Research Article

Finite-Element Method for Calculating the Sound Field in a Tank with Impedance Boundaries

Qi Li,1,2,3 Junhua Xing,1,2,3 Rui Tang,1,2,3 and Yiming Zhang1,2,3

1Acoustic Science and Technology Laboratory, Harbin Engineering University, Harbin 150001, China
2Key Laboratory of Marine Information Acquisition and Security (Harbin Engineering University), Ministry of Industry and Information Technology, Beijing, China
3College of Underwater Acoustic Engineering, Harbin Engineering University, Harbin 150001, China

Correspondence should be addressed to Rui Tang; tangrui@hrbeu.edu.cn

Received 19 July 2019; Revised 8 January 2020; Accepted 3 March 2020; Published 10 April 2020

Academic Editor: Fotios Georgiades

Copyright ©2020 Qi Li et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

In this paper, a finite-element method for calculating the sound field in a water tank with impedance boundaries is proposed based on the theory of standing waves in a tube. The equivalent acoustic impedance of the tank walls is calculated by establishing a three-dimensional axisymmetric virtual standing-wave tube in finite-element software, whereupon boundaries with that impedance are used as the tank boundaries. Since the impedance is the property of the material itself, the calculated impedance value can be used for the calculation of the three-dimensional sound field. The sound field due to a point source in a glass tank is calculated using the proposed method, the correctness of which is assessed experimentally. By comparing the experimental and numerical results, the proposed method is shown to be correct.

1. Introduction

Sound field in a rectangular enclosed space is an important and classical problem in acoustics and has applications to office buildings, production facilities, cars, ship cabins, and reverberant rooms, among others. Understanding this problem helps with predicting sound fields and controlling noise. In applications, it is usually more practical to consider a rectangular enclosed space with general impedance boundary conditions than one with ideal boundaries, but the former approach is yet to be studied fully in the field of acoustics. A nonanechoic tank is an important acoustic measuring device that is used widely for calibrating transducers [1, 2] and measuring sound power [3, 4] and the sound absorption coefficient of underwater acoustic materials [5, 6]. Being able to predict the sound field in a nonanechoic tank reasonably and effectively will help understand the sound field characteristics and provide the necessary basis for acoustic measurements using nonanechoic tanks.

The accuracy of predicting the sound field in a nonanechoic tank depends largely on the correctness with which the boundaries of the sound field are modeled. Compared with an ideal boundary, an impedance boundary is more in line with an actual sound field. An impedance boundary combines both the attenuation of the energy amplitude and the phase change of the response, and it has obvious advantages compared with an absorption boundary [7].

In 1944, Morse and Bolt [8] established a theoretical model of the sound field in a rectangular room. By combining sound pressure modes and impedance boundaries, they derived a nonlinear transcendental characteristic equation. When the impedance values of the sound-field boundaries are the same, the well-known Morse table is obtained by solving that characteristic equation. Maa [9] deduced the nonlinear transcendental characteristic equation of the sound field in a rectangular enclosed space when the impedance values of the sound-field boundaries differ. Because that characteristic equation has no analytical solution, it can be solved only numerically.
2 Mathematical Problems in Engineering

Classical numerical methods for solving nonlinear equations (e.g., Newton iteration) are not suitable for solving the aforementioned equations because they yield only one root for a single initial guess. Naka et al. [10] proposed an interval Newton/generalized bisection method to help solve such nonlinear equations. Bistafa and Morrissey [11] proposed a new numerical method in which the eigenvalue problem is posed as one of homotopic continuation from a nonphysical reference configuration in which all eigenvalues are known and obvious. Du et al. [12] proposed using Fourier series to analyze the acoustic field in a rectangular cavity with general impedance boundary conditions, thereby transforming the problem of finding the characteristic root of a nonlinear transcendental equation into that of finding a standard matrix eigenvalue. However, the aforementioned approaches require the sound field to be rectangular. Instead, Xie et al. [13] proposed an approach based on a weak variational principle to study the sound field inside an acoustic enclosure whose walls have arbitrary inclinations and impedance conditions.

Because the physical properties of air and water are quite different, the acoustic characteristics of a nonanechoic tank are quite different from those of a reverberation chamber in air [14]. It is more complicated to analyze the sound field in a water tank with impedance boundaries. To calculate an underwater sound field with impedance boundaries, Stotts and Koch [15] proposed a two-way coupled modal method that satisfies energy conservation.

