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Peculiarities of sound insulation in building and technological constructions

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Introduction

The law of air sound insulation has been known since old times. It defines the property of sound insulation which depends upon the Mass Law, i.e., the heavier the construction, the better air sound insulation. In calculation of air sound insulation, the weight of m^2 of this construction, i.e., kg/m^2 is taken into consideration. In the course of time, the need of lighter partition constructions in multistoried buildings arose. Acousticians in cooperation with architects have been searching for ways to reduce the weight of constructional partitions not decreasing the properties of sound insulation. Thus, new theories of sound insulation have been originated and experimentally corroborated.

Two main types of sound insulation and their properties are discussed in this article.

1. Some sound insulation of a single wall

The most common constructions used in buildings are flat one-layer and multi-layer constructions. For its proper purpose these constructions may be used for partitions – wall constructions and slabs – floors and ceiling sound insulation constructions.

A. Normal incidence mass law

When a plane wave which angular frequency in $\omega = 2\pi f$ is incident normally on an infinitely wide thin wall some is reflected and some transmitted. Let the sound pressures of the incident, reflected and transmitted sounds be denoted by p_i, p_r and p_t, respectively, as shown in Fig. 1 (a). The wall is excited by the sound pressure difference between the two surfaces of the wall and the equation of motion is

$$(p_i + p_r) - p_t = m\frac{dv}{dt} \tag{1}$$

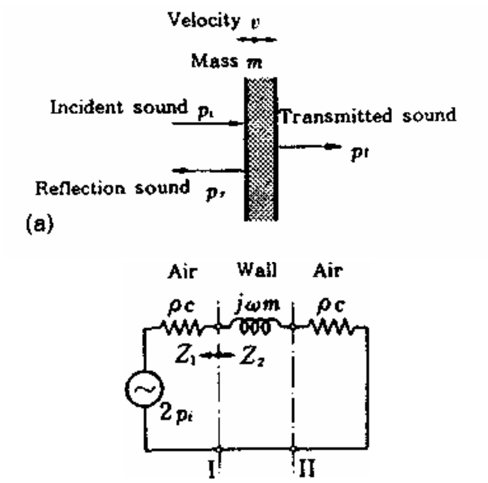
where m is the surface mass of the wall, and v is its velocity.

In the case of simple harmonic motion, $d/df\text{–}j\omega$ can be used

$$(p_i + p_r) - p_t = P = j\omega m v \tag{2}$$

$$\therefore \quad j\omega m = \frac{P}{v} \tag{3}$$

This is the impedance per unit area of the wall. Since it is assumed that the particle velocity of the air adjacent to both wall surfaces is equal to



(b)

Fig. 1. (a) **Sound insulation of single wall and (b) its analogous circuit**

$$\frac{p_i}{\rho c} - \frac{p_r}{\rho c} = \frac{p_t}{\rho c} = v$$

$$\therefore \quad p_i - p_r = p_t = \rho c v \tag{4}$$

From Eq. 2 and 4

$$\frac{p_i}{p_t} = 1 + \frac{j\omega m}{2\rho c}. \tag{5}$$

Therefore, the transmission loss is

$$R_0 = 10\log_{10}\frac{1}{\tau} = 10\log_{10}\left|\frac{p_i}{p_t}\right|^2 = 10\log_{10}\left\{1 + \left(\frac{\omega m}{2\rho c}\right)^2\right\}. \tag{6}$$

Generally, $(\omega m)^2 >> (2\rho c)^2$; hence

$$R_0 \approx 10\log_{10}\left(\frac{\omega m}{2\rho c}\right)^2 = 20\log_{10} f\cdot m - 43 \text{ dB} \tag{7}$$

which is proportional both to frequency and the surface mass m of the wall. This is called the 'mass law' for airborne sound insulation. Doubling the weight of the wall or the frequency gives an increase of 6 dB in ^Q– This wall motion may be expressed by an analogous electrical circuit as shown in Fig. 1(b). In this circuit, if no wall exists, it becomes as shown in Fig. 2 (a) and at the wall surface the sound pressure is p_i. When the wall is completely rigid in which the velocity becomes zero, the current is zero,

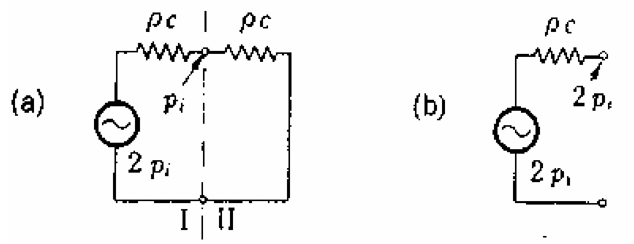


Fig. 2. **Analogous circuit: (a) Open air and (b) Rigid wall**

thus the circuit is open as shown in Fig.2 (b), and at the wall surfaces

$$p_i + p_r = 2p_i, \qquad \therefore \quad p_r = p_i .$$

In Fig. 1 (b), at the junction I of the circuit, the impedances to the left and the right are denoted by Z_1 and Z_2 respectively. Since there is no internal absorption, the transmission coefficient τ can be expressed by Eq. 1, where $Z_1 = \rho c$, $Z_2 = j\omega m + \rho c$, then Eq. 6 is obtained.

