Iterative approach for transmission loss simulations

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Introduction

To obtain reliable results when performing transmission loss measurements in test facilities, great care must be taken to close all openings that connect source and receiver room and to minimize the flanking transmission. In the process of developing new or improved products, several tests are required and the costs increase.

To reduce costs, simulations can be introduced for design optimization by means of "virtual measurements", so that only the best prototypes or the ones with the most interesting properties are selected for the real measurements.

In the present work, a numerical method which emulates a measurement of the transmission loss is presented. For extending the frequency range of application, an iterative approach is developed which treats source room, partition and receiver room separately so that three smaller systems are solved instead of a larger one.

Definition of transmission loss

The transmission loss (R) of a partition is defined as:

$$R = 10\lg\left(\frac{W_{inc}}{W_{tr}}\right) \tag{1}$$

where W_{inc} is the incident power and W_{tr} is the transmitted power. W_{tr} is the power coming exclusively through the partition and not through other paths.

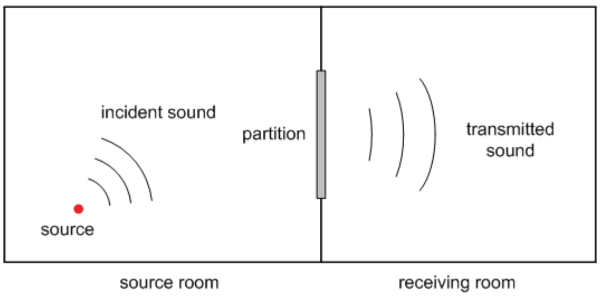

Figure 1: Test facility for transmission loss measurements.

Numerical model

A coupled BEM/FEM method is considered in order to keep the size of the system small. The BEM is used to compute the sound field in both rooms and the FEM is used to determine the displacement of the partition. For simplicity, a thin partition is assumed, so that it is modelled using shell elements. But thick partitions or multilayered specimens can be treated as well.

The discretized surfaces are grouped in 4 different types, denoted as $S_1 – S_4$. S_1 and S_4 are absorbing surfaces in source and receiving room, respectively. S_2 correspond to the partition and S_3 is a rigid surface separating the two rooms. Two acoustic domains are considered, Ω_I for the source room and Ω_{II} for the receiving room.

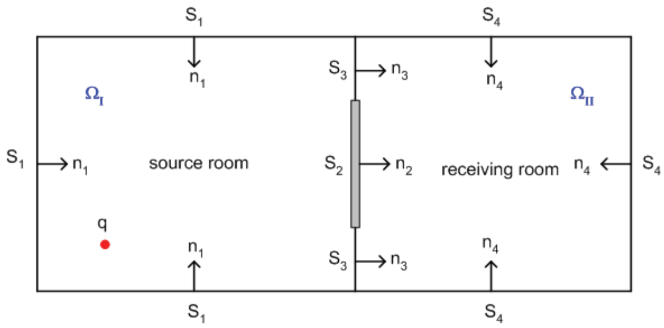

Figure 2: Numerical model for test facility and partition.

For the absorbing materials, the following relation applies

$$\frac{\partial p}{\partial n} = j\rho\omega\frac{p}{Z} \tag{2}$$

and for the thin partition the condition

$$\frac{\partial p_I}{\partial n} = \frac{\partial p_{II}}{\partial n} = \rho\omega^2 w \tag{3}$$

holds. w is the normal displacement of the partition. The quantities that need to be calculated are the pressure in source room, $p_I=[p_{I1}\ p_{I2}\ p_{I3}]^T$, the pressure in the receiving room, $p_{II}=[p_{II2}\ \ p_{II3}\ \ p_{II4}]^T$ and the displacement w.

Solution approaches

The sound transmission from source to receiving room through the partition is a result of an acoustic-structure interaction. On one hand, the pressure difference excites the partition. On the other hand, the vibration of the structure changes the pressure field in both rooms.

