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Estimation Methods for Sound Levels and Reverberation Time in a Room with Irregular Shape or Absorption Distribution

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Summary
The reverberation time of an enclosed space is an important parameter to describe the acoustic quality in enclosed spaces. Mainly due to its simplicity Sabine's equation is normally used even though the considered situations seldom comply with its preconditions: regular room shape, regular distribution of absorption and not to much absorption. For certain groups of applications the prediction errors can be countered by adding empirical corrections, by applying specific interpretations of the input data or by modifications to the Sabine equation. For other applications more sophisticated modelling could be appropriate. But it would be very practical for many applications to have available an engineering type of prediction scheme for the reverberation time or the sound level distribution in enclosed spaces using well-known laboratory data as input. Possibilities for such prediction schemes are developed and discussed in this paper, as back ground information to the informative annexes to the European standard EN 12354, part 6.

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1. Introduction

The acoustic quality of sound fields in an enclosed space is either expressed in the reverberation time, the amount of absorption or the resulting sound levels from sources in the room. To design for a good quality or to improve existing situations, it is useful to be able to estimate these quantities from the shape, lay-out and material properties of such enclosed spaces. With the help of diffuse field theory and Sabine this seems to be an easy task. However, that regular proves to be wrong, not because these theories are not correct but because real-life enclosed spaces do not comply with the theoretical assumptions.

Now, if the enclosed space considered is a concert hall, a theater or a factory several kinds of tools can be applied to solve the problem: mirror-image models, ray-tracing models and such. These are generally rather complicated and sophisticated models, research tools rather than engineering and design tools, needing experienced users. For some kind of spaces work has been done to facilitate the application in a more practical manner, like for factories. But if the enclosed space is much simpler like an office room, a corridor, a classroom, a gym or the heating room for a building, such models are not appropriate, even if they apply, but to the general user other possibilities than applying ïSabine' are hardly available. Hence there is a need for reliable and practical models on an engineering level to estimate the reverberation time and/or resulting sound levels in common rooms as encountered in office buildings, schools, hospitals or dwellings.

In this paper we will indicate some of the main problems and propose some possible solutions.

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2. Main problems

We restrict the subject to rooms and small halls. Several effects will have to be addressed, as:

- non-diffuse sound field due to uneven distribution of absorbing materials;
- non-diffuse sound field due to non-cubic room shapes;
- non-diffuse sound field due to irregular air spaces as a result of room shape or a large amount of objects;
- the effect of high absorption coefficients;
- the influence of scattering objects;
- the influence of scattering wall shapes.

All these aspects have an influence on the applicability of simple theory like diffuse field and Sabine, though the extensions of the effects vary and not all effects are independent. For instance, scattering by shaped walls and objects reduces the deviations by irregular room shapes or irregular absorption distribution.

On the one hand we have essentially one air volume, but with irregularities in dimensions or absorption distribution or both. This will still result in a rather even sound level distribution and a constant reverberation time, but not necessarily according to simple theories. On the other hand we encounter situations where the air spaces is actually subdivided in two or more sub-spaces, resulting in large variations in sound pressure levels, and probably also reverberation times, between these subspaces. In that case it is not relevant to consider quantities for the whole enclosed space, but we need estimations for these subspaces separately. It is clear that there will be situations in real life in between those two. But it can be hoped that if we are able to tackle those two main situations, we can also estimate with more reliability the in-between situations, at least by approaching them from the two sides.

Based on literature some possibilities will be indicated to incorporate the mentioned aspects in the predictions. However, no additional comparison is made with measurement results and hence no indication of accuracy can be given. The given approaches are only illustrated by some examples to indicate globally the effects and are compared with results of other prediction methods.

3. Irregular room shapes

3.1. A possible model

In rooms with a shape that deviates much from a rectangular shape or in rooms filled to a large amount with machinery and equipment, it is unlikely that the sound field is diffuse and even unlikely that the reverberation time will be the same throughout the space. In such cases the enclosed air space will have an irregular shape, which we could model by considering more or less cubic sub-spaces. Such connected subspaces will form the whole space. That modelling opens the possibility to apply simple diffuse field relations for each sub-space and derive the total sound field by considering the sound power balance between all sub-spaces [1, 2]. In that case the steady state sound field follows quite easily from the geometry, the known sound sources and the distribution of absorbing material and objects. This is normally what is needed; the results could of course also be expressed as a reverberation time, depending on the position. This approach resembles a SEA model, but, since the sub-spaces will clearly by strongly coupled through the ïinvisible walls' between them, it also violates one of the restrictions of such a model. It can therefore only be hoped that this approach will give a reasonable estimation of the main effects. So it is preferred to address it just as considering the power balance between subsystems.