Finite element method is one of the widely used numerical analysis techniques. Because it can be applied to almost all the continuous medium problems or field problems, the finite element method has attracted great attention in various fields of physics including acoustics. Thanks to the advances in computer technology, the acoustic finite element method is nowadays widely used to solve the acoustic problems. Vorlander [16] investigated the concepts and uncertainties of computer simulations in room acoustics, and he believes that the reliability depends on the skills of choosing the correct input data of boundary conditions such as absorption and scattering. Aretz and Vorlander [17] investigates the influence of different boundary representations of porous absorbers on the simulated sound field in small rooms. They analyzed the sound field of a scale model room with a well-defined geometry through experimental measurements and finite element methods. Naka et al. [18] combined the geometric methods and finite element method to calculate the sound field in rooms with realistic impedance boundary conditions.

All the methods mentioned above require the room to have a regular shape. In the present paper, a method is presented for using acoustic software to calculate an impedance-boundary sound field. The advantages of this method are that it is not limited by the shape of the sound field during the calculation and the impedance value can be arbitrary. There is no specific requirement for the material of the tank wall. As long as the parameters of the material are known, the sound field prediction can be performed using this method. This method uses the principle of a standing-wave tube (SWT) to measure the acoustic impedance of materials, and a virtual SWT is established in software to obtain the impedance value at the boundary of the sound field. With the obtained impedance value, the sound field in a water tank with the same impedance boundaries can be calculated. We use the method proposed herein to calculate the sound field due to a point source in a glass water tank. By comparing the numerical results with experimental results, we verify the accuracy of the method. We then calculate and compare the sound fields in a large nonanechoic pool with different impedance values and analyze how the boundary impedance values influence the sound field.

2. Sound Field with Impedance Boundaries

2.1. Acoustic Description of Rectangular Enclosed Space with Impedance Boundaries. A rectangular enclosed space of dimensions $L_x \times L_y \times L_z$ and the associated coordinate system are shown in Figure 1. For an enclosed space with uniform impedance on each of its walls, the Helmholtz equation can be written as

$$-\nabla^2 p - k^2 p = 0,$$

where $k = (2\pi f/c)$ is the driving wave number with frequency $f$ and $c$ is the speed of sound. The Helmholtz equation can be solved by separating the variables. Because the solutions in different directions have the same form, we take the $x$ direction as the example here. The eigenfunction in the $x$ direction can be written as

$$p_x = Ae^{ik_x x} + Be^{-ik_x x},$$

where $k_x$ is the wave number in the $x$ direction and $A$ and $B$ are constants. The boundary conditions in the $x$ direction can be written as

$$\begin{cases}
\nabla \cdot \overrightarrow{n} = -i \frac{\rho c}{Z_{x=0}} k p, \\
\nabla \cdot \overrightarrow{n} = -i \frac{\rho c}{Z_{x=L_x}} k p,
\end{cases}$$

where $\overrightarrow{n}$ is the outward normal unit vector on the walls, $c$ is the density of the medium, and $Z$ is the impedance of the walls. Substituting equation (2) into equation (3) gives

$$\begin{cases}
A\left(k_x - k \frac{\rho c}{Z_{x=0}}\right) - B\left(k_x + k \frac{\rho c}{Z_{x=0}}\right) = 0, \\
A\left(k_x + k \frac{\rho c}{Z_{x=L_x}}\right) e^{ik_x L_x} - B\left(k_x - k \frac{\rho c}{Z_{x=L_x}}\right) e^{-ik_x L_x} = 0.
\end{cases}$$

For equation (4) to have a nonzero solution, the coefficient determinant must equal zero, giving

$$e^{2ik_x L_x} - \frac{(k_x - (\rho c/Z_{x=0})k)(k_x - (\rho c/Z_{x=L_x})k)}{(k_x + (\rho c/Z_{x=0})k)(k_x + (\rho c/Z_{x=L_x})k)} = 0.$$


2.2. Standing-Wave Tube Theory. A wave tube is a common device with which the acoustic parameters of materials are measured. The sound absorption coefficient and acoustic impedance of the acoustic material can be measured by using the standing-wave ratio method and the transfer-function method in the wave tube. These methods are standard test methods for which international standards have been established [19, 20]. A tube of diameter $D$ is shown schematically in Figure 2, and an acoustic material with a normal acoustic impedance of $Z_a = R_a + X_a j$ (or a normal acoustic impedance ratio $R_a$) is mounted at the end of the tube. When the tube wall is rigid, plane waves will be generated in the tube below the cutoff frequency $f_0$, which is calculated by

$$f_0 = \frac{1.84 \cdot c}{\pi \cdot D}$$

where $c$ is the sound speed in air.