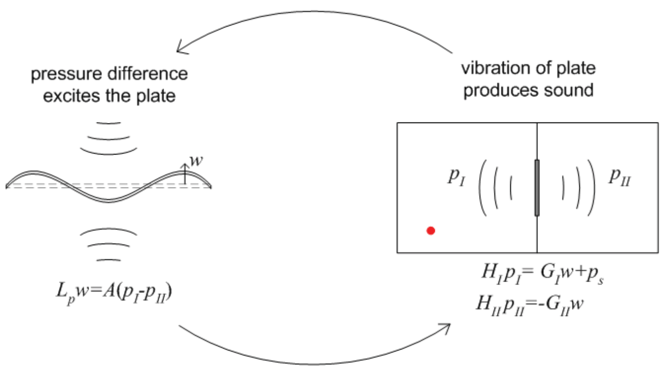

Figure 3: Acoustic-structure interaction.

In Figure 3, the basic equations governing the structure vibration and the sound radiation are written. L_p is the operator describing the motion of the plate in vacuum (FEM). A is the matrix relating the elements of S_2 with the elements of all surfaces in the source room. H_I, G_I, H_{II} and G_{II} are operators relating the acoustic pressure of the surfaces S_1-S_4 with the normal displacement on S_2 (BEM) and p_s is the pressure due to the source.

Direct approach

The usual way of solving the problem is using a direct approach. Both equations can be combined to obtain a single system of equations

$$\begin{bmatrix} L_p & -A & A \\ -G_I & H_I & 0 \\ G_{II} & 0 & H_{II} \end{bmatrix} \begin{bmatrix} w \\ p_I \\ p_{II} \end{bmatrix} = \begin{bmatrix} 0 \\ p_s \\ 0 \end{bmatrix} \tag{4}$$

If N_2 is the number of elements of the partition and N is the number of total elements of the model, the size of the matrix that is inverted is $(N+2N_2)^2$.

Iterative approach

An alternative way of solving the problem is to apply an iterative approach. Starting from certain initial values, the sound pressure and normal displacement of the partition are successively improved through separate calculations of structure vibration and sound radiation until a convergence is achieved. For the initial value, a good choice is the so called blocked-pressure approximation which considers a rigid partition $w^{(0)}=0$. The expressions for displacement and sound pressures for the i-th iteration are given by

$$H_I p_I^{(i)} = G_I w^{(i-1)} + p_s \tag{5.a}$$

$$H_{II} p_{II}^{(i)} = -G_{II} w^{(i-1)} \tag{5.b}$$

$$L_p w^{(i)} = A(p_I^{(i)} - p_{II}^{(i)}) \tag{5.c}$$

In this case, 3 smaller systems are solved instead of the larger system of the direct approach. Hence, the frequency range can be extended to higher frequencies.

Another advantage of this approach is that a better understanding of the acoustic-interaction can be obtained.

The drawback of the iterative approach is that the success of the method depends on the convergence. No analysis of the convergence behaviour of the iteration scheme in (5.a)-(5.c) has been performed, but the inclusion of damping in the partition and in its fixation (elastic boundary conditions) and the absorption material in both rooms should increase the convergence.

If eqs. (5.a)-(5.c) are combined, an iterative scheme for the displacement of the form

$$w^{(i)} = Tw^{(i-1)} + \phi \tag{6}$$

is obtained. For such scheme, the convergence is ensured if the spectral radius of T is smaller than 1 (ρ_T<1). The spectral radius is defined as $\rho_T = max(|\lambda_i|)$, where λ_i are the eigenvalues of T. For more details see [1] and [2].

Numerical test

The window test facility of the "Institut für Bauphysik – Fraunhofer Institut" in Stuttgart, Germany, was considered. The source and receiving rooms are rectangular rooms with dimensions 5.74m × 3.75m × 3.11m and 4.85m × 3.75m × 3.11m respectively. The opening is also rectangular with dimensions 1.25m × 1.5m. In the source room a small absorption α = 0.18 was considered while in the receiving room a high absorption α = 0.89 was used.

The simulated plate has the dimensions of the opening and a thickness of 0.004m. For aluminium, the values taken for Young's modulus, density and Poisson's ratio are: E = $64 \cdot 10^9$ Pa, ρ_p = 2,700 kg/m^3 and ν = 0.3. Damping was considered in the plate (η = 0.05). Elastic boundary conditions with a translational and a rotational springs with stiffness $K_b=10^7$ N/m^2, C_b=700N and damping η_b=0.1 were assumed.

In Figure 4, the difference in sound power level at 50 Hz at different points in both rooms is presented. Iterative and direct approaches provide practically the same results at all frequencies.

