Prediction of sound insulation of flexible panels in low frequencies with dynamics and volume relations

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Abstract. The problem of sound insulation in the low-frequency region (up to 100 Hz) is relevant, but little has been studied theoretically and practically has not been investigated experimentally due to the lack of the necessary experimental base. Especially important is the solution of the problem for local isolation of low-frequency mechanisms by perspective mesh-plate flexible panels (MPP) for various configurations of isolated and protected volumes and at low first natural frequencies. The purpose of this study is to evaluate the effect of forced vibration panel and room volume ratio of the source of the perturbation and the protected volume. The objective is to develop a mathematical model, which can verify the adequacy of the experimental results obtained at frequencies above 30 Hz and acoustic reverberation chambers interferometer. As a result, a formula for calculating the sound insulation has been received, allowing taking into account the volume ratio of the shared spaces and the transition through resonance. The results of the research can be used in the design of direct soundproofing of low-speed mechanisms or for the organization of the first stage of soundproofing and modernization of the design of the MPP in relation to specific conditions and requirements for sound insulation.

1. Introduction
One of the key problems of modern acoustics is the provision of reliable sound insulation in the low frequency region. The range has a number of features, such as possible forced oscillations of soundproof barriers and the need to take into account the volume ratio from the source of sound and the protected volume. These factors require the engineer to adopt a non-standard approach to the design of protective structures.

In the area of medium and high frequencies, the use of classical sound insulating structures that have reflective and absorbing layers is effective. To provide a similar effect in the low-frequency range, the thickness of the absorbing material is several tens of centimeters, which is not always possible for installation, transportation and operation. Achieving the given sound insulation at low frequencies significantly complicates and increases the cost of construction. In the general formulation, the solution of the problem is the use of resonant absorbers, which are mechanical and acoustic oscillatory systems. Absorbers have several layers tuned to a certain range of absorbed frequencies. One of the developed classes of such systems is the appearance of panels having different degrees of perforation. They are able to provide effective absorption at resonant frequencies [1], a corrugated panel is a particular case [2]. Microperforated panels with a perforation of less than 1 mm have proven to be more effective in various applications [3], [4]. Designing multi-layer perforated panels includes questions of optimization of geometric dimensions, number and orientation of the cavities.

To reduce the level of low-frequency noise, artificially formed, arranged in a certain way (periodic) structures, called metamaterials [7], are also used. These innovative solutions have great prospects of
use, because they allow you to obtain designs with specified properties [5], [6]. Metamaterials of the lamellar type are capable of demonstrating high losses in the transmission of sound at a sufficiently small thickness [7]. The advantage of this technology is the ability to adjust the finished protective structure to changing operating conditions with minimal time and labor. This is possible by changing the thickness, composition and location of the elements. An example of a successful implementation in industry, aircraft construction and space exploration are cellular (honeycomb) structures [8], [9], [10]. They are made of an advantage of polymer structural materials, which have high flexibility and strength, low specific mass.

When creating artificial structures, the use of obstacle systems in the form of cuts, grooves, stiffeners or point defects is often used [11], [12]. This approach is due to the effect of the captured regime. The cutouts usually screen traveling waves reflecting most of the energy, so that the incident wave bounces off such an obstacle. However, at certain frequencies the incident wave is not reflected, but captured, as a result of which there are long-term oscillations with a gradual decay of the wave energy. Experimental way it was revealed that the reduction of the flexural rigidity of the structure, connected with the presence of cutouts, increases the sound insulation by 3-6 dB [13].

An example of the implementation of these approaches is a flexible soundproof barrier, which is a mesh-plate panel (hereinafter referred to as «MPP»). The MPP has a multilayer structure with evenly spaced out cutouts. The design is protected by patents of the Russian Federation [14], [15]. There are test results confirming the effectiveness of the use of memory bandwidth in a wide frequency range, including low frequencies [16]. Because of design features, the MPP is characterized by relatively low frequencies of natural oscillations (less than 20 Hz), in which the experimental investigation of vibration isolation is hampered by the required dimensions of the respective chambers.

Existing models for calculating sound insulation mean infinite radiated and protected spaces, and also do not take into account the panel's full vibrations. In certain technical conditions, it is necessary to know the effect of closed space on the expected sound insulation characteristics. This is especially important at low frequencies and the piston form of oscillations of the soundproof panel, when the volume of transported air is commensurate with the emitting and protected volumes. The latter is typical for the MPP in the area of its most effective application as the first stage of reducing the noise level from the source.