B. Random incidence mass law

When the incident angle is θ, Eq. 8 can be derived as follows:

$$R_\theta = 10\log_{10}\frac{1}{\tau_\theta} = 10\log_{10}\left\{1+\left(\frac{\omega m\cos\theta}{2\rho c}\right)^2\right\}. \quad (8)$$

If we are calculating the average value in the range $\theta = 0 \sim 90°$, Eq. 9 is obtained as the 'random incidence mass law'

$$R_{\text{random}} = R_0 - 10\log_{10}(0{\cdot}23/R_0). \quad (9)$$

However, in an actual sound field using the range of $\theta = 0 \sim 78°$, is more realistic and the following approximate formula is obtained:

$$R_{field} = R_0 - 5\text{dB}, \quad (10)$$

which is recognised as closer to reality and called the “field incidence mass law”.

2. Sound insulation of cylindrical housings

Industrial development and application of new technologies made it necessary to apply cylindrical shells and housings for noise reduction. The literature indicated [21] presents the application of different cylindrical housings and selection of their shapes. We shall discuss the sound insulation properties of universal cylindrical housings.

Whenever a question concerning sound-insulation of cylindrical housings comes forth, it is at once followed by another: what is the difference between a cylindrical housing and a cylindrical shell, sound-insulation of which has been studied by a good variety of authors? This question is not so simple, though some demarcation between them is possible.

When one speaks about housings, he has in mind certain sources, the noise of which is hindering. This noise may be considerably diminished, if the sources are surrounded by some sound insulation device – and that is a housing. We speak about sound insulation of a cylindrical shell in cases these sources cannot exist without a shell. For example, pressure fluctuations that are due to the fan in a ventilating system or the booster in a gas piping. Further on we shall keep to that position.

The first attempts to calculate and apply cylindrical housings were linked with the noise of extensive pipings, which sometimes reaches considerable values. Thus, for example, the noise of gas pipings from outside reaches 110–115dB. Computations [1] have shown that maximum noise is determined by discrete frequencies, linked to the blade frequency of the booster f_1, where n is the rotation frequency and z is the quantity of the booster blades, and with its harmonics.

Since the frequency is commonly below the first critical frequency of the piping and the damping of vibrations along it constituted only some decibels per 50m, it was clear that noise is caused by zero-order normal ("almost" plane) wave. Therefore quite a number of works appeared that were designed for elaboration of calculation methods and designing of the housing for reducing its noise [1,2,3,4,5,6]. In practice, this problem occurs rather frequently, thus we shall make more detail investigation into it. The essence of the problem is that during axially symmetric vibrations inside the piping the sound pressure can be written in the form of

$$p_1 = p_{10} J_0\left(\mu^2\right) e^{i\sqrt{k_1^2-\mu^2}}\, z , \quad (2.1)$$

where J_0 Bessel zero-order function, μ and $\gamma = \sqrt{k_1^2 - \mu^2}$ radial and axial wave numbers, $k_1 = \omega / c_1$, wave numbers in the medium inside the piping, and p_{10} is a complex amplitude of vibrations.

If we assume that impedance of the housing for normal axially symmetric vibrations is equal Z_0 [13], while the impedance of radiation into the environment is Z_{u31} [14,15], (then we shall come to dispersion control [14] in the form of impedance

$$Z_{10}\frac{J_0(\mu_1 a)}{J_1(\mu_1 a)} = Z_0 + Z_{u31} = Z_0 + Z_{20}\frac{H_0^{(1)}(\mu_2 a)}{H_1^{(1)}(\mu_2 a)}. \quad (2.2)$$

Here, $H_0^{(1)}$ and $H_1^{(1)}$ are the Hankel functions of the first kind of the zero and first order, a is the radius of the piping, and $Z_{10} = i\rho_1\omega/\mu_1$, $Z_{20} = i\rho_2\omega/\mu_2$, ρ_1 and ρ_2 is the density of mediums inside and outside the piping. Radial numbers in mediums outside and inside the piping arc linked together by the correlation

$$\mu_2^2 = k_2^2 - k_1^2 + \mu_1^2 , \quad (2.3)$$

where $k_2 = \omega / c_2$ – the wave number in the environment.

The roots of Eq. 2.2 determine the permissible normal waves inside the piping, which may exist. For the cases of similar and different mediums inside and outside at $n = 0$ they are calculated in works [14, 15].