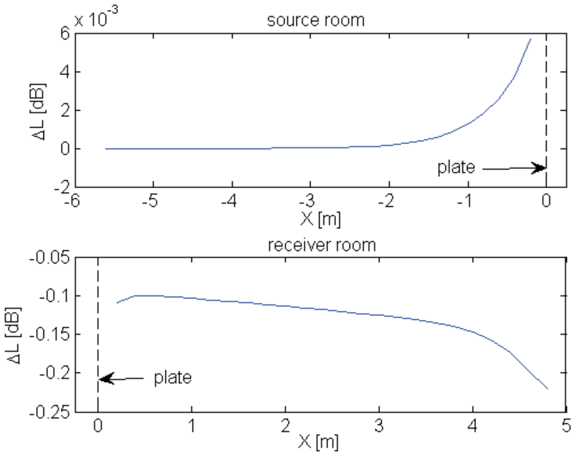

Figure 4: Difference in SPL between direct and iterative methods.

In Figure 5, the spectral radius and number of iterations are shown. Since ρ_T <1 for all frequencies, the convergence is ensured. For frequencies where ρ_T is smaller than 0.5, the number of iterations is also small.

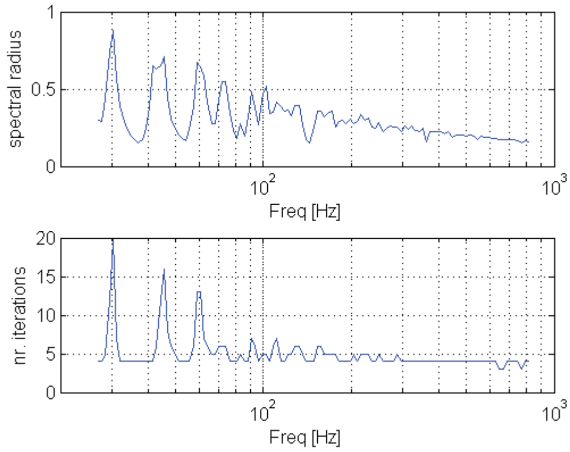

Figure 5: Parameters of the iteration

Calculation of *R*

In test facilities, usually sound pressure levels or intensity levels are measured. In order to compute the transmission loss, (1) has to be expressed in terms of those quantities. This can be achieved if some assumptions are made.

R from pressure levels

The norm DIN EN ISO 10140-2 defines the transmission loss with the expression

$$R_p = L_1 - L_2 + 10\lg\left(\frac{S}{A}\right) \tag{7}$$

where L_1 is the mean pressure level in source room in dB, L_2 the mean pressure level in receiving room in dB, S the area of the opening where the element is mounted (in m^2) and A the equivalent absorbing area in the receiving room (in m^2).

Equation (7) requires that the sound fields in both rooms are diffuse and that the sound in the receiving room is exclusively due to the sound coming through the test element.

R from intensity levels

The norm DIN EN ISO 15186-1 defines an expression that includes the case that the receiving room can be replaced by the open space. For this reason, the method considers the sound intensity measured on a surface involving completely the specimen at the receiving side S_m. The formula reads

$$R_I = L_1 - 6 - \left[L_I + 10\lg\left(\frac{S_m}{S}\right)\right] \tag{8}$$

where L_I is the normal intensity level averaged over S_m.

Equation (8) requires that the normal distance d, between partition and enveloping surface lies in the range $0.1<d<0.3$ and the difference between the intensity level and pressure level satisfies $0<L_p-L_I<10$ dB.

Mean sound pressure level

According to the norm DIN EN ISO 10140-4, the spatial averaged sound pressure level is defined as

$$L = 10\lg\left(\frac{1}{n}\sum_{i=1}^{n}\frac{p_i^2}{p_0^2}\right) \tag{9}$$

where n is the number of microphone positions in the room and p_i the rms value of the pressure. The norm recommends to place the microphones outside the direct field (e.g. a minimum of 1 m from the source) and at least 0.7 m away from the room borders and 1 m away from the specimen. The separation between microphone positions should be greater than 0.7 m. A minimum of 5 positions distributed in the room has to be considered, but they should not form a regular grid and no pair of microphones should lie in the same plane parallel to the room borders.