The task of the study is the development and verification of a mathematical model for recording the vibration isolation of low-frequency oscillations of the memory bandwidth for various ratios of the volume containing the disturbance source and the volume of the protected space.

2. Statement of the problem
In Fig. 1 is a schematic diagram of adjacent rooms separated by a soundproof barrier. Zone 1 contains the sound source, zone 2 represents the protected space.
Figure 1. Schematic diagram

Notation:
- \( p_i \) - pressure in the zone, \( i = 1,2 \) – zone number
- \( \Delta p_i \) - sound pressure in the zone from the sound source
- \( \Delta p_i (u) \) - change in pressure in the zone from the movement of the soundproofing plate \( u(y,z) \) with its oscillations with the circular frequency \( \omega \) of the source of sound
- \( S \) - soundproofing plate area
- \( u(y,z) \) - the equation of motion of a sound-insulating plate during its oscillations with the frequency of the sound source, represented in the form \( u(y,z) = u \cdot e^{i\omega t} \)
- \( U(\omega) \) - the average amplitude of displacements (forced oscillations of a plate under perturbation by the frequency \( \omega \))
- displaced volume is

\[
V(u) = \iiint u \cdot K_u \cdot S,
\]

where \( u \) is amplitude of oscillations of a plate on the first (piston) form. Assuming a sinusoidal shape of the deflection of a square hinged plate:

\[
K_u = \frac{4}{\pi^2} = 0.41
\]

In solving the problem, the following assumptions were made:
1) only the first (piston) form of oscillations is considered, since the radiation of sound at subsequent resonances does not have a significant effect on the soundproof characteristic,
2) the density of the medium, the speed of sound in the zones is the same \( \rho_1 = \rho_2 = \rho, c_1 = c_2 = c \),
3) the temperature in the zones is constant \( T = \text{const} \).

3. Theory
The accepted assumptions make it possible to apply the Boyle-Mariotte law, then the pressure amplitude \( \Delta p_i(U) \) in the zone from oscillations of the soundproofing plate is estimated as follows:
\[(p_i + \Delta p_i)V_i = (p_i + \Delta p_i + \Delta p_i(u)) \cdot (V_i + (-1)^{i+1} \cdot u \cdot K_u \cdot S) \]

i.e.
\[
\Delta p_i(u) = -\frac{u \cdot K_u \cdot S}{V_i + u \cdot K_u \cdot S} (p_i + \Delta p_i), \quad \Delta p_2(u) = \frac{u \cdot K_u \cdot S}{V_2 - u \cdot K_u \cdot S} (p_2 + \Delta p_2)
\]

(1)

Consider the movement of the plate [15]:
\[
\frac{\partial^2 u}{\partial t^2} + B(1 + i \eta)u = p_1 + p_2 - p_3
\]

where \(p_1, p_2, p_3\) - sound pressure respectively in incident, reflected and transmitted waves, \(B\) - cylindrical rigidity of the plate, connected with the resonant frequency of the plate by the relation
\[
\omega_p \approx \left(\frac{B}{m}\right)^{1/2}
\]

(3)

We introduce the change in pressure from displacement (1), taking into account the assumptions
\[
p_1 = p_2 = p, \quad V_i \gg u \cdot K_u \cdot S, \quad \Delta p_i \ll p_i
\]

we get (2) as:
\[
m \frac{\partial^2 u}{\partial t^2} + B(1 + i \eta)u = p_{11} - p_{12} - p_{13}
\]

\[
m \frac{\partial^2 u}{\partial t^2} + \left(B(1 + i \eta) + p \cdot K_u \cdot S \left(\frac{1}{V_1} + \frac{1}{V_2}\right)\right)u = p_{11} + p_{12} - p_{13}
\]

Expressing the speed of movement through the vibrational
\[
v = \frac{\partial u}{\partial t} = i \omega u \rightarrow u = \frac{v}{i \omega}, \quad \text{obtain an expression for the impedance barriers:}
\]
\[
i \left(m \cdot \omega - \frac{B(1 + i \eta)}{\omega}\right) - p \cdot K_u \cdot S \left(\frac{1}{V_1} + \frac{1}{V_2}\right) = Z_{np}
\]

