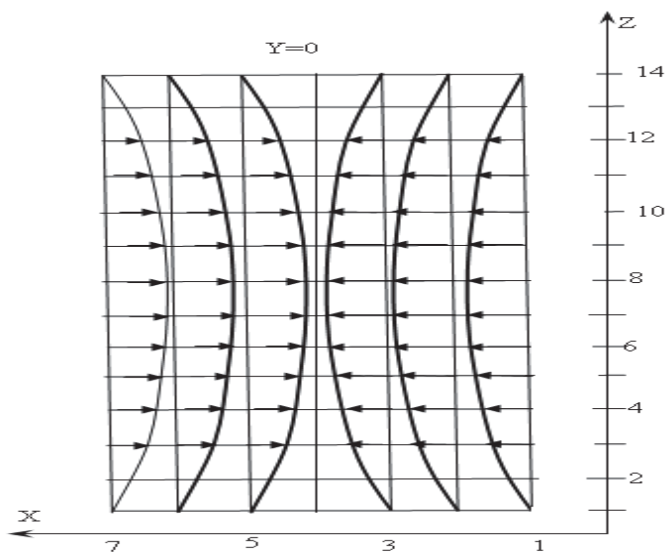



**FIGURE 4.** Direction of shear displacements in the Cartesian coordinate system.

**TABLE 2.** The value of the movements of the monolithic block of the central point under vibration influences.

| Z | 0 | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|
| 0 | 0.0 | 0.0 | 0.0 | 00 | 0 | 0 | 0 |
| 1 | 0.01197 | 0.01036 | 0.00598 | 0.0 | -0.00598 | -0.01036 | -0.01597 |
| 2 | 0.02324 | 0.02013 | 0.01162 | 0.0 | -0.01162 | -0.02013 | -0.02322 |
| 3 | 0.03316 | 0.02872 | 0.02058 | 0.0 | -0.01638 | -0.02836 | -0.03381 |
| 4 | 0.04115 | 0.03564 | 0.02338 | 0.0 | -0.01658 | -0.03558 | -0.04218 |
| 5 | 0.04675 | 0.04049 | 0.02337 | 0.0 | -0.02058 | -0.04059 | -0.04567 |
| 6 | 0.04963 | 0.04298 | 0.02481 | 0.0 | -0.02337 | -0.04371 | -0.04732 |

| Z | 0 | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|
| 7 | 0.04963 | 0.04298 | 0.02337 | 0.0 | -0.02481 | -0.04981 | -0.04116 |
| 8 | 0.04675 | 0.04049 | 0.02058 | 0.0 | -0.02537 | -0.04337 | -0.04231 |
| 9 | 0.04115 | 0.03564 | 0.01658 | 0.0 | -0.02658 | -0.03647 | -0.03036 |
| 10 | 0.03316 | 0.02872 | 0.01162 | 0.0 | -0.01658 | -0.02783 | -0.02754 |

Table 2 shows the value of the displacements of the monolithic block of the central point under vibration influences At this point there is no node, so the displacements are calculated as the average of the arithmetic displacements in the four neighboring nodes (4,4,7), (4,5,7),(4,4,8) and (4,5,8). According to the above C++ program, calculations were performed in 450 steps. To calculate the resonance for bending vibrations, the approximate shape of the oscillations is set as the initial conditions for the maximum deviation of the nodes of the grid model from the equilibrium position. To do this, the initial values of the displacements at the two initial moments of time can be set distributed over the volume of the block in the form of a sinusoid (Fig. 3 and 4). According to the compiled program, the calculation was made for $10^{-3}$ in time.

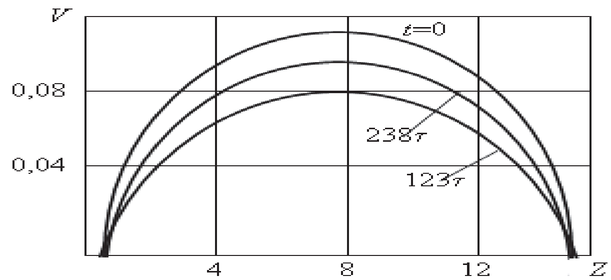



**FIGURE 5.** Dependences of changes in the displacement of the coordinate (the central point of the block at ω=424.671.

Fig. 5 shows the graphs of the movements of V nodes located on the axes of symmetry of the block at different times when the oscillations R.E.E. reached the amplitude values. The graphs show how the natural form of oscillations is formed at the frequency of the first harmonic.