The sound field in the tube can be expressed as

$$\vec{p} = p_i + p_r = p_{ai} e^{i(\omega - kx)} + p_{ar} e^{i(\omega + kx)},$$

where $p_i$ is the incident wave and $p_r$ is the reflected wave. The reflected wave is caused by the acoustical material load at the tube end and not only differs in size from the incident wave but also may have a different phase. The relationship between $p_{ai}$ and $p_{ar}$ can be expressed as

$$\frac{p_r}{p_{ai}} = r_p = |r_p| e^{i\phi},$$

where $r_p$ is the sound pressure reflection coefficient and $\phi$ is the phase difference. Substituting equation (11) into equation (10) gives

$$P = p_{ai} \left( e^{-jkx} + |r_p| e^{j(kx+\phi)} \right) e^{j\omega t}.$$  

From equation (12), the velocity in the tube can be obtained as

$$v = \frac{p_{ai}}{\rho c} \left( e^{-jkx} + |r_p| e^{j(kx+\phi)} \right) e^{j\omega t}.$$  

At the interface $x = 0$ between the medium and the material, the acoustic impedance can be obtained as

$$Z_a = R_a + X_a j = \frac{2}{v} \left( \frac{1 + |r_p| e^{j\phi}}{1 - |r_p| e^{j\phi}} \right) = \rho c.$$  

Therefore, the impedance of the acoustic material can be obtained when the sound field in the tube is known, and the sound field can also be calculated from the impedance of the material.

3. Finite-Element Solution

Acoustic finite element applications are nowadays widely used to predict the sound field in enclosed cavities and water tanks. There is many mature commercial acoustic finite element software with high computing efficiency and accuracy. When using acoustic finite element software to calculate the sound field in a closed space, the difficulty is an accurate description of the boundary conditions. The proposed numerical method for calculating the sound field in a water tank can be divided into two main steps. The first step is to use software to build a virtual SWT and calculate the acoustic impedance of the material of the tank walls. The second step is to calculate the sound field using the obtained impedance value. The acoustic calculation software used...
here is Actran. The sound field of a glass tank with 1.474 m inner length, 0.9 m inner width, 0.013 m thickness, and 0.6 m depth is calculated using this method. The proposed method can not only calculate the sound field of glass tanks but also be used for calculation as long as the material parameters of the tank wall are known.

3.1. Sound Field in Standing-Wave Tube. Although it is complicated to calculate a three-dimensional sound field with impedance boundaries, the sound field in an SWT with an impedance boundary at the end of the tube is easy to calculate. To verify the ability of the finite-element (FE) software Actran and to calculate a sound field with impedance boundaries, the sound field in an SWT with an impedance boundary at the end of the tube is calculated by the FE method and analytically.

The analytical calculation is based on Section 2.2, with the impedance value set to $Z_s = 1.5 \times 10^6 + 1.5 \times 10^6 j$. The FE calculation model is shown in Figure 3, for which a tube with a rigid wall is built. The tube is 1 m in length and 0.1 m in diameter. The sound velocity of the water is 1500 m/s and the density is 1000 kg/m$^3$. A velocity source of 1 m/s was added to one end of the virtual SWT as a sound source. The size of the acoustic grid is 0.05 meters, which can satisfy more than 10 grids per wavelength at 2000 Hz. A velocity source of 1 m/s is imposed at one end of the tube, and an impedance boundary with an impedance value of $Z_s = 1.5 \times 10^6 + 1.5 \times 10^6 j$ is imposed at the other end. Field points are placed in the tube to sample the sound pressure.

The sound field in the tube at 2,000 Hz is calculated, and the results are shown in Figure 4. Because the dimensional amplitudes of the sound field differ between the analytical and numerical calculations, the results are normalized by dividing by the maximum value. The origin of the coordinate system is at the impedance boundary.

