Comparison of modeling the natural frequencies and vibration modes of a sound isolation panel fragment with full-scale tests

E A Romanenko, G S Russkikh and Z N Sokolovskiy
Omsk State Technical University, Mira ave., 11, Omsk, 644050, Russia

E-mail: elinaromanenko18@gmail.com, russkikh_gs@mail.ru, ninasok@yandex.ru

Abstract. The article discusses the results of modeling natural frequencies and modes of vibration, performed in the Ansys Workbench software package in comparison with a full-scale experiment. The aim of the study is to verify the model built in Ansys Workbench. The results allow us to make a conclusion about the adequacy of the model and the possibility of its application for solving the problem of determining the natural frequencies and vibration modes.

1. Introduction

The finite element method (FEM) is used by many researchers to study various technical systems. FEM is an affordable, convenient and reliable way to determine the required parameters. This is especially true in cases where the system has a complex structure, the parameters of which can be determined by the apparatus of the classical theory only with the use of significant assumptions and it is impossible or difficult to conduct a natural experiment.

The article [1] provides a study on the harmonic analysis of propellers using the Ansys Workbench software package. The purpose of the harmonic analysis is to verify the hypothesis of enhancing the performance of the screw due to metal replacement composite material. Carrying out such a comparative analysis makes it possible to test the hypothesis in a short time, the operation of replacing the material without changing the geometry of the model does not cause difficulties. Amplitude-frequency characteristics, damping and displacement estimates were obtained. This calculation template can be used to predict the operating frequencies and vibration modes of propellers.

The work [2] describes the modal and harmonic analysis of a multilayer spring in order to select the optimal material. The calculation was carried out in Ansys APDL. The resulting frequency response made it possible to compare the selected materials, assess their compliance with the set goals and determine the most effective options.

The solution to the problem of the dynamics of multilayer plates and the reduction of unwanted vibrations using Ansys APDL is also considered in [3]. The dependence of the resonance pattern and the response of a symmetric plate is investigated for different boundary conditions, different number of layers and different thicknesses. The calculation also made it possible to determine the condition for reducing the maximum amplitude at the resonance point, which is an important result that can be used in the design of such systems.

Another example of the calculation of natural frequencies and vibration modes performed in the Ansys Workbench is [4]. Prediction of the damping properties of a beam is carried out for various combinations of composite materials. More than 5 modes, frequency response and phase response (Bode plot), damping factors have been determined. This prediction of natural frequencies and modes of vibration allows a more confident approach to design and more freedom in the choice of materials.
The object of the study is a flexible soundproofing fence, which is mesh-plate flexible panels (MPP). MPP has a multilayer structure with notches uniformly distributed on the area, which is caused by the presence of flexibility. The design is protected by patents of the Russian Federation [5], [6].

Some of the first mathematical models of panel vibrations were published in [7], [8]. Were investigated MPP's single row consisting of 15 doubled plates. The strip is positioned horizontally and is rigidly fixed to the frame plates extremes. Steel wires working only in tension and compression link them together. The proposed model describes small vibrations of the panel near the equilibrium position under harmonic action. In [8], the values of the generalized coordinates in the position of static equilibrium and the values of natural frequencies are determined. In the article [9], natural frequencies and vibrations modes were also determined using the finite element method in the Ansys software package. The results have sufficient convergence with the analytical calculation given in [8].

Modeling of static deflection and the first natural frequency of panel vibrations in a horizontal position is presented in article [10]. Research object represented as a set of elastic yarns with the reduced mass of the plates.

The research described in this article does not contradict previously published results, but shows the possibility of a different approach to solving the problem.

2. Statement of the problem

The sound-insulating mesh-plate flexible panel consists of segments - rows of square plates, fixed on a metal mesh. To assess the sound-insulating properties of the panel, it is necessary to determine the type of amplitude-frequency characteristic (AFC), which makes it possible to estimate the value of the first natural frequency and the first form of free oscillations. AFC type depends on the size of the panel and its individual components - the plates, the materials used (modulus of elasticity, weight) and the carrier grid parameters.

When upgrading the panel by replacing materials, it is difficult to conduct full-scale tests for each replacement due to the complexity of the assembly and the large amount of materials. It requires the development of a universal mathematical model and computer model that will evaluate the feasibility of using alternative materials in the design panel.

3. Theory

In solving the problem of constructing the AFC a key parameter is the size of the mesh of the elastic modulus. It has already been established by the fact that the modulus of elasticity of the wire and correspondingly woven mesh several times different from the modulus of elasticity of the starting material - steel [11]. In [10] presented rheological model, the application of which was built chart stretching wire mesh, calculated tangent modulus and unloading unit with the experimental data. The experimentally determined elastic modulus was $E_{\text{mesh}} = (7.94-8.77) \times 10^4$ MPa.