Diffuse sound field

A diffuse sound field is to be produced by loudspeakers in at least two positions or by a single loudspeaker moved to at least two positions. At low frequencies, especially below 100 Hz, the minimum number of loudspeakers increases to three. The sound field should be constant and have a uniform spectrum, i.e. the difference in the pressure level between adjacent 1/3-octave bands should not exceed 6 dB. The lack of diffusivity can be compensated by averaging the sound pressure obtained with different source positions. The norm recommends positions at least 0.7 m away from the room borders. The separation between source positions should be greater than 0.7 m, no pair of sources should lie in the same plane parallel to the room borders or be symmetric respect to the middle planes.

R at low frequencies

In rooms with small volumes and not favourable dimensions is not always possible to obtain reliable results at low frequencies using (7) or (8). Both require that at least one room dimension contains a wavelength and another room dimension at least a half wavelength of the lowest band middle frequency.

For frequencies between 50 and 160 Hz, the norm DIN EN ISO 15186-3 introduces the following definition for R

$$R_{low} = L_{pS} - 9 - \left[L_I + 10\lg\left(\frac{S_m}{S}\right)\right] \tag{10}$$

where L_{pS} is the mean pressure level in the source room averaged over the surface of the partition.

Comparison of R_p, R_I and R_{low}

To compare the results from (7), (8) and (10), 10 source positions, 30 microphone positions in the source room and 24 microphone positions in the receiving room meeting the norm recommendations were considered.

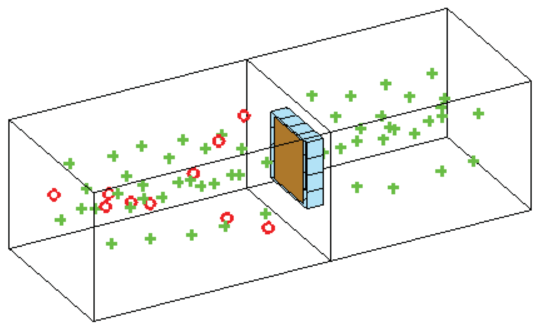

Figure 6: Source (red) and microphone (green) positions.

In Figure 6, all source and microphone positions are shown. The brown surface corresponds to the partition and the light blue surface is the enveloping surface used for the intensity measurements.

According to the norm 10140-4, one calculation is performed with each source position and the averaged transmission loss $\langle R\rangle$ is determined using the formula

$$\langle R\rangle = -10\lg\left(\frac{1}{Q}\sum_{i=1}^{Q}10^{-R_i/10}\right) \tag{11}$$

where Q is the number of source points.

The curves of transmission loss, R_i and the averaged value $\langle R \rangle$ were calculated for the 1/3-octave bands from 31.5 Hz up to 800 Hz. The 1/3-octave band sound levels are obtained from the narrow band values using the expression

$$L_{1/3} = 10\lg\left(\frac{1}{n_{1/3}}\sum_{i=1}^{n_{1/3}}\frac{p_i^2}{p_0^2}\right) \qquad (12)$$

where $n_{1/3}$ is the number of frequencies in the 1/3-octave band. In this calculation, $n_{1/3}$=8 for all bands.