According to this model the sound pressure level $L_{p,s}$ in each sub-space is estimated by the direct sound at distance r_k from the sources with sound power W_k in that sub-space and the indirect sound from the reverberant field resulting from the distribution of the sound energy density e_s over all sub-spaces. The direct sound might be attenuated by various factors like screening, denoted with the attenuation factor χ_k.

$$L_{p,s} = 10\lg\left[\frac{\rho_0 c_0}{p_0^2}\left(4c_0 e_s + \sum \frac{W_k}{4\pi r_k^2}\chi_k\right)\right]. \quad (1)$$

The density and sound speed in air are denoted by ρ_0 and c_0, while p_0 is the reference pressure of 20 μPa.

The reverberant sound energy density follows from the sound power injected into each sub-space, the equivalent sound absorption area for each sub-space and the power balance between all connected sub-spaces. The equivalent sound absorption area A_s in each sub-space s with volume V_s is calculated according to equation (2), from the absorption by objects A_{obj}, the absorption by the boundaries with area S_b and absorption coefficient α_b and the absorption by air, characterized by the attenuation coefficient m in Np/m. For all parts of the boundary that are actually an open surface S_{sj} between the considered subspace s and the connected sub-space j the assumption is $\alpha_b = 1$.

$$A_s = \sum A_{\text{obj}} + \sum \alpha_b S_b + 4mV_s. \quad (2)$$

The following relation then holds for each sub-space s in which a reverberant sound power W_s is injected by sources,

$$W_s = c_0 e_s A_s - \sum_{j\neq s}^{n} c_0 e_j S_{sj}. \quad (3)$$

The reverberant sound power follows from the sound power W_k of each sound source in that sub-space and the average absorption coefficient $\overline{\alpha_s}$ for that sub-space, taking into account all absorption that occurs (i.e. surfaces, openings, objects, air).

$$W_s = \left(1 - \overline{\alpha_s}\right)\sum W_k. \quad (4)$$

Equation (3) can be rewritten in the following matrix representation for all considered sub-spaces.

$$\{M\}\{e\} = \{W\}/c_0, \quad (5)$$

where $\{e\}$ and $\{W\}$ are the vectors representing the energy densities and the input powers for each sub-space and matrix $\{M\}$ contains the absorption terms. The intensity for each sub-space and thus the sound pressure level for the reverberant field follow by matrix inversion.

$$\{e\} = \{W\}/c_0\{M\}^{-1}. \quad (6)$$

The sound pressure levels then follow from equation (1).

3.2. Examples

3.2.1. Example 1

Take an L-shaped room (9×4×2,5 m^3 and 3×4×2,5 m^3) as in Figure 1 with a sound source in the left corner (O) with L_W = 100 dB re 1 pW. The absorption coefficient for all boundaries is 5% in case A and the absorption coefficient is 50% for two of the walls and 20% for the ceiling in case B. The room is divided in three sub-spaces. The resulting sound pressure levels for the considered frequency band, neglecting air absorption, are indicated in the figure.

Only in the sub-space with the source the direct sound will have a marginal influence, in the other sub-spaces

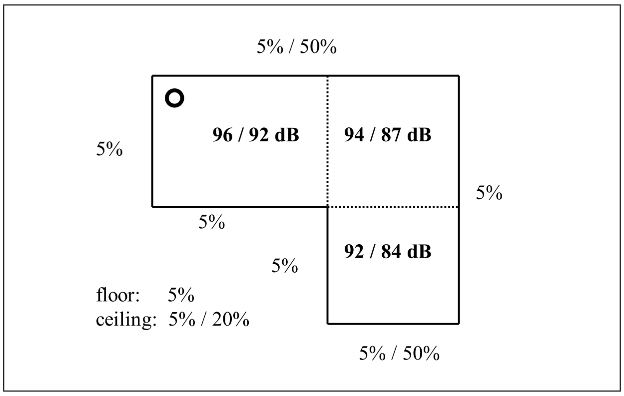

Figure 1. L-shaped room with source (O) and two different absorption situations A and B; the resulting sound pressure levels are indicated.

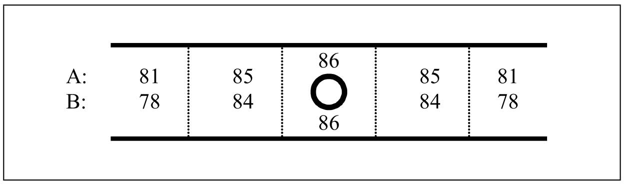

Figure 2. Corridor with central source (O) and two different absorption situations A and B; the resulting sound pressure levels are indicated.

even with direct sight to the source that influence is negligible.

Considering the whole room as a 'Sabine' room would result in a sound pressure level of 94 dB en 88 dB respectively. The detailed modelling gives a level distribution with deviates clearly from these values, in the more absorbing situation between +4 dB and −4 dB. This corresponds more with experience, say in a living room than the even distribution of a diffuse field.