Among them there is a zero root μ_{00}. It differs from the others since it is complex, very small according to the modules, i.e. $|\mu_{00}a| << 1$ and its imaginary part significantly exceeds the real one. With respect to this $\left|J_0\left(\mu^2\right)\right| \approx 1$ and (1) we can approximately write

$$p_1 \approx p_{10} l^{ik_1 z} \qquad (2.4)$$

In other words, this root corresponds to the "almost" plane wave, which propagates along *z*-axis of the piping.

In the first approximation, which is valid when

$$|Z_0| >> \left| Z_{20} \frac{H_0^{(1)}(\mu_2 a)}{H_2^{(1)}(\mu_2 a)} \right|,$$

$$\mu_{00} \approx i\sqrt{\frac{2i\rho_1\omega}{aZ_0}}\ . \qquad (2.5)$$

Here

$$Z_0 = i\omega m\left(\frac{f_n^2}{f^2} + \frac{f^2}{f_0^2} - 1\right), \qquad (2.6)$$

where $m = \rho h$, ρ and h are the density of the material and thickness of the shell, $f_n = c_n / 2\pi a$ is the longitudinal resonance frequency of the shell, on which the length of longitudinal wave is placed along the circumference of the shell, f_0 is the frequency of coincidence, at which length of a flexural wave in the shell λ_a equals to the wave length in the medium. It should be noted that z_0 never turns into zero, if $2f_n > f_0$.

Substituting Eq. 2.6 into Eq. 2.5 we shall obtain

$$\mu_{00} = i\sqrt{\frac{2\rho_1}{ma\left(\frac{f_n^2}{f^2} + \frac{f^2}{f_0^2} - 1\right)}}\ . \qquad (2.7)$$

Since μ_{00} is imaginary, $J_0(i|\mu_{00}|) = I_0(|\mu_{00}|)$, where I_0 is the modified Bessel's function. One more peculiarity of that wave is revealed. If the mediums inside and outside the shell are the same, then $\mu_2 = \mu_{00} = i|\mu_{00}|$. In the unbounded space the outside wave can be written in the form

$$p_2 = p_{20} H_0^{(1)}\left(\mu_{00}^2\right) e^{i\sqrt{k_1^2 - \mu_{00}^2}\, z} = \frac{2p_{20}}{\pi i} k_0\left(|\mu_{00}| z\right) e^{i\sqrt{k_1^2 + |\mu_{00}^2|}\, z},$$

where p_{20} is the pressure amplitude, k_0 is the McDonald's function (modified Hankel's function). It exponentially coincides with the increase in *r*. Thus, this wave does not radiate the sound outside and can spread along the pipe for extensive distances. The same will be also observed in the case the sound velocity in the internal medium c_1 will be less than that in the external medium c_2. Another picture will be observed when $c_2 > c_1$. In accordance with Eq. 2.3

$$\mu_2^2 = \frac{\omega^2}{c_2^2}\left(1 - \frac{c_2^2}{c_1^2}\right) - \mu_{00}^2 .$$

Beginning with the frequency $f_{00} = \dfrac{\mu_{00} c_2}{2\pi\sqrt{1 - \frac{c_2^2}{c_1^2}}}$, μ_2 becomes valid. Practically at the frequencies $f > (2 \div 3) f_{00}$, $\mu_2^2 \approx k_2^2 - k_1^2$ and from the pipe the wave will spread at the angle

$$\Theta \approx \arcsin(c_2 / c_1) \qquad (2.8)$$

to *r* axis.

Analogous situation occurs in the case of similar mediums, if the medium inside moves. Then the velocity of sound propagation in the direction of movement will be greater than the sound velocity in the motionless medium.

In [2, 6] the problem of sound insulation of the zero wave with the help of a sound-measuring housing, represented in the form of the coaxially arranged cylindrical shell, is considered in detail. The piping of radius and the housing of radius are considered infinitely extended in length. Sound pressure

$$p_3^1(r,z) = p_{30}^1 H_0^{(1)}(\mu_3 r) e^{i\sqrt{k_2^2 - \mu_3^2}\, z},$$

which is created by the piping without the housing at some point (r, z), is obtained by a usual method by satisfying the boundary conditions when $r = a_1$, expressing the equation of the radial velocities of the piping v_0 and the medium.

When determining the sound pressure $p_3(r,z)$, which is created by the piping with the housing, the following assumptions were made:

1. A thin housing, i.e. radial oscillation velocities of its surfaces inside and outside are similar.
2. Radial velocity of the piping v_0 does not change if the housing is installed.