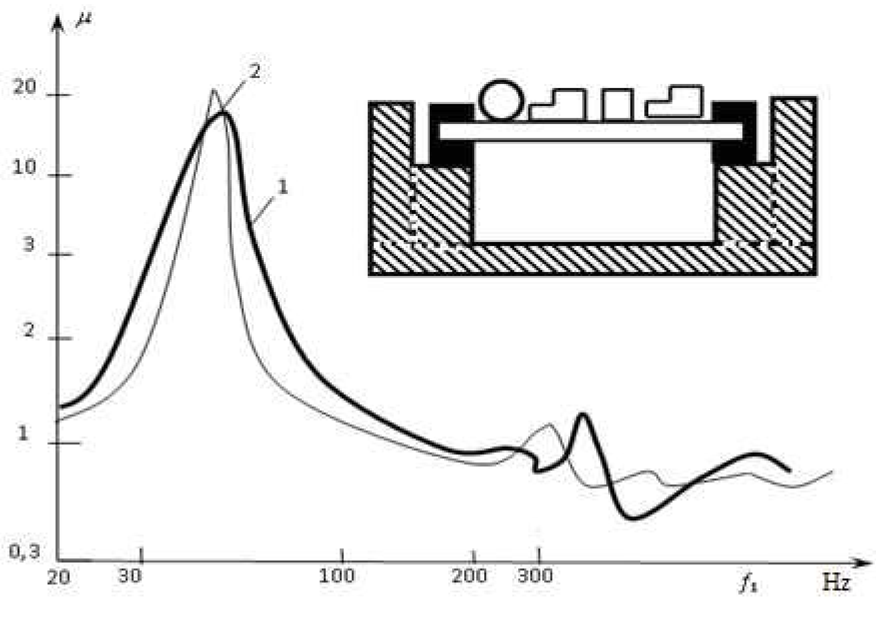



**FIGURE 6.** Dependences of the change in the dynamic coefficient on the frequency with elastic viscoelastic fastening to supports (rubber gaskets) .1-A = 0.0078; 2. A = 0.0048.

As a material, we used foam E=2·10^7N/m^2. There are small radio components, the influence of which on the value of elastic deformations of the foam not taken into account. The figures correspond to the case when the mass of radio components was not taken into account. Figure 6 shows graphs of the coefficient of dynamism of the central point of the block in time, taking into account the addition of mass. In the first case, all the mass coefficients were taken equal to one, which made it easier to check the stability of the calculation process and analyze the results. During the oscillation period is equal to 2.3460 s, the frequency f=424.671 Hz. At the beginning of the calculation, the oscillations are very different from the sinusoid, since the initial shape of the oscillations is given approximately, especially when taking into account the mass of radio components. The fig. 6 shows that with an increase in the viscosity of R.E.E. amplitude, the maximum value of the dynamic coefficient decreases to 10%. In the process of oscillation, the higher harmonics, due to the consideration of viscosity, are attenuated and the first harmonic is released. Graphs of the displacements of V nodes located on the axes of symmetry of the block at different times when the oscillations R.E.E. reached the amplitude values are shown. Now we study the vibrations of the block, taking into account the attached masses and external influences are kinematic (applied on the basis of vibrational displacement). Fig. 7 shows graphs of the movement of the block node from the frequency, where 1 is the mass located at the central point 2 and 3 is the mass located at the extreme points (Fig. 6). Radio components are available with different weights, which is taken into account in the calculation using the additional mass coefficient.