3.2.2. Example 2

Consider a corridor with a length of 50 m and a cross-section of 10×5 m^2. Both ends are considered to be openings (100% absorption). Two situations are calculated: (A) all surfaces have an absorption coefficient of 5% and (B) walls and ceiling near the opening are treated with absorbing material over a length of 10 m (absorption coefficient 50%). The corridor is divided in five sub-spaces of equal length (10 m). The resulting sound pressure levels with a sound source of $L_W = 100$ dB re 1 pW in the middle are indicated in the figure.

Considering the whole corridor as a 'Sabine' space would result in a sound pressure level of 83 and 79 dB respectively. Especially the last situation is well off in the interior part of the corridor.

4. Irregular absorption distribution

4.1. Various approaches

It has long time been recognised that in rooms with clearly other shapes than a cube, the sound field will have a preference for certain directions and thus absorbing treatment will be more effective on some surfaces than others. Several attempts have been made to formulate this in adjusted reverberation time equation, either based on Sabine or Eyring: Millington [3], Fitzroy [4], Arau [5] and others. But none of these approaches seems to be general enough to work in all relevant situations (see for instance the comparison in paragraph 4.5). A more physical and fundamental approach has been used by Nilsson [6, 7]. But for an engineering prediction tool, that approach has some drawbacks. Hence an attempt is made to start from that work in developing an engineering solution to the problem.

4.2. Nilsson's model

A theoretical study that deals with the irregular distribution of absorption in a rectangular room has more recently been done by Nilsson [6], who considered mainly office rooms with essentially a much smaller height than width and length and mainly absorbing material on the ceiling. The approach is to consider separately grazing and non-grazing room modes with respect to the absorbing surfaces i.e. the ceiling, characterised by its acoustic impedance. To predict the overall reverberation process these two types of modes are combined in a kind of SEA model for two uncoupled systems; if scattering objects are present, these objects introduce a coupling between the two systems. Modal theory is applied to describe the attenuation for the grazing modes and the division of the total field in grazing and non-grazing modes, using the acoustic impedance of the surfaces as parameter. The SEA-model is than solved in the time domain to get the sound pressure decay curves. This approach clearly gives calculated results that correspond quite well with measurements; giving longer reverberation times than according to Sabine's relation [6]. This model gives much insight, but for a more practical approach it is not fit yet and needs adjustments. This we will consider in the next section.

4.3. Possible general approach

In order to apply this approach more general there are some points that need to be considered:

- other absorbing surfaces than ceilings should be considered;
- the attenuation of grazing modes should be estimated from readily available absorption data, in stead of impedance;
- some simplifications are necessary to become practical;
- for the larger rooms air attenuation should also be considered;
- for occupied rooms the volume fraction of objects should also be considered;
- at low frequencies the room dimensions will become too small to use a model based on numerous room modes.

We first consider the higher frequencies with sufficient room modes and look at a rectangular room.

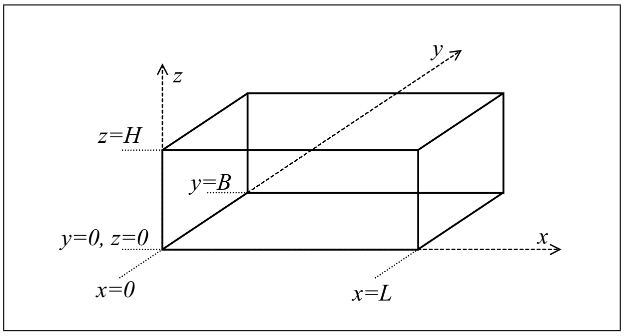

Figure 3. Definition of dimensions for a rectangular space.

The room dimensions are defined as in Figure 3 for the room with volume $V = L \cdot B \cdot H$. For the higher frequencies the total sound field is divided into three oblique fields, grazing to the surfaces perpendicular to the axis x, y and z, and a diffuse field. The effective absorption and corresponding reverberation time is determined for each of these fields. The importance of each of these sound fields is determined by the number of modes in those fields as deduced from the room dimensions. For the lower frequencies the total sound field is considered with a reduced absorption effect due to the lack of diffusion in the room at those frequencies.