Sound pressures below and behind the housing are written in the form

$$\left.\begin{aligned} p_2(r,z) &= \left[A H_0^{(1)}(\mu_2 r) + B H_0^{(2)}(\mu_2 r)\right] e^{i\sqrt{k_2^2 - \mu_2^2}\, z},\\ p_3(r,z) &= p_{30} H_0^{(1)}(\mu_3 r) e^{i\sqrt{k_3^2 - \mu_3^2}\, z}. \end{aligned}\right\} \qquad (2.9)$$

The unknown amplitudes *A*, *B* and p_{30} are defined by the boundary conditions in the piping (the equation of the radial velocity of the medium of the velocity v_0) and in the housing (the equation of the radial velocities and the equation of motion for a cylindrical shell in the form of the impedance). The sound insulation *R* is defined in respect to pressure amplitudes at some point *(r, z),* created by the piping with and without the housing:

$$R = 10\lg\left|\frac{p_3'}{p_3}\right|^2 . \qquad (2.10)$$

In Eq. 2.6 a case when mediums inside and outside the housing are similar, i.e. $k_2 = k_3$ and $\mu_2 = \mu_3$, $\rho_2 = \rho_3$ is presented. It was obtained that

$$R = 20\lg\left|1-\frac{Z_{\text{об}2}\pi\,\mu_2^2 a_2}{4\rho_2\omega}H_1^{(1)}(\mu_2 a_2)H_1^{(2)}(\mu_2 a_2)\left[\frac{H_1^{(1)}(\mu_2 a_2)H_1^{(2)}(\mu_2 a_1)}{H_1^{(2)}(\mu_2 a_2)H_1^{(1)}(\mu_2 a_1)}-1\right]\right|, \qquad (2.11)$$

where $Z_{\text{об}2}$ – the impedance of the shell (housing) for axially symmetric vibrations. It should be noted that the expression Eq. 2.11 holds true for any normal axially symmetric waves, not only for the zero wave. Everything is defined by selecting the corresponding values of the radial wave number μ_2 and by correctly expressing the impedance Z_{02} for a normal wave under consideration. The work [3] gives a description of a practical method for calculating sound insulation of the housing of the zero wave. Diagrams of the amplitudes and phases of the Hankel function, significantly simplifying computation, are provided.

It is shown that minimum sound insulation is observed in two cases:

1. The impedance of the shell approaches zero in cases of frequencies close to the critical frequency $f_n = c_n / 2\pi a_2$, at which one longitudinal wavelength $\lambda_n = c_n / f$, will be equal to the circumference of the housing and to the coincidence frequency f_{02} on which the flexural wavelength in the housing equals the wavelength in the medium λ_c.
2. On the resonances of the medium, at the clearance between the piping and the housing.

Since the dimensions of the clearance are not large, these resonance frequencies, as a rule, lie higher than the frequency f_n.

Here is presented a typical curve of dependence R upon the frequency, calculated for the aluminium housing h = 0,1 cm in thickness for the piping with radius $a_1 = 36$ cm and the clearance between the piping and the housing $d = 5$ cm (see Fig. 2.1).

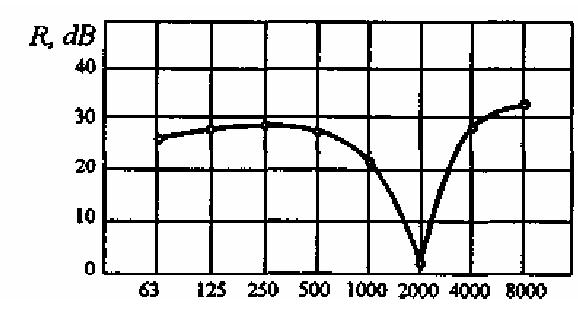


Fig. 2.1. **Sound insulation frequency dependence of a housing**

In [7] a case is presented when the media inside and outside the housing are different. It is of importance for practice since the application of the soundproofing materials at the clearance under the housing may greatly increase sound insulation of the housing at the frequencies where it has minimum values. It has been shown that at the optimum clearance $d \approx 5$ cm a simplified equation may be used for assessing the minimum sound insulation at the resonance frequencies:

$$R_{\min} = R_{КОЖ} + 8{,}7\beta_m d\,, \qquad (2.12)$$

where β_m is the attenuation coefficient of the sound-absorbing material.

In [8, 9] computation data of frequency dependencies of sound insulation R of steel and aluminium housings at different clearances d between the piping and the housing are provided. It follows that optimum values d are from 5 to 6 cm, and the optimum value of housing thickness h=1–1,5 mm.

In [12] another case is presented which is of practical interest for noise decreasing with the help of the housing, when the extended piping performs beam-like vibrations. They occur frequently, for example, at vibrations of water-supply pipes. The computation is based on an approximate equation of housing oscillations from [17], which presents an equation of flexural vibrations of the beam with a correction for rotational inertia. In the same manner as previously an expression for R type (Eq. 2.11) is obtained. All conclusions earlier also hold true in this case. Minimums of sound insulation are observed at the coincidence angles of the housing and at the resonance frequencies of the medium at the clearance between the piping and the housing.