Initial data:
- fragment length $L = 600$ mm (the end plates fixed to a frame, not taken into account),
- elastic modulus $E = 90 \cdot 10^3$ MPa,
- weight of one plate $m = 0.27$ kg,
- plate size $b = 65.4$ mm,
- the grid has square cells with a clear pitch $h = 0.9$ mm,
- mesh wire diameter $d = 0.22$ mm,
- number of mesh threads $n = b / (d + h) = 57$.

The frequency response depends on the form of the initial position of the sample, which corresponds to the application of the load deflection. The calculation of the deflection of a panel fragment was carried out according to the model of plane bending and tension of the bar in large displacements with a uniform linear load.
\[
\begin{align*}
\frac{dv}{dx} &= -(1 + \frac{N \cdot \cos \varphi - Q \cdot \sin \varphi}{E \cdot A}) \cdot \sin \varphi, \\
\frac{dw}{dz} &= \frac{E \cdot A}{N(\cos \varphi - Q \cdot \sin \varphi)} \cdot \cos \varphi + \cos \varphi - 1, \\
\frac{dN}{dz} &= -q_x, \\
\frac{dQ}{dz} &= -q_y, \\
\frac{dM_z}{dz} &= (N \cdot \sin \varphi + Q \cdot \cos \varphi) \cdot (1 + \frac{N \cdot \cos \varphi - Q \cdot \sin \varphi}{E \cdot A}), \\
\frac{d\varphi}{dz} &= \frac{M_x}{E \cdot J_z}.
\end{align*}
\]

where \(v(z)\) – lateral movement, \(N(z)\) – longitudinal force, \(\varphi\) – angle of rotation of the section about the x axis, \(E\) – elastic modulus, \(A\) – cross-sectional area, \(w(z)\) – longitudinal movement, \(q_y\) and \(q_x\) – linear loads.

with \(\frac{dM_x}{dz} \approx 0\) : \(Q = -N \cdot \tan \varphi \Rightarrow \frac{dq}{dz} = -q_y = \frac{dN}{dz} \cdot \tan \varphi - N \cdot \frac{1}{\cos^2 \varphi} \cdot \frac{d\varphi}{dz}\)

\[
\frac{d\varphi}{dz} = \frac{\cos^2 \varphi}{N} \cdot (q_y + q_z \cdot \tan \varphi)
\]

we get a system of differential equations of the 4th order:

\[
\begin{align*}
\frac{dv}{dz} &= -(1 + \frac{N(\cos \varphi + t \varphi \cdot \sin \varphi)}{E \cdot A}) \cdot \sin \varphi, \\
\frac{dw}{dz} &= (1 + \frac{N(\cos \varphi + t \varphi \cdot \sin \varphi)}{E \cdot A}) \cdot \cos \varphi - 1, \\
\frac{d\varphi}{dz} &= \frac{\cos^2 \varphi}{N} \cdot (q_y + q_z \cdot t \varphi), \\
\frac{dN}{dz} &= -q_x.
\end{align*}
\]

Boundary conditions: at the fixed ends \(v (0) = w (0) = 0; v (L) = w (L) = 0\).

The system was solved numerically, the maximum deflection \(\max v(z, L) = \delta\) and \(N(\delta, L)\) was determined.

The yarn deflection shape is described by the following equation:

\[
v(z, \delta, L) = 4 \cdot \delta \cdot \frac{z}{L} \cdot \left(\frac{z}{L} - 1\right),
\]

where \(\delta\) – maximum deflection, \(z\) – current cross-section coordinate, \(L\) – distance between supports.

\[
N_{w=0}(E, A, \delta, L) = K_N \cdot E \cdot A \cdot \left(\frac{\delta}{L}\right)^n,
\]

with \(w=0\): \(K_N = 2.6, n = 2\).

When changing the sample length, the coefficients change insignificantly.

The determination of the first vibration mode and natural frequency was carried out by the Rayleigh method (energy method). The basis of this approach is the law of conservation of the total mechanical energy of an oscillating system. During oscillations, transitions from kinetic energy to potential energy (and vice versa) occur, that is, their sum remains constant, and energy dissipation in this case can be neglected.
The potential energy of an elastic system is the work performed by the internal and external forces of the system when it is transferred from the deformed state to the initial state (before deformation). Potential energy is a function of displacement, and kinetic energy is a function of speed.