The number of modes primarily grazing to the room boundaries is given by [6, 7]

$$\begin{aligned} n_{gx} &= 2\frac{4\pi V}{c_0^3} f^2\bigl(1-\cos\varphi\bigr) + \frac{B+H}{2c_0} \\ &\quad + 2\varphi\frac{f}{c_0^2}\bigl(LB+LH+2BH\bigr), \\ n_{gy} &= 2\frac{4\pi V}{c_0^3} f^2\bigl(1-\cos\varphi\bigr) + \frac{L+H}{2c_0} \\ &\quad + 2\varphi\frac{f}{c_0^2}\bigl(BL+BH+2LH\bigr), \qquad (7a) \\ n_{gz} &= 2\frac{4\pi V}{c_0^3} f^2\bigl(1-\cos\varphi\bigr) + \frac{L+B}{2c_0} \\ &\quad + 2\varphi\frac{f}{c_0^2}\bigl(HL+HB+2LB\bigr). \end{aligned}$$

The rest of the modes are considered to belong to the diffuse field (non-grazing).

$$\begin{aligned} n_{ng} &= \frac{4\pi V}{c_0^3} f^2\bigl(3\cos\varphi - 2\bigr) \qquad (7b) \\ &\quad + \bigl(\pi - 4\varphi\bigr)\frac{f}{c_0^2}\bigl(LB+LH+BH\bigr). \end{aligned}$$

Here n_{ng} refers to the non-grazing modes and n_{gx} to the modes grazing to the planes $x = 0$ and $x = L$, n_{gy} to the modes grazing to the planes $y = 0$ and $y = B$ and n_{gz} to the modes grazing to the planes $z = 0$ and $z = H$. From this the ratio $N_x = n_{gx}/n_{ng}$ etc. between the modal density of grazing and non-grazing modes can be calculated. This ratio determines the power distribution by a sound source over the groups of modes.

The relevant angle to distinguish between grazing and non-grazing depends on the impedance of the surface in a rather complicated way [6]. It is necessary to simplify this and a reasonable choice for a fixed the grazing angle is $\pi/30$ (about 6°). With this assumption the terms with cosines become 0,0055 and 0,9835. Thus the relative mode numbers can be expressed as

$$\begin{aligned} N_x &= 0.14 + 1.43\Bigl(\frac{B+H}{2c_0} + \frac{\pi f}{c_0^2} BH\Bigr)\frac{c_0^3}{4\pi f^2 V}, \\ N_y &= 0.14 + 1.43\Bigl(\frac{L+H}{2c_0} + \frac{\pi f}{c_0^2} LH\Bigr)\frac{c_0^3}{4\pi f^2 V}, \qquad (8) \\ N_z &= 0.14 + 1.43\Bigl(\frac{L+B}{2c_0} + \frac{\pi f}{c_0^2} LB\Bigr)\frac{c_0^3}{4\pi f^2 V}. \end{aligned}$$

Another simplification needed is to characterise the acoustic properties of the surfaces by the absorption coefficient instead of the wall impedance. The absorption term for grazing incident with the planes at $x = 0$ and $x = L$ is normally dominated by its expression for the higher frequencies that can be simplified as follows:

$$A_{gx} \approx \frac{c_0^2}{f^2 L^2}\bigl[\Re e\{Y'_{x=0}\} + \Re e\{Y'_{x=L}\}\bigr], \qquad (9)$$

where $\Re e\{Y'\}$ is the real part of the normalized admittance of the indicated side. This value can be a factor 2 large or smaller depending on one or two opposite sides being absorbing. Equivalent relations exist for the grazing fields perpendicular to the y- and z-axes.

On the other hand for the other sides with non-grazing incidence the sound field will tend to a two-dimensional sound field with more emphasize on normal incidence rather than diffuse incidence. This can also be characterized by the admittance as

$$A_{ng,x} \approx \frac{\pi}{4} 8BH\Re e\{Y'\}. \qquad (10)$$

To be practical it is necessary to replace the admittance by the absorption coefficient as it is readily known from standardized measurements [8]. A rough estimate of the real part of the wall impedance admittance follow from the absorption coefficient by

$$\Re e\{Y'\}\bigr] \approx 0.45\alpha_s\sqrt[3]{f/f_{\mathrm{ref}}} \text{ with } f_{\mathrm{ref}} = 1000\,\mathrm{Hz}. \qquad (11)$$

This estimate is based on some data given in [6] and on calculated values [9, annex B] for four absorbing wall linings, with varying thickness and flow resistance. The results are illustrated in Figure 4.