Analogous expressions (Eq. 2.11) for sound insulation are also obtained for other normal waves with azimuthal numbers not equal to zero. In work [10] sound insulation for different n can be written in the form

$$R = 20\lg\left|1-\frac{Z_{\text{об}2}\pi\,\mu^2 a_2}{4\omega\rho}\dot{H}_n^{(1)}(\mu a_2)\dot{H}_n^{(2)}(\mu a_2)\left[\frac{\dot{H}_n^{(1)}(\mu a_2)\dot{H}_n^{(2)}(\mu a_1)}{H_n^{(2)}(\mu a_2)\dot{H}_n^{(1)}(\mu a_1)}-1\right]\right| \qquad (2.13)$$

where $H_n^{(1)}$ and $H_n^{(2)}$ are the Hankel functions of n-th order of first and second kind, a point above the Hankel functions means an argument derivative. Minimum sound insulation is obtained on the same conditions as for a zero wave: at the small values of the impedance $Z_{\text{об}2}$ and at the appearance sound waves between the piping and the housing.

In [11] an expression is derived for sound insulation of cylindrical shells when excitation and sound pressure depend only on r and φ (with no wave propagation along z-axis). As to its form, it almost doesn't differ from Eq.2.11 and Eq. 2.13. The impedance of the housing used in it is written in the form [16]*

$$Z_{\text{об}2} = \frac{k_1^4 a_2^4 - k_2^2 a^2 (n+1) - k_1^2 a^2 n^2 \beta - n^4 \beta + n^6 \beta}{j\omega\gamma\left(n^2 - k_2^2 a^2\right)}, \qquad (2.14)$$

* Note. In these cases when a temporary factor is used an imaginary unit is written as.

where $k_2 = \omega / c_2$ is the wave number of longitudinal vibrations in the material of the shell, $\beta = h^2 / Ra_2^2$, $\gamma = a_2^2 / Eh\left(1-\sigma^2\right)$, E and σ are the Young's modulus and Poisson's ratio of the material. Computation curves of sound insulation of the brass shell, 1 mm in thickness, radius $a_2 = 7{,}5$ cm (an inner pipe has radius $a_1 = 5$ cm) for normal waves with n = 2, 3, 4, 5 are provided. To confirm computations measurements of brass pipes, the ends of which were produced soundproof, were conducted in the room with small reflections. The experiment showed that at critical frequencies a decrease in a sound insulation up to – 10 dB was observed (i.e. sound on these frequencies intensifies), as it was predicted by theory. In addition, it was noticed that critical frequencies are shifted to the direction of the higher frequencies.

In the above works infinite cylindrical shells were considered. For practical purposes it will be of special interest to study the effect of the limited dimensions both of the sources and of the housings on sound insulation. Lately quite a number of works on these problems appeared in the literature.

In [18] a problem on sound insulation of the cylindrical shell of length l from the extended sound sources is being solved. The source is represented in the form of a cylinder with radius a_0 and is located normally to the two rigid parallel walls. It is considered that arbitrary distribution of radial velocities are present on its surface. The housing of radius a_k is located coaxially with the sound source. To simplify the solution of the problem boundary conditions of securing the source and the housing on the butt ends are thus selected that eigenfunctions along z-axis will be the same in the medium and in shells, i.e. $\psi_m(z) = \cos k_m z$, where $k_m = m\pi / l$ and $m = 0, 1, \ldots$ – the integral number. Then the pressure expansion is possible in the medium inside and outside the housing, as well as that of the radial velocities of the source and the housing in series according to $\psi_m(z)$ and, having satisfied boundary conditions, it is possible to obtain the unknown quantities of pressure amplitudes. Sound insulation of the shell for normal waves in that case is written as

$$R = 20\lg\left|1 - \frac{\pi a_k\left(k_2^2 - k_m^2\right)\cdot z_{mn}}{4\rho_2\omega}\frac{\dot{H}_n^{(1)}(a_k)}{\dot{H}_n^{(1)}(a_0)}\left[\dot{H}_n^{(1)}\dot{H}_n^{(2)}(a_0) - \dot{H}_n^{(2)}\dot{H}_n^{(1)}(a_0)\right]\right| \quad (2.15)$$