Take the form of natural oscillations steady horizontal fragment as a static deflection $\delta_{st}$ from extreme positions $\delta^+$ and $\delta^-$:

![Figure 1. Fragment positions during oscillations](image)

Calculate the amplitude of oscillations:

$$\delta_a = \frac{\delta^+ - \delta^-}{2}$$

Determine the dynamic equilibrium position (when the vertical speed is zero):

$$\delta_0 = \frac{\delta^+ - \delta^-}{2} + \delta_{st}$$

The kinetic energy at the moment of passing the position of dynamic equilibrium is maximum and is equal to:

$$K = \frac{1}{2} \int_0^L v^2(\delta_a, z, L) \cdot \omega^2 \cdot m \cdot dz = \frac{4^2}{2 \cdot 30} \cdot m \cdot (\delta_a)^2 \cdot \omega^2 \cdot L,$$

where $\omega$ – circular frequency of natural vibrations, $m$ – mass per unit length of the fragment.

At the positions $\delta = \delta^-$ and $\delta = \delta^+$, the kinetic energy is zero.

Calculate the increase in the potential energy of weight at $\delta = \delta^-$:

$$\Delta \Pi^- = \int_0^L m \cdot g \cdot v(z, |\delta^- - \delta_0|, L) \cdot dz = \frac{2}{3} \cdot m \cdot L \cdot g \cdot |\delta^- - \delta_0|, (\leq 0)$$

where $g$ – acceleration of gravity.

Determine the increment the weight potential energy at $\delta = \delta^+$:

$$\Delta \Pi^+ = -\int_0^L m \cdot g \cdot v(z, |\delta^+ - \delta_0|, L) \cdot dz = \frac{2}{3} \cdot m \cdot L \cdot g \cdot |\delta^+ - \delta_0|, (\geq 0)$$

Compute the potential tension energy of the strip:

$$T = \frac{N^2 \cdot L}{E \cdot A}$$

Determine the increment the strip tension potential energy at $\delta = \delta^-$:

$$\Delta T^- = \frac{L}{E \cdot A} \left[ N^2(\delta^-) - N^2(\delta^0) \right]$$

Calculate the increment the strip tension potential energy at $\delta = \delta^+$:

$$\Delta T^+ = \frac{L}{E \cdot A} \left[ N^2(\delta^+) - N^2(\delta^0) \right]$$
The problem was solved numerically. Variable parameters are $\delta^+$ and $\delta^-$, solution constraints:

$$\Delta T^+ + \Delta \Pi^+ = K,$$

$$(\Delta T^+ + \Delta \Pi^+) - (\Delta T^- + \Delta \Pi^-) \rightarrow 0$$

In fig. 2 shows particular cases of calculating the position of a fragment for different variants of the energy distribution at $\delta_{st} = -18$ mm, $f = 8$ Hz.

**Figure 2.** Fragment positions

As a result of the calculation, an assessment of the forces and energy distribution was obtained, the experimental characteristic of a panel fragment (strip) is the dependence of the amplitude on frequency. Wherein:

– the value of the elastic modulus accepted in the calculations is largely determined by the assembly technology,
– graphs characterize the band with specific parameters and illustrate the general patterns of frequency response,
– the initial approximation was taken $\delta \leq \delta_{st}$.

In fig. 3 shows the curves describing the amplitude-frequency response at different values of the static deflection.

**Figure 3.** Frequency response at different values of static deflection

The graphs show that for any value of static deflection for a strip of 22 plates with the indicated parameters, the peak of the resonance curve is reached at a frequency of 12 Hz, while the amplitudes will be different.

4. Experimental results

For the study was used a fragment of a panel of 22 plates attached to both sides of a metal grid with a lining 4 of basalt fabric layers disposed symmetrically with respect to the grid.
To install the sample, a steel frame was assembled from corners fixed in each corner with 4 screws. A fragment of the panel was installed on the frame and fixed with M8 screws at the extreme points. Excitation was carried out using a pneumatic sleeve, which was acted upon by a shaker pusher. To obtain the values of vibration accelerations, three sensors were used - accelerometers with a magnetic mount: the first accelerometer was mounted on the central plate, the second on the frame, and the third on the base of the shaker. In fig. 5 shows the layout of the stand. The tests were carried out on a vibration unit ZET 027 with the use of specialized software ZetLab of the Department of OTMiAU OmSTU (Fig. 6).

**Figure 4.** Sample fixed to the frame

**Figure 5.** Schematic of the experimental stand: 1 – frame, 2 – a fragment of the MPP (strip), 3 – vibration stand, 4 – vibration sensor, 5 – pneumatic sleeve, 6 – soft supports.
To construct the AFC three frequency bands were selected: [60 ... 3], [3 ... 2000], [3 ... 10000] Hz. The task is definition resonance at transverse oscillations. The view of the panel for removing the frequency response is shown in Fig. 7.