The total equivalent sound absorption areas, A_x, A_y, A_z and A_d for the grazing sound fields x, y and z and the

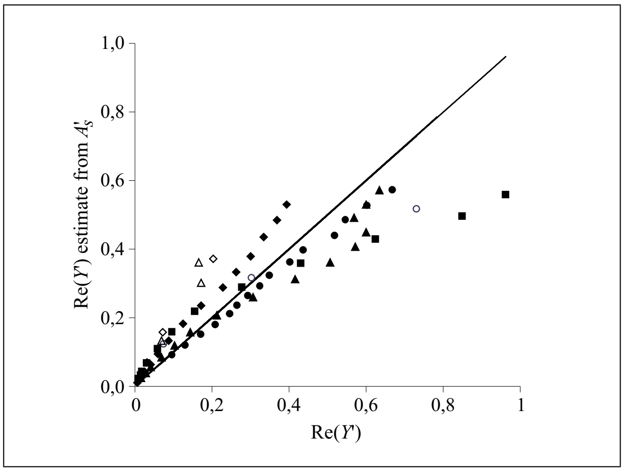

Figure 4. Comparison between direct determined and estimated value of the real part of the admittance for some absorbing wall linings in the frequency range 100 Hz to 2500 Hz.

diffuse field due to the room surfaces and air absorption may than be determined from

$$\begin{aligned}
A_x &= 0.45\frac{c_0^2}{f^2L^2}\left[A_{x=0}+A_{x=L}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}} \\
&\quad + 2.83\left[A_{y=0}+A_{y=B}+A_{z=0}+A_{z=H}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}}+\pi mV, \\
A_y &= 0.45\frac{c_0^2}{f^2B^2}\left[A_{y=0}+A_{y=B}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}} \\
&\quad + 2.83\left[A_{x=0}+A_{x=L}+A_{z=0}+A_{z=H}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}}+\pi mV, \\
A_z &= 0.45\frac{c_0^2}{f^2H^2}\left[A_{z=0}+A_{z=H}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}} \qquad (12) \\
&\quad + 2.83\left[A_{x=0}+A_{x=L}+A_{y=0}+A_{y=B}\right]\sqrt[3]{\frac{f}{f_{\text{ref}}}}+\pi mV, \\
A_d &= \big(A_{x=0}+A_{x=L}+A_{y=0} \\
&\quad + A_{y=B}+A_{z=0}+A_{z=H}\big)+4mV,
\end{aligned}$$

where the sound absorption area A of the surfaces perpendicular to the x, y and z-axes as denoted by the index $x=0$, $x=L$ etc. are based on the absorption coefficients [8] and m is the power attenuation coefficient in air [9]. After a preliminary comparison with some measurement results during the development of this approach, the two constants 0.45 and 2.83 were replaced by ½ and $\sqrt{2}$ in the published version of EN 12354 [9]. The more extensive comparison presented in this paper confirmed this adjustment.

The different sound fields are coupled by the scattering effects of surfaces and objects [7]. This can be expressed in a scattering absorption area A'. For objects this is directly represented by the absorption area, but for surfaces the scattering effect is given by a new quantity, the scattering coefficient δ [10]. This results in the scattering sound absorption area A_x', A_y', A_z' and A_d' for each sound field as it may be determined from

$$\begin{aligned}
A_x' &= \left[LH(\delta_{y=0}+\delta_{y=B})+LB(\delta_{z=0}+\delta_{z=H})\right] \\
&\quad + A_{obj,y}+A_{obj,z}+A_{obj,central}, \\
A_y' &= \left[BH(\delta_{x=0}+\delta_{x=B})+LB(\delta_{z=0}+\delta_{z=H})\right] \\
&\quad + A_{obj,x}+A_{obj,z}+A_{obj,central}, \qquad (13) \\
A_z' &= \left[BH(\delta_{x=0}+\delta_{x=B})+LH(\delta_{y=0}+\delta_{y=H})\right] \\
&\quad + A_{obj,x}+A_{obj,y}+A_{obj,central}, \\
A_d' &= \sum_{\text{all}} A_{obj}+N_xA_x'+N_yA_y'+N_zA_z',
\end{aligned}$$

using the scattering coefficient of the surface indicated with the index $x=0$, $x=L$ etc. The scattering effect of the absorbing objects is either associated to a surface (x, y, z) or to the central area of the enclosed spaces.

The equivalent sound absorption area A of a surface is deduced from the sub-area and corresponding absorption coefficients for that surface. Both absorption coefficient as absorption of objects can be determine by standardized measurement methods [8]. Additionally a scattering coefficient δ can be attributed to the room surfaces that indicate the fraction of the reflected energy that is reflected diffusively, its value ranging from 0.0 to 1.0. Even if data on this coefficient is yet hardly available, a global estimation can be used to get an impression of the effect of scattering on the results in specific situations. A measurement method is being standardized [11]. The scattering coefficient takes into account irregularities in the plane surfaces. For hard plane surfaces a typical value will be 0.05 or less, but for walls with niches like a facade the value at mid and higher frequencies can take typical values of 0.4 to 0.6.