As shown in Figure 4, the normalized results of the analytical and numerical calculations are consistent. Thus, the accuracy of using the FE software Actran to calculate a sound field with an impedance boundary is verified.

3.2. Obtaining Material Impedance Using Virtual Standing-Wave Tube. When using impedance boundaries to simulate the boundaries of a nonanechoic tank, the specific value of the impedance should first be determined. Based on the actual situation, a three-dimensional axisymmetric virtual SWT is established in Actran, as shown schematically in Figure 5. Because the calculation model is axisymmetric, a two-dimensional rotation model is established to save computing time. The length of the water-filled virtual SWT is 5 m, from equation (7), the diameter of the virtual SWT is selected as 0.2 m, and the corresponding cut-off frequency is 4,400 Hz. The sound velocity of the water is 1500 m/s and the density is 1000 kg/m$^3$. A velocity source of 1 m/s was added to one end of the virtual SWT as a sound source. A glass with a thickness of 0.013 m, Young’s modulus of $6 \times 10^{10} + 1.2 \times 10^7$ N/m$^2$, Poisson’s ratio of 0.23, and a density of 2500 kg/m$^2$ is added to the other end of the standing-wave tube. To obtain a more realistic impedance value, an air region, of which the sound speed is 340 m/s and the density is 1000 kg/m$^3$, and a sound absorption boundary is added at the outermost end of the virtual SWT. To obtain a plane wave in the tube, a rigid boundary is included at the boundary of the virtual SWT. To measure the material impedance using an SWT, the sound pressure in the tube must be measured. However, in the virtual SWT, the measuring point can be placed directly at the interface between the material and the medium to obtain the sound pressure and vibration velocity. A field point is added to the interface of water and glass to obtain the complex sound pressure and the velocity of the complex point at the interface.

After performing the calculation, we extract the complex pressure and the complex velocity at the field point and calculate the impedance of the material at different frequencies according to

$$Z_s = \frac{\bar{p}}{\bar{v}}$$  \hspace{1cm} (15)

where $Z_s$ is the impedance, $\bar{p}$ is the complex sound pressure, and $\bar{v}$ is the complex velocity.

The results are given in Table 1. Figure 6 shows the sound-field distribution at 1,300 Hz inside the SWT; a standing wave is formed in the tube, and the nodes and antinodes can be seen clearly.

The sound field in the SWT with the calculated equivalent impedance at the tube end is also calculated to see whether it is consistent with the sound field with glass at the
To verify the proposed method, we measured the pressure-release boundary is added at (1.3m, 0.15m, and 0.3m), and 100 field points are spaced equally in the $x$ direction obtained with the two models at 1,500 Hz. The results calculated with the two models are indeed consistent, thereby verifying the feasibility of the proposed method.

### 4. Sound Field due to Point Source in Glass Tank

#### 4.1. Finite-Element Method.

Using the obtained equivalent impedance, the sound field in the glass tank can be calculated. The current idea [5] of simulating the boundaries of glass or iron tanks in air is to simulate them as pressure-release boundaries. Because the characteristic impedances of glass and water are close and are far bigger than that of air, this approximation is reasonable. To compare the difference, both boundary conditions are used and the results are compared with the test results. The FE model with a length of 1.5 m, a width of 0.9 m, and a depth of 0.6 m is shown in Figure 8, and the coordinate system is defined according to Figure 1. A point source of 1 Pa is placed at the coordinates (1.3 m, 0.15 m, and 0.3 m), and 100 field points are spaced equally in the $x$ direction with $(y, z)$ coordinates of (0.45 m and 0.4 m). Since the impedance is the property of the material itself, the calculated impedance values are directly used for the calculation of the three-dimensional sound field.

When simulating the tank boundaries using the obtained equivalent impedance, the impedance is imposed at the surface of the tank except at $z = 0.6$ m. The surface $z = 0.6$ m is treated as a water surface given that the characteristic impedance of air is much smaller than that of water and instead a pressure-release boundary is added at $z = 0.6$ m.