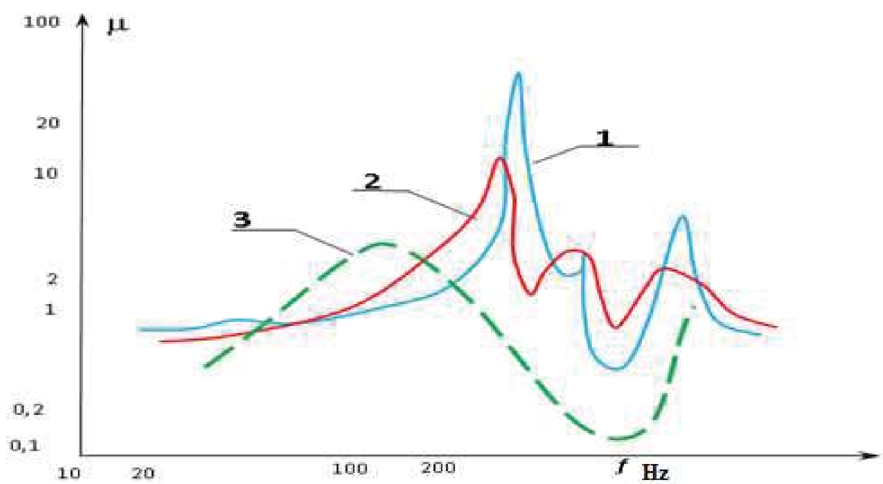



**FIGURE 7.** Dependence of the dynamic coefficient change on the frequency.

Fig. 8 shows the graphs of the movements of U and V of the central point of the block in time. Under the influence of pulsed loads in the form of unit, Heaviside functions. The dimensions of the plate are 140x100x2 mm. The steps of the elements are selected as h_$h_1 = h_2 = 1 \cdot 10^{-3} m$,$h_3 = 0.5 \cdot 10^{-4} m$. Next, you need to determine the value of the time step from the stability condition of the numerical solution and the values of the coefficients included in the finite element scheme. It can be seen that the maximum value of mixed decreases over time. The foam contains small radio components, the influence of which on the amount of elastic deformations of the foam also not taken into account.

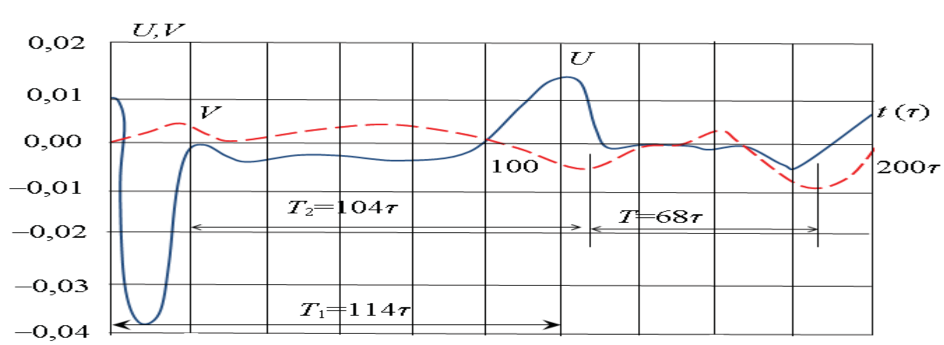



**FIGURE 8.** Change of longitudinal and transverse displaced points of the block depending on time 4.

## CONCLUSIONS

Thus, the paper developed a solution method and an algorithm for determining the resonant state of the R.E.E. block from vibration loads, the analysis of the amplitude-frequency response showed a satisfactory convergence of the calculation using the finite difference method. Taking into account the viscosity of the shock absorber block (plate) material reduces the displacement amplitudes by 15% to 20%. It was also found that the presence of rubber shock absorbers reduces the amplitudes of vibrations of the equipment by up to 30%.

## REFERENCES


1. S. A. Lychev, *XXXII Summer School - Conference "Advanced problems in mechanics"*, Saint Petersburg, 2004, Academy, p. 89.
2. S. A. Lychev, *XXXIII Summer School - Conference "Advanced problems in mechanics"*, Saint Petersburg, 2005, Academy p. 80.
3. A.V. Lysenko, Reliability and quality of complex systems, **4,** 41-44, (2013).
4. A. Nashif, D. Jones and J. Henderson *Damping of vibrations: Trans. from English,* (Mir, Moscow, 1988), p. 448.
5. A.V. Zatylkin, Innovations based on information and communication technologies, **1,** 223-225, (2013).
6. A.V. Lysenko, *Reliability and quality-2012: proceedings of the International Symposium*, Penza, 2012, edited by N. K. Yurkov (Publishing House of PSU, Russia, 2012) vol. **2,** pp. 371-373.
7. A.V. Zatylkin, Izvestiya SFU, **5,** 94-99, (2012).
8. I. I. Safarov, M. K. Teshaev and Z. I. Boltaev, Journal of Critical Reviews, **7(12),** 893-904, (2020).
9. M. Mirsaidov, I. Safarov, Z. Boltayev and M. Teshaev, IOP Conference Series: Materials Science and Engineering, **869(4),** 042011, (2020).
10. I. Safarov, M. Teshaev and J.A.Yarashev, IOP Conference Series: Materials Science and Engineering, **1030(1),** 012073, (2021).
11. M. Mirsaidov, I. Safarov and M. Teshaev, Journal of Applied Mathematics and Physics, **7(11),** 326-331, (2019).
12. M. Mirsaidov, I. Safarov and M. Teshaev, E3S Web of Conferences, **164,** 14013 (2020).
13. M. K. Teshaev, I. I. Safarov, I. Kuldashev and N. Ishmamatov, Journal of Vibration Engineering, **4(7),** 78-92, (2020).
14. K. I. Bilibin, A. I. Vlasov, L. V. Zhuravleva, *Engineering design of electronic equipment*, (Publishing house of MSTU N.E.Bauman, Moscow, 2002), p. 528.
15. I. I. Safarov, European Journal of Engineering Research and Science, **3(6),** 128-135, (2018).