Now by applying the power balance between the four sound fields as in SEA, the effective sound absorption area A^* for each sound field is deduced as

$$\begin{aligned}
A_d^* &= \left[A_d+A_d'-\frac{N_xA_x'^2}{A_x+A_x'}-\frac{N_yA_y'^2}{A_y+A_y'}-\frac{N_zA_z'^2}{A_z+A_z'}\right] \\
&\quad \cdot\left[1+\frac{N_xA_x'}{A_x+A_x'}+\frac{N_yA_y'}{A_y+A_y'}+\frac{N_zA_z'}{A_z+A_z'}\right]^{-1}, \qquad (14a)
\end{aligned}$$

$$\begin{aligned}
A_x^* &= \frac{A_x+A_x'}{1+A_x'/A_d^*}, \quad A_y^* = \frac{A_y+A_y'}{1+A_y'/A_d^*}, \\
A_z^* &= \frac{A_z+A_z'}{1+A_z'/A_d^*}, \qquad (14b)
\end{aligned}$$

and the steady state reverberation time for each sound field, diffuse field and fields grazing to surfaces at x, y and z, is given by

$$\begin{aligned}
T_x &= \frac{0.16V}{A_x^*}, \qquad T_y = \frac{0.16V}{A_y^*}, \\
T_z &= \frac{0.16V}{A_z^*}, \qquad T_d = \frac{0.16V}{A_d^*}. \qquad (15)
\end{aligned}$$

In principal, here and elsewhere, the volume should refer to the free space excluding the space occupied by objects. Though the difference between that volume and the room volume is normally small, it can have some influence

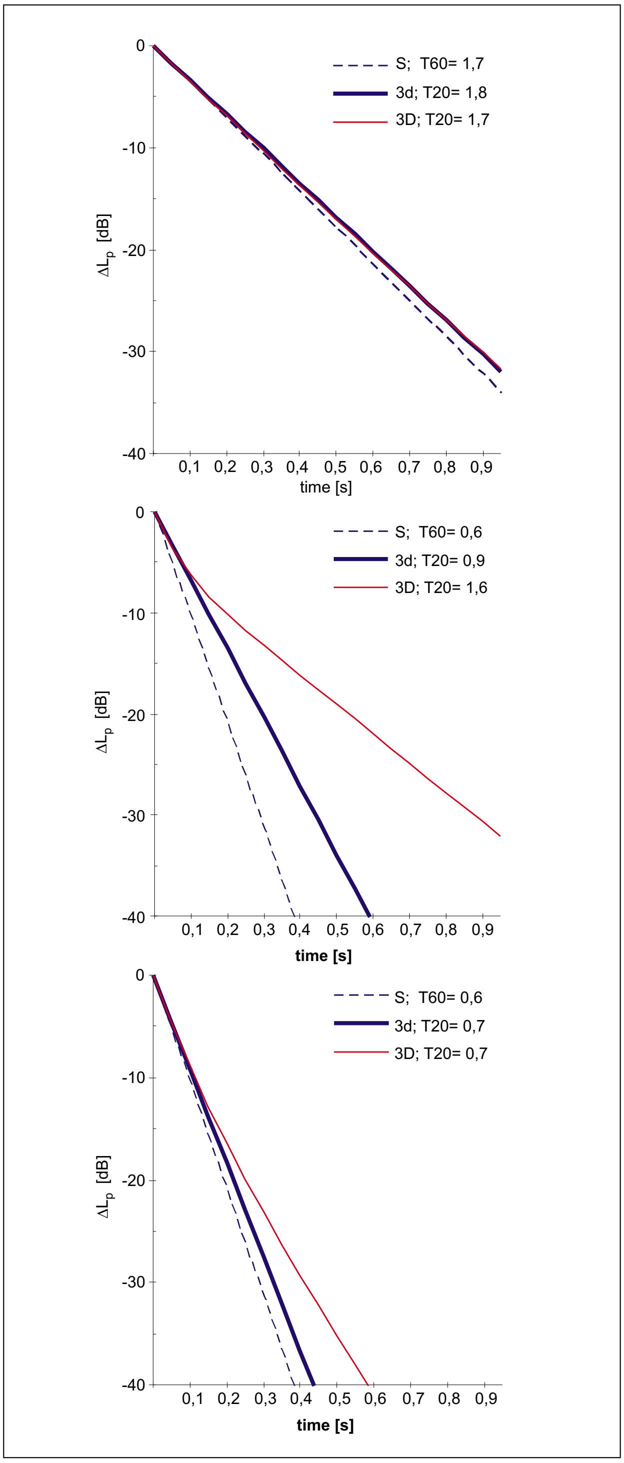

Figure 5. Illustration of estimated decay curves at 1000 Hz and resulting reverberation time in a room of 5,60×3,00×2,45 m^3; S = Sabine; 3d = EN 12354/annex D; 3D = from curve based on EN 12354/D. Top: bare; all surfaces $\alpha = 5\%$, middle: As top with absorbing ceiling ($\alpha = 50\%$), bottom: as middle with scattering head wall ($\delta = 50\%$).

so should be taken into account. For convenience we will simply use the symbol V for the appropriate volume.