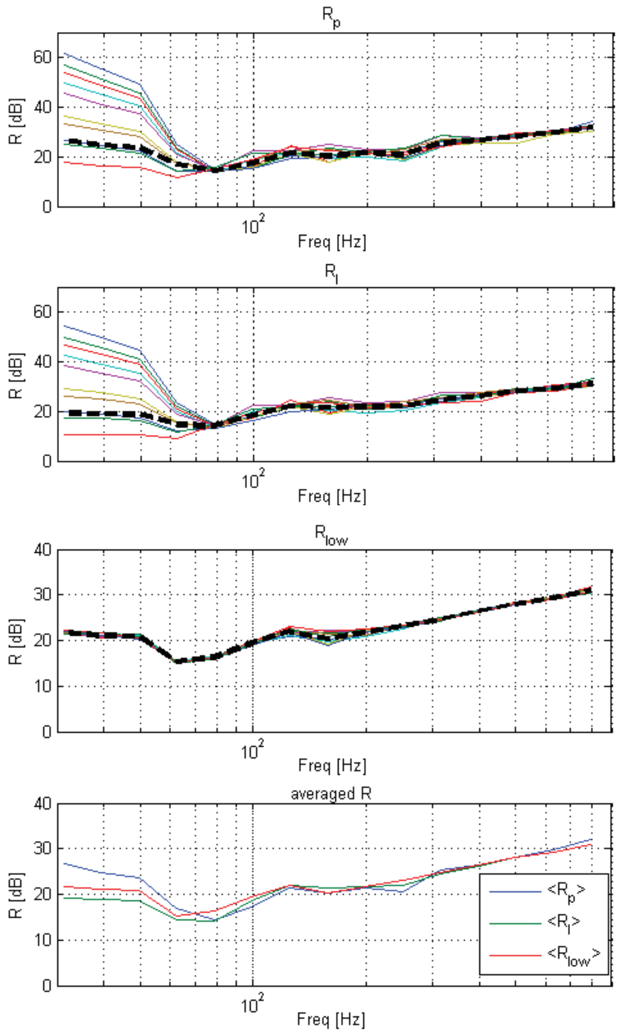

Figure 7: Transmission loss curves

The first 3 plots in Figure 7 show the curves of R for each source position (coloured solid lines) and the mean value (dotted black line). The last plot shows a comparison between the mean values. These results show the sensitivity of expressions (7) and (8) to the source position at low frequencies and confirm their validity above 100 Hz. The expression (10) is completely independent of the source position not only below 100 Hz but at all frequencies. The mean values of R obtained with the three expressions are very similar.

Looking at the sound pressure distribution in the source room (Figure 8), we find that at low frequencies, the microphones are placed always in the direct field of the source. Therefore, it is expected that the results differ significantly from one position to the other of the source.

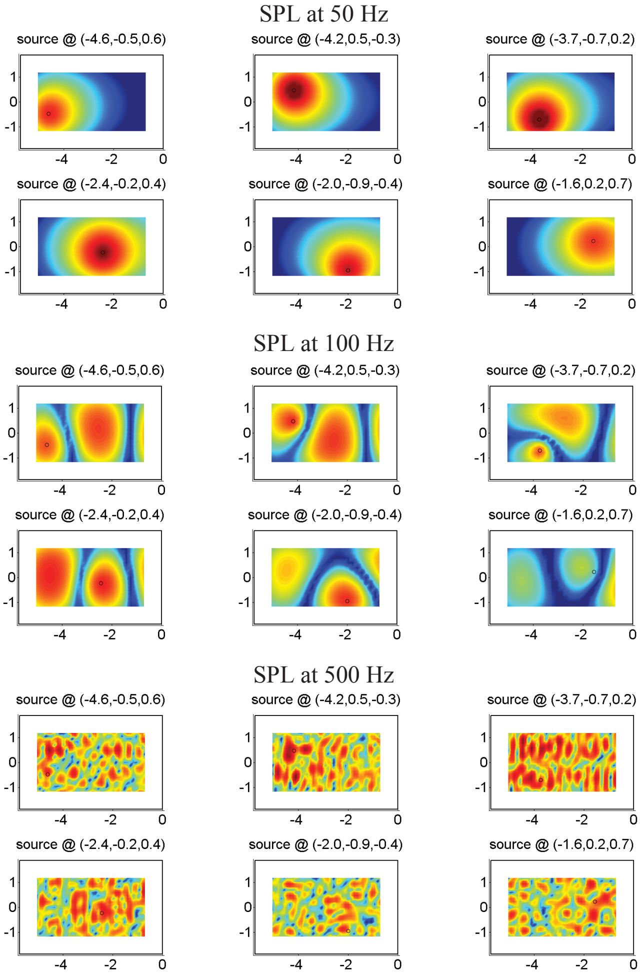

Figure 8: Pressure level distributions

Conclusions

This work presents an iterative approach to simulate transmission loss measurements. With this method, three separate smaller systems are solved instead of one large system. When the spectral radius of the system matrix is less than one, the method converges in only a few iterations. Three expressions for R given in the norms were investigated. Two of them are valid beyond 100 Hz and can also be used for lower frequencies if calculations are performed for several source positions. The third formula is valid for low frequencies and is insensitive to the position of the source. The results for the studied case show that this expression can be used also for higher frequencies.

Acknowledgements

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References

[1] R. Piscoya, M. Ochmann, R. Burgschweiger: Numerical simulation of the transmission loss of plates. Proceedings Acoustics 2012, Hong Kong.

[2] R. Piscoya, M. Ochmann: Numerical simulation of transmission loss test facilities. Proceedings Internoise 2012, New York.