The graphs shown in Figure 8 are plotted in the coordinates frequency (Hz) - acceleration from the accelerometer (m/s²). To recalculate the amplitude of acceleration into the amplitude of displacement (mm), the following formula was used:

$$A_x = A_a \cdot \frac{10^3}{(2\pi \cdot f)^2}$$
Figure 8. View AFC

Expanded uncertainty of frequency measurements using the spectrum analyzer of the Zetlab setup $u(f) = 6.7\%$.

The deviation of the measured values from the calculated one was $[1.5 \ldots 8] \%$. The magnitude of the deviation is proportional to the size of the investigated frequency range.

5. Discussion of results

To determine the frequency response of the sample in the form of a strip, numerical modeling was carried out in the ANSYS Mechanical.

The problem is split into two components: calculation of deformations and stresses in Static Structural and determination of frequency response in Harmonic Response. This is due to the need to take into account the dependence of the AFC type on the initial position of the sample - the deflection and displacement of one of the supports. The problem was solved in a linear setting.

Based on the geometry, a finite element mesh was generated with the following parameters: number of nodes: 607250, number of elements: 211235.

As a fixing, a rigid embedding of two of the extreme lower plates was used; for the second pair of extreme plates, the displacement of the support was set in the range from 1 to 20 mm. Loads: gravity (Standard Earth Gravity) and pressure, evenly distributed over the surface, of 100,000 Pa.
Earlier, during a full-scale experiment, the shape and size of the deflection were determined, the maximum value was 18 mm. The longitudinal displacement corresponding to this value is 2 mm.

The deformations defined in the specified setting are shown in the following figure. The deflection shape corresponds to the shape observed in a field experiment. The stresses on the supporting mesh do not exceed the permissible ones, \( \sigma_{\text{max}} = 160 \leq [\sigma] = [540] \) MPa.

The model with the resulting deformations was submitted to Harmonic Response. The sample was accelerated up to 30 m/s\(^2\). An amplitude-frequency characteristic was built, 1 natural frequency was determined.
Figure 12. View frequency response obtained from the computer simulation

The value of the first natural frequency was 12 Hz, which corresponds to the previously obtained data.

6. Conclusions
Based on the results of calculations, a full-scale experiment and computer modeling of the vibrations of a fragment of a sound-insulating panel, the first natural frequency was determined and an amplitude-frequency characteristic was constructed. Computer simulations have shown a solution that converges at a sufficient level to the data of the analytical calculation and experimental results. It can be concluded that the use of the model built in Ansys Workbench is possible and advisable in the case when testing is difficult. The authors plan to conduct a series of similar studies in order to modernize the panel by replacing materials.

Reference
[1] Rao Y S and Reddy B S 2012 *Intern. J. of Research in Engineering and Technology* 1-3 pp 257–260
[2] Rao P S and Venkatesh R 2015 *Intern. J. of Novel Research in Electrical and Mechanical Engineering* 2-3 pp 67–75
[3] Narwariya M, Choudhury A and Sharma A K 2018 *Adv. Comput. Des* 3-2 pp 113–132
[4] Saini M K et al. 2020 *Materials Today: Proceedings*, https://doi.org/10.1016/j.matpr.2020.08.717
[5] Zubarev A, Tribelskiy I, Adonin V and Malyutin V 2007 *Sound Insulation Panel* [http://www1.fips.ru/wps/portal/IPS_Ru#1518716254883] Patent 2340478 Russian Federation
[6] Tribelskiy I, Adonin V, Bobrov S, Denisov V, Bokhan V and Gidion V 2012 *The sound-insulating panel and the method of its manufacturing* [http://www1.fips.ru/wps/portal/IPS_Ru#1518716165441] Patent 2457123 Russian Federation
[7] Fedorova M A and Korneev S A 2011 *proc. Russia is young: advanced technologies to the industry!* (Omsk) 1 pp 133–136
[8] Korneev S A and Fedorova M A 2011 *Omskiy nauchnyy vestnik* 103 pp 129–133
[9] Grakov S A 2014 *proc. Dynamics of Systems, Mechanisms and Machines* 1 pp 50–52
[10] Taran V A, Russkikh G S and Sokolovskiy Z N 2016 *Omskiy nauchnyy vestnik* 148 pp 51–55
[11] Fedorova M A et al. 2014 *proc. Dynamics of Systems, Mechanisms and Machines* 1 pp 161–164