#### 4.2. Test.

To verify the proposed method, we measured the sound field in a glass tank with the same dimensions as those of the FE model, the thickness of the glass is 0.013 m. The test system is shown in Figure 9. A spherical sound source driven by a power amplifier was placed at the coordinates (1.3 m, 0.15 m, and 0.3 m) and transmitted single-frequency signals. A hydrophone array (type B&K 8103) was placed in the glass tank and moved along the $x$ direction to measure the sound field. The hydrophone (type B&K 8103) used in the test is 50 mm in length and 9.5 mm in diameter. The size of the hydrophone and hydrophone holder is much smaller than the wavelength at 2000 Hz, and the sound scattering of the hydrophone can be ignored. The sound energy in enclosed space reaches dynamic equilibrium when the sound energy radiated by the sound source is consistent with the acoustic energy absorbed by the wall and the medium. The steady state sound field is obtained during the simulation calculation. In order to obtain the steady state sound field during the experiment, data acquisition is performed 10 seconds after the sound source starts transmitting.

#### 5. Results of Finite-Element Method and Tests

For the sound field in the closed space, as the frequency increases, the number of acoustic modes in the sound field increases and the distribution of the sound field becomes more and more complicated. At low frequencies, due to the interference, the sound field will have a distinct distribution. When the number of modes is sufficient, the sound field tends to diffuse. At low frequencies, in addition to considering the absorption of the boundary, it is also necessary to consider the phase of the reflected wave in order to accurately calculate the sound field. At high frequencies, since the sound field tends to diffuse, it is of little significance to consider the distribution of the sound field, mainly to calculate the acoustic energy density in the tank. In order to verify the accuracy of the proposed method, the calculation results and experimental results of the proposed method at low frequencies are mainly compared.

In Figure 10, we compare, for different values of the frequency, the FE results with those measured with the hydrophone located at $(y, z) = (0.45$ m, 0.4 m) and various values of $x$.

It can be seen from Figure 10 that the sound field in the glass tank is simulated accurately by using the obtained equivalent impedance. When using pressure-release boundaries to simulate the tank boundaries, the sound field in the glass tank is also calculated accurately at 1,300 Hz. However, as the frequency is increased, the sound field simulated with pressure-release boundaries differs from the actual sound field obtained by the test measurements, the sound field becoming increasingly sensitive to the phase and

![Figure 5: Schematic of a three-dimensional axisymmetric virtual SWT.](image)
Figure 6: Sound pressure distribution in virtual SWT at 1,300 Hz.

Figure 7: Sound pressure levels in axial direction with two different models at 1,500 Hz.

Figure 8: FE model.

Figure 9: Diagram of the measurement system.
Figure 10: Results of normalized sound pressure at (a) 1,300 Hz; (b) 1,400 Hz; (c) 1,500 Hz; (d) 1,600 Hz; (e) 1,700 Hz; (f) 1,800 Hz.
absorption at the boundaries. Therefore, with increasing frequency, the sound field obtained by simulating the tank boundaries as impedance boundaries becomes more accurate.

6. Conclusions

In this study, the FE method was used to calculate the sound field in a tank with impedance boundaries. The accuracy of doing so was verified by comparing the sound field in an SWT calculated by the FE method and analytical method. To obtain the equivalent normal impedance of the wall, a virtual SWT was established in FE software and the wall material was imposed at one end of the tube. From the sound pressure and vibration velocity on the interface between the medium and the wall, the equivalent normal impedance was obtained and used in the calculation of the sound field.

The sound field due to a point source in a glass tank was calculated by the proposed method. For comparison, the results of using pressure-release boundaries as the tank boundaries were also calculated. Experiments were carried out to verify the proposed method, and the results of simulating the tank boundaries as impedance boundaries agreed with the experimental results. The tank boundaries could be simulated as pressure-release boundaries only when the frequency was lower than the first mode frequency. The proposed method is suitable for calculating not only water tanks but also the sound field in rooms lined with acoustic materials.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

Acknowledgments

This research was funded by the National Natural Science Foundation of China (Grant no. 11874131).