Here, $k_2 = \omega / c_2$ – the wave number in the medium, $k_{rm} = \sqrt{k_2^2 - k_m^2}$ – the radial wave in the medium, arguments of derived Hankel functions $\dot{H}_n$ and $\dot{H}_n(a_0)$ are equal correspondingly $(k_{rm}a_k)$ and $(k_{rm}a_0)$. The impedance of the shell for a normal wave *(m, n)* is written as

$$Z_{mn} = -i\omega m_0\left[1 - \frac{k_m^2 + \frac{n^2}{a_k^2}}{k_4^4} - \frac{1}{k_n^2 a_n^2}\frac{\left(k_m^2 - k_t^2\right)\left(k_m^2 - k_n^2\right) - k_n^2\frac{n^2}{a_k^2}}{\left(k_m^2 + \frac{n^2}{a_k^2} - k_0^2\right)\left(k_m^2 + \frac{n^2}{a_k^2} - k_t^2\right)}\right], \quad (2.16)$$

where $k_t = \omega / c_t$, $k_0 = \omega / c_0$, $k_n = \omega / c_n$, $k_4 = \sqrt[4]{\omega^2 m_0 / B}$ – the wave number of flexural vibrations in the plate,

$$B = Eh^3 / 12\left(1 - \sigma^2\right)$$

– flexural rigidity of the plate, from which the shell is made, E and σ – Young's modulus and Poisson's ratio of the material, c_t, c_n and c_0 – the velocities of shear and longitudinal wave propagation (in the rod and in the plate, correspondingly). The velocities c_n and c_0 are linked with the relation $c_n^2 = c_0^2\left(1 - \sigma^2\right)$.

The relation (2.15) as to its type is fully identical with (2.11) and (2.13). The difference consists only in different representation of radial and axial wave numbers in the expressions for and the impedances of the shells. As a result, the previously drawn conclusions are also valid here: minimums of sound insulation are observed if the impedance $Z_{mn} = 0$ and at the resonance's of the medium layer under the housing. At frequencies in which $\left(k_b^2 - k_n^2\right)$ does not lead to $R_{mn} = 0$. The value of sound insulation at these frequencies $f_m = mc_2 / 2l\, l$ equals

$$R_{mn} = 20\lg\left|1 + i\frac{Z_{mn}n\left[1 - \left(\frac{a_0}{a_k}\right)^{2n}\right]}{2\rho_2\omega a_k}\right|, \quad (2.17)$$

i.e. determined by the value Z_{mn}.

Note should be taken of the peculiarity R_{mn}; in addition to a wave with m = 0, which propagates along r-axis at all frequencies, each normal wave has a critical frequency f_m, below which there is no propagation of the m-th wave. It attenuates exponentially along r. Sound insulation then has great values and no minimums, linked with the medium resonance's at the clearance under the housing. Before the frequency $f_1 = c_2 / 2l$ only a wave with m = 0 propagates. If the parameters of the housing are selected such that it will possess the good sound insulation of a zero wave, then the housing will decrease effectively the noise at low frequencies.

In the present work sound insulation using various methods for excitation is considered, as well as the computation R results for the shell with $l = 3$ m, $a_2 = 0{,}3$ m and $h = 2$ mm (Fig. 2.2) when n = 0 and a small radius of the source a_0 as compared with the wavelength in the air are provided. Here, along abscissa axis the unlimited frequency $\alpha = f / f_0$ is plotted, where $f_0 = c_2 / 2\pi a_k$. The diagram shows that curves differ significantly only in the area of the critical frequency. Higher they merge together.

In /19/ an analogous problem is considered, but the sound source is located outside the casing, while the point at which the noise decreases is inside. The solution is found the same as in [18], only for transferring the beginning of the coordinates the theorem of addition for cylindrical functions is used.

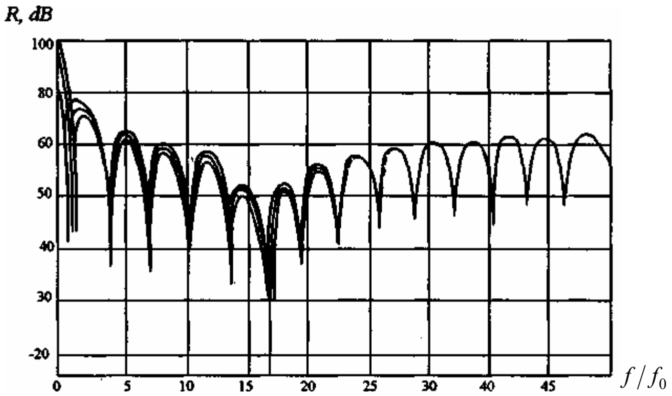


Fig. 2.*2*. **Sound insulation frequency characteristic of the first three modes**

The general expressions for sound insulation using various methods of excitation are obtained. It was shown that *R* depends on the position of the observation point. If it is located on the axis of the housing, then at low frequencies at $f < f_1 = c_2 / 2\pi a_k$ sound insulation from the external and internal sources is similar.

This conclusion also holds true at the concentrated excitation, when a small surface of the radiator emits and its radius a_0 is small.