If the differences between the four reverberation times from equation (15) are small, the diffuse field reverberation time can be considered as an adequate estimation for the considered situation. If not, the reverberation time is probably longer and a more realistic estimate can be deduced as the average of the reverberation times from equation (15) for the higher frequency range. This estimation of the reverberation time cannot be shorter than that for the diffuse field though. This is the estimate given in annex D of EN 12354, in the examples in Figure 5 indicated with 3d.

$$T_{\text{estimate}} = \frac{T_x + T_y + T_z + T_d}{4} \geq T_d. \quad (16)$$

The relative sound level at $t = 0$ s for each sound field x, y, z and d may be determined from

$$\begin{aligned} L_{p,d} &= -10\lg\left(1 + N_x\frac{A_d^*}{A_x^*} + N_y\frac{A_d^*}{A_y^*} + N_z\frac{A_d^*}{A_z^*}\right), \\ L_{p,x} &= L_{p,d} + 10\lg\left(N_x\frac{A_d^*}{A_x^*}\right), \\ L_{p,y} &= L_{p,d} + 10\lg\left(N_y\frac{A_d^*}{A_y^*}\right), \\ L_{p,z} &= L_{p,d} + 10\lg\left(N_z\frac{A_d^*}{A_z^*}\right). \end{aligned} \quad (17)$$

Adding the sound levels gives the total decay curve which will normally be curved in cases with irregular absorption distribution; see the examples in Figure 5. An estimation of the reverberation time should be deduced from the best fitting straight line to this curve, for instance between −5 dB and −25 dB (estimates indicated with 3D in Figure 5).

This approach to consider different sound fields is relevant for the higher frequencies, but no longer at low frequencies. The frequency range with insufficient modes is in [8] characterised by the following transfer frequency, and that is adopted in EN 12354 too:

$$f_t = \frac{8.7c_0}{\sqrt[3]{V}}. \quad (18)$$

The effective sound absorption area for the total field for low frequencies $A*_{xyzd}$ may be determined from equation (19a) by reducing the effectiveness of the absorption of the surfaces denoted as $\overline{A}$ in equation (19b),

$$\begin{aligned} A^*_{xyzd} = \Big(&\overline{A_{x=0}} + \overline{A_{x=L}} + \overline{A_{y=0}} \\ &+ \overline{A_{y=B}} + \overline{A_{z=0}} + \overline{A_{z=H}}\Big) + \sum A_{obj} + 4mV, \end{aligned} \quad (19a)$$

with for each index x,y and z

$$\overline{A} = A\mathrm{e}^{-A/s}, \quad (19b)$$

where A and S are the equivalent sound absorption area and surface area of the considered surface. This proposed reduction of the effectiveness of the absorption area in equation (19b) is purely a pragmatic approach to take into account the reduced absorption for low frequencies with mainly normal incidence.

For the lower frequencies the estimation of the reverberation time is then given by:

$$T_{\text{estimate}} = \frac{0.16V}{A^*_{xyzd}}. \quad (20)$$

Table I. Dimensions, absorption coefficients (α) and scattering coefficient (δ) for four room types, each under two conditions.

enclosed space	α	125	250	500	1000	2000	4000
large, cubic room $7\times7\times7\,m^3$	A1. all surfaces	0.02	0.02	0.03	0.04	0.05	0.06
	A2. + abs. ceiling	0.45	0.70	0.80	0.90	0.90	0.80
normal room $6\times4\times3\,m^3$	B1. floor	0.02	0.03	0.03	0.04	0.06	0.05
	walls	0.02	0.02	0.03	0.04	0.05	0.05
	abs.ceiling	0.45	0.70	0.80	0.90	0.90	0.80
	B2. floor+carpet	0.02	0.04	0.06	0.20	0.30	0.35
	hard ceiling	0.02	0.02	0.03	0.04	0.05	0.05
	B3. floor+carpet + abs. ceiling						
flat room $16\times8\times3\,m^3$	C1. floor	0.02	0.02	0.03	0.04	0.05	0.05
	sidewalls	0.25	0.15	0.10	0.05	0.03	0.03
	walls	0.03	0.03	0.02	0.04	0.05	0.08
	ceiling	0.02	0.02	0.03	0.04	0.05	0.05
	C2. floor+carpet	0.02	0.04	0.06	0.20	0.30	0.35
	C3. +abs.ceiling	0.45	0.70	0.80	0.90	0.90	0.80
long room $32\times4\times3\,m^3$	D1. floor	0.01	0.01	0.02	0.02	0.03	0.03
	walls	0.02	0.02	0.03	0.04	0.05	0.05
	ceiling	0.02	0.02	0.03	0.04	0.05	0.05
	D2. floor+carpet	0.02	0.04	0.06	0.20	0.30	0.35
	abs.ceiling	0.45	0.70	0.80	0.90	0.90	0.80
	D3. only abs.ceiling	0.45	0.70	0.80	0.90	0.90	0.80
all rooms	δ	125	250	500	1000	2000	4000
	smooth	0.05	0.05	0.05	0.05	0.05	0.05
	rough (façade)	0.05	0.05	0.20	0.40	0.60	0.60