References

[1] S. P. Robinson, P. M. Harris, J. Ablitt et al., “An international key comparison of free-field hydrophone calibrations in the frequency range 1 to 500 kHz,” The Journal of the Acoustical Society of America, vol. 120, no. 3, pp. 1366–1373, 2006.
[2] A. E. Isaev and A. N. Matveev, “Calibration of hydrophones in a field with continuous radiation in a reverberating pool,” Acoustical Physics, vol. 55, no. 6, pp. 762–770, 2009.
[3] R. A. Hazelwood and S. P. Robinson, “Underwater acoustic power measurements in reverberant fields,” in Proceeding of the IEEE Oceans, Aberdeen, UK, June 2007.
[4] Y. M. Zhang, R. Tang, Q. Li, and D. Shang, “The low-frequency sound power measuring technique for an underwater source in a non-anechoic tank,” Measurement Science and Technology, vol. 29, no. 3, Article ID 035101, 2018.
[5] S. Takahashi, T. Kikuchi, and A. Ogura, “Measurements of underwater sound absorption coefficient by the reverberation method,” Japanese Journal of Applied Physics, vol. 25, no. S1, p. 112, 1986.
[6] O. Robin, A. Berry, O. Doutrles, and N. Atalla, “Measurement of the absorption coefficient of sound absorbing materials under a synthesized diffuse acoustic field,” The Journal of the Acoustical Society of America, vol. 136, no. 1, pp. 13–19, 2014.
[7] C.-H. Jeong, “Absorption and impedance boundary conditions for phased geometrical-acoustics methods,” The Journal of the Acoustical Society of America, vol. 132, no. 4, pp. 2347–2358, 2012.
[8] P. M. Morse and R. H. Bolt, “Sound waves in rooms,” Reviews of Modern Physics, vol. 16, no. 2, pp. 70–150, 1944.
[9] D. Y. Maa, “Non-uniform acoustical boundaries in rectangular rooms,” The Journal of the Acoustical Society of America, vol. 12, no. 1, pp. 39–52, 1940.
[10] Y. Naka, A. A. Oberai, and B. G. Shinn-Cunningham, “Acoustic eigenvalues of rectangular rooms with arbitrary wall impedances using the interval Newton generalized bisection method,” The Journal of the Acoustical Society of America, vol. 118, no. 6, pp. 3662–3671, 2005.
[11] S. R. Bistafa and J. W. Morrisssey, “Numerical solutions of the acoustic eigenvalue equation in the rectangular room with arbitrary (uniform) wall impedances,” Journal of Sound and Vibration, vol. 263, no. 1, pp. 205–218, 2003.
[12] J. T. Du, W. L. Li, Z. G. Liu, H. A. Xu, and Z. L. Ji, “Acoustic analysis of a rectangular cavity with general impedance boundary conditions,” The Journal of the Acoustical Society of America, vol. 130, no. 2, pp. 807–817, 2011.
[13] X. Xie, H. Yang, and H. Zheng, “A weak formulation for interior acoustic analysis of enclosures with inclined walls and impedance boundary,” Wave Motion, vol. 65, pp. 175–186, 2016.
[14] W. K. Blake and L. J. Maga, “Chamber for reverberant acoustic power measurements in air and in water,” The Journal of the Acoustical Society of America, vol. 57, no. 2, pp. 380–384, 1975.
[15] S. A. Stotts and R. A. Koch, “A two-way coupled mode formalism that satisfies energy conservation for impedance boundaries in underwater acoustics,” The Journal of the Acoustical Society of America, vol. 138, no. 5, pp. 3383–3396, 2015.
[16] M. Vorlander, “Computer simulations in room acoustics: concepts and uncertainties,” The Journal of the Acoustical Society of America, vol. 133, no. 3, pp. 1203–1213, 2013.
[17] M. Aretz and M. Vorlander, “Efficient modelling of absorbing boundaries in room acoustic FE simulations,” Acta Acustica United with Acustica, vol. 96, no. 6, pp. 1042–1050, 2010.
[18] Y. Naka, A. A. Oberai, and B. G. Shinn-Cunningham, “Acoustic modeling in rectangular rooms with impedance using the finite element method with the Dirichlet-to-Neumann map,” The Journal of the Acoustical Society of America, vol. 117, no. 4, p. 2580, 2005.
[19] ISO 10534-1:1996: Acoustics – determination of sound absorption coefficient and impedance in impendance tubes – part 1: method using standing wave ratio.
[20] ISO 10534-2:1998: Acoustics – determination of sound absorption coefficient and impedance in impulse tubes – part 2: transfer-function method.