In [20] the problem on sound insulation, restricted by the rigid shell, is solved. The radiator with radius a_0 in it represents part of the infinite rigid cylinder of length *e,* on which an arbitrary distribution of radial velocities is preset. The housing represents the elastic shell of length *e,* which is also part of the infinite rigid cylinder of radius a_k. The medium, characterized by the plane ρ_2 and sound velocity c_2, is inside and outside the housing, The side walls at the clearance between the radiator and the housing at $z = 0$ and $z = e$ are considered rigid, in consequence of which eigenfunctions $\psi_m(z) = \cos k_m z$, where $k_m = m\pi / e$ and $m = 0, 1, \ldots$ – the integral number. As in /18, 19/ boundary conditions of securing the shell and the radiator at $z = 0$ and $z = e$ are such that the eigenfunctions of radial displacement of the housing and the radiator are also $\psi_m(z)$. Making use of plotting along $\psi_m(z)$ of the radial velocities of the radiator V_0, the housing V_k and the sound field at the clearance $P_r(r, \varphi, z)$, it is possible to find the link between the amplitudes of normal waves V_{kmn} and V_{0mn}.

Sound pressures, formed by the radiator and the housing in the unlimited space are written in the form of Fourier integrals:

$$p_0 = \sum_n e^{in\varphi} \int_{-\infty}^{\infty} \tilde{p}_0(k) H_n^{(1)}\left(\sqrt{k_2^2 - k_m^2}\, r\right) e^{ikz} dz$$

$$p_k = \sum_n e^{in\varphi} \int_{-\infty}^{\infty} \tilde{p}_k(k) H_n^{(1)}\left(\sqrt{k_2^2 - k_m^2}\, r\right) e^{ikz} dz$$

From the boundary conditions at $r = a_0$ and $r = a_k$ the relation between $\tilde{p}_0(k)$, $\tilde{p}_k(k)$ and V_{0mn} of the radiator is determined. Integrals in the expressions $\tilde{p}_0$ and $\tilde{p}_k$ are assessed by the method of passage.

As a result expressions for sound insulation at arbitrary excitation are obtained. Radiators in the form of the pulsating cylinder and the concentrated source of the small dimension were considered separately.

It is shown that in the case of the pulsating cylinder $R = R_0 + \Delta R$, where R_0 – the known sound insulation of the infinite cylindrical shell, which performs the radial vibrations. The second part ΔR contains the dependence on the angle Θ between the plane $z = 0$ and the direction at the point of observance. A diagram is provided for computing.

In the case of the concentrated excitation sound insulation at $\Theta = 0$ ($z = 0$ in the plane) is by 6 dB lower than R_0.

In /23/ a problem of somewhat another type is considered, but the method of its solution and the results obtained are similar.

The source of finite dimensions is located on the rigid base. For decreasing its noise a semi-cylindrical shell is used. To simplify the solution of the problem it is taken that the source is part of the rigid semi-cylinder with its length *e.* The housing is located coaxially with the source and is unlimited.

In the absence of the housing the sound pressure, which is formed by the radiator, is located as in [20], i.e. in the case of the source in the rigid screen. The field outside the housing is found analogously. The integrals obtained are estimated by the method of passage.

The expression for sound insulation of the n^{th} normal wave coincides with (2.11). The difference is only that the radial wave number μ is replaced with $K_2 \cos\Theta$, where Θ – the angle between the plane $z = 0$ and the direction at the point of observance. All inferences and methods of computation, elaborated for determining sound insulation of the closed infinite cylindrical shell for normal waves, may supplement the problem under investigation.

Conclusions

By means of cylindrical form constructions (housings) it is easier to isolate low frequencies, as Weight Law of construction surface is circumvented. In this case, the main role is played by rigidity of the construction.

Sound insulation of flat constructions increases proportionally to the effected sound frequency. At high frequencies the maximum of sound insulation of flat constructions (except resonances) is achieved

Meanwhile, sound insulation of cylindrical housings (constructions) at low frequencies is rather high and remains equal over the whole audible sound frequency range except resonance display.

The foundation of slab (partition) sound insulation theory was laid by L.Cremer [21] what led to the development of present slab sound theory and practice.

As it is seen from the given solutions, the slab sound insulation depends on the Mass Law and sound insulated by frequency. It should be mentioned that slab sound insulation greatly depends on sound speed *cf* as well as on propagation velocity *ci* of bending waves in a slab. The coincidence of the mentioned speeds leads to wave coincidence which is known as critical frequency. At these frequencies which are most common at the range of high frequency, sound insulation may be reduced to zero.

The theory of cylindrical shells and housings was investigated later.