(4)

The vibrational velocity is represented as a function of the sound pressures on both sides of the barrier:
\[
v = -\int \frac{1}{\rho_0 c_0} \frac{\partial p}{\partial t} \frac{\partial t}{\partial t} = \frac{p_{11} - p_{12}}{\rho_0 c_0} = \frac{p_{13}}{\rho_0 c_0}
\]

\[
Z_{np} \cdot v = p_{11} + p_{12} - p_{13} = p_{11} + (p_{11} - p_{13}) - p_{13} = 2(p_{11} - p_{13}) \rightarrow
\]

\[
\rightarrow Z_{np} \cdot v = 2p_{13} \left(\frac{p_{11}}{p_{13}} - 1\right) \rightarrow \frac{p_{11}}{p_{13}} = 1 + \frac{Z_{np}}{2Z_0}
\]

(5),

where \(Z_0\) - air impedance, \(Z_0 = \rho_0 \cdot c_0\).

Sound insulation plate is equal to:
\[
R = 10 \cdot \lg \left|\frac{p_{11}}{p_{13}}\right|^2
\]

(6)

After substitution (4) into (5), (5) into (6) and transformation with the (3) we obtain the solution of equation:
\[
R = 10 \cdot \lg \left(1 + \frac{\eta_0 \omega}{2 \rho_0 c_0} \left(\frac{\omega_p}{\omega_p}\right)^2 + \frac{m \cdot \omega}{2 \rho_0 c_0} \left(1 - \left(\frac{\omega_p}{\omega_p}ight)^2 - \frac{K_{uu} \cdot K_u \cdot S \cdot (V_1 + V_2)}{V_1 V_2 \omega^2 \rho_0 c_0}\right)\right)
\]

(7),

where \(K_{id}\) - coefficient reflecting the change of the oscillation phase at the transition through resonance:
\[
K_{id} = \begin{cases} 1, f < f_p \\ -1, f > f_p \end{cases}
\]

Thus, the formula (7) is a special case of the classical theory of formula insulation, allowing considering the ratio of volume of spaces separated by the transition through resonance.

We consider three cases of placement of the memory bandwidth at the interface between two rooms, when:

a) the volumes are equal;

b) the volume of the protected space is several times greater than the volume with the source of sound.
c) the volume of the protected space is less than the volume containing the source;

4. Experimental results

There are known the results of studies of soundproofing MPP, performed with the help of reverberation chambers and acoustic interferometer of experimental base of NPP "Progress" [16].

The volumes of reverberating adjacent cameras are equal, respectively 180 and 130 m$^3$, i.e. $\frac{V_2}{V_1} = 0.722$.

Acoustic interferometer is a tube length 32.5 m, the inner square section has a size 0.8 m. The plate is installed at a distance 15.5 m from the source, i.e. $\frac{V_2}{V_1} = 1.097$.

Figure 3 shows a comparison of the experimental and calculated data on the determination of the sound insulation of the MPP in the indicated conditions.

5. Discussion of results

The following results were obtained during the research:

1) The lower bound to obtain adequate soundproofing promising flexible panels, taking into account the dynamics of the disturbance in the low frequency range and the ratio of the shared volume.
2) A significant influence of the natural frequency of the natural oscillations of the memory bandwidth on sound insulation at frequencies close to the resonance is established, in particular, an increase in isolation at low frequencies (less than 10 Hz) with an increase in the natural frequency of the MPP.

3) A significant increase in insulation at low frequencies at a relative decrease in the volume chamber perimeter, which allows to recommend the "stepped" insulation.

4) In all cases, sound insulation at medium and high frequencies is practically independent of the first frequency of natural oscillations and the ratio of the divided volumes.

Formula (7) gives a lower estimate of sound insulation, since it does not take into account the positive effect of perforation and only approximately takes into account the multilayeredness of the memory bandwidth. As can be seen from Fig. 3, the maximum deviation of calculated and experimental data does not exceed 6 dB.

6. Conclusions
The results of the present work is recommended to use in the preliminary design of flexible panels, soundproofing oriented in the low frequency range.

However, the features revealed in the comparison of simulation results and experimental data require further refinement of the model, as well as additional experimental studies.

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