References

1. **Gužas D.** Methods of Noise Reduction Radiated by Elements of the Gas Supply System. Candidate's Thesis. Moscow, 1978. P. 261 (In Russian).
2. **Gužas D.** Sound insulation of housing for gas-main // Proceedings of Higher Institutions. Machine-building. 1977. Vol. 11. P. 70–74. (In Russian).
3. **Gužas D.** Method of sound insulation calculation of cylindrical housing // Proceedings of Higher Institutions. Machine-building. 1979. VoL 11. P. 92–94. (In Russian).
4. **Gužas D.** Noise reduction of piping by sound insulation // Fighting against Noise and Fire Safety at Compressor Stations of Gas-Spatial Enterprises of the Ministry of Gas Industry. Moscow, 1977. P. 43–49. (In Russian).
5. **Gužas D.** Sound insulation calculation of cylindrical housing by designing compressor stations // Materials of School of Progressive Experience at the Exhibition of Economic Achievements. Moscow, 1985. P. 84–94. (In Russian).
6. **Gužas D.** Sound insulation of coaxial cylindrical housing // Fighting against Noise. Proceedings of 3rd All-Union Conference on Fighting against Noise and Vibration. Chelyabinsk, 1980. P. 143–146. (In Russian).
7. **Gužas D.** Sound insulation of coaxial cylindrical housing with different medium on both sides of the housing // Proceedings of Higher Institutions of the USSR. Machine-building. 1982. Vol. 12. P.70–74. (In Russian).
8. **Gužas D.** Fighting against piping noise by method of sound insulation // Fighting against Noise and Harmful Vibrations in Construction. Leningrad, 1982. P. 25–28. (In Russian).
9. **Gužas D.** Noise reduction of piping by sound insulation // Gas Industry. 1979. Vol. 8. P. 32–33. (In Russian).
10. **Leskov E. F., Mogilevski M. I.** Calculation of sound insulation of cylindrical housing // Investigations into Building Acoustics. Collection of Works of NIISF GOSSTROY SSR. Moscow, 1981. P. 119–123. (In Russian).
11. **Heckl M., Heitkamper W.** Zum schalldurchgang Durch Doppelwandige Zylinder. Fortschritte der Akustik – DAGA'85. S. 365–388.
12. **Gužas D.** Sound insulating housings by beam oscillations of pipings // Vibrotechnica. 1987. Vol. 2 (59). P.135–141. (In Russian).
13. **Gužas D.** Normal impedance of thin-walled cylindrical shell // Vibration Engineering. 1987. P. 133–140.
14. **Gužas D.** On dispersion equation roots in problem of gas fluctuations in cylindrical shell // Proceedings of Higher Institutions of the USSR. Machine-building. 1981. Vol. 7. P. 56–61. (In Russian).
15. **Gužas D.** Roots of dispersion equations in a problem of gas fluctuation in cylindrical shell // Proceedings of Higher Institutions of the USSR. Machine-building. 1981. Vol. 9. P. 58–61. (In Russian).
16. **Heckl M.** Vibration of point-driven cylindrical shells // JASA. Vol.34. 1962. P. 1553.
17. **Kornecki A. A.** Note on beam-type vibration of circular cylindrical shell // J. Sound and Vibration. 1971. Vol. 14. P. 1–6.
18. **Gužas D.** Sound insulation of cylindrical shell in limited space with reflecting walls // Collection of Works: Noise Protection in Buildings and on Construction Sites. Moscow: NIISF GOSSTROY USSR, 1987. P. 35-39. (In Russian).
19. **Gužas D.** Sound insulation of cylindrical shell in limited space from outside noise source // Vibrotechnica. 1989. Vol. 1 (62). P. 7–13. (In Russian).
20. **Gužas D.** Sound insulated of limited cylindrical shell in rigid screen // Inter-noise 88 "The Sources of Noise". Vol. 1. P. 301–305.
21. **Cremer L.** Theorie der Luftschalldämmung Zylindrischer Schallen. Acustica, 1955, Vol. 5.

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Statybinių ir technologinių konstrukcijų garso izoliacijos ypatumai

Reziumė

Straipsnyje išanalizuoti statybinių ir technologinių orinio garso izoliacijai skirtų konstrukcijų ypatumai. Primenami garso izoliacijos dėsniai ir galimybės, kaip pakeisti nurodytas garso izoliacijos savybes. Trumpai paliesti plokščių konstrukcijų orinio garso izoliacijos gerinimo klausimai. Plačiau aptariami cilindrinių gaubtų garso izoliacijos skirtumai nuo plokščių (pertvarų) garso izoliacijos. Cilindrinių formų konstrukcijomis lengviau izoliuoti žemuosius dažnius, nes apeinamas konstrukcijos paviršiaus svorio dėsnis. Čia pagrindinį vaidmenį vaidina konstrukcijos standumas.

Išvadoje pažymima, kad plokščių konstrukcijų garso izoliacija didėja proporcingai veikiamo garso dažniui. Esant aukštiems dažniams pasiekiamas plokščios konstrukcijos garso izoliacijos (išskyrus rezonansus) maksimumas.

Tuo tarpu cilindrinių gaubtų (konstrukcijų) garso izoliacija, esant žemiems dažniams yra gana didelė ir išlieka tolygi visame girdimame garso dažnių diapazone, išskyrus rezonansinius efektus.

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