4.4. Examples

Take a room of $L \times B \times H = 5.60 \times 3.00 \times 2.45\,m^3$ and consider the 1000 Hz octave band. The room is (a) bare with an absorption coefficient of 5% for all walls, (b) with an absorbing ceiling ($\alpha = 50\%$) and (c) with the absorbing ceiling and scattering surfaces on the head walls ($\delta = 50\%$). In every situation three curves are shown with the average decay time (T_{20} in s): according to Sabine (S), according to the presented approach of EN 12354 annex D for the estimated reverberation time (straight decay curve assumed) and according to the estimated decay curve from EN 12354 annex D. The effects discussed are clearly illustrated and the effects are globally in accordance with the measured effects from Nilsson [6].

4.5. Validation by ray-tracing models

One of the main difficulties in comparing any prediction model with measurement results is the availability of appropriate input data. In the considered cases that would mean the actual absorption coefficients of all surfaces and objects. Most of the time those data are not available and one starts with guesses or fits the preliminary calculations to the measurement data. A possibility to avoid such problems is to compare the results of a prediction model with the results of a known accurate or better model using exactly the same input data. This is the approach taken here.

Four typical room shapes have been considered and in each the amount of absorbing material on the surfaces has been varied. The reverberation time is calculated by a ray-tracing model (CAESAR [12, 13]) and by the proposed estimation method as it is incorporated in annex D of EN 12354-6 [9]. Also predictions by some other models have been applied (Sabine, Fitzroy, Arau).

It could be argued that rather Eyring should be used than Sabine [14]. The general effect of using Eyring for these types of rooms is that the reverberation time is equal or shorter than that according to Sabine. However, the experience is that, if anything, the reverberation times are normally equal or longer than it follows from Sabine. So using the theoretically more correct model of Eyring, with input data according to Sabine in the reverberation room, does not seem to touch the real field situation sufficiently.

The considered rooms are a large cubic room ($343\,m^3$), a flat room ($384\,m^3$), a long room ($384\,m^3$) and a normal room ($72\,m^3$); see Table I. Absorption coefficients ranged from 2 to 5% for bare surfaces to 90% at high frequencies for absorbing surfaces. The calculations were performed with the assumption of low scattering for walls and floors (5%) and also with the assumption of higher scattering for all surfaces, ranging from 5% at low frequencies to 60% at high frequencies also used in [9]. It may not be realistic to apply the same scattering coefficients to all surfaces in the same way, but here we are more concerned in effects than in estimating real situations.

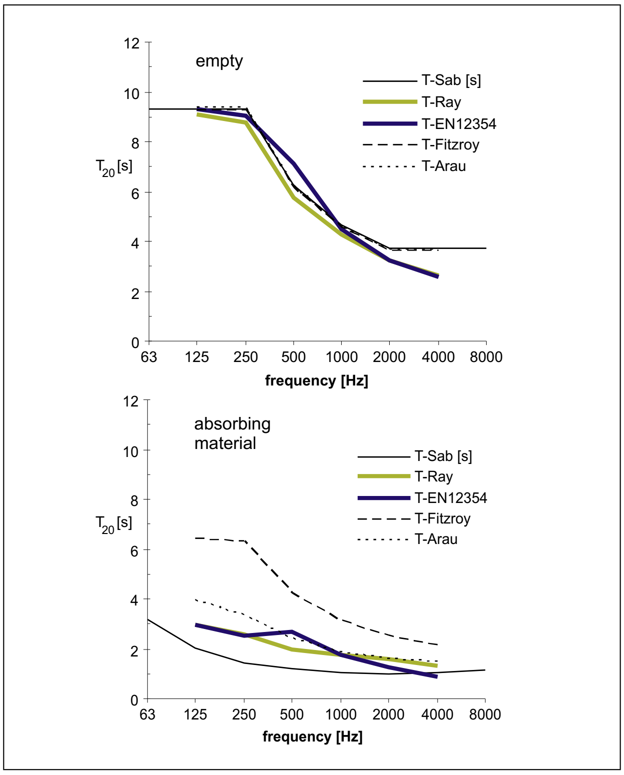

Figure 6. Large cubic room (reverberation room); empty and with absorbing material (A1, A2); reverberation time (T_{20}) according t0 five different models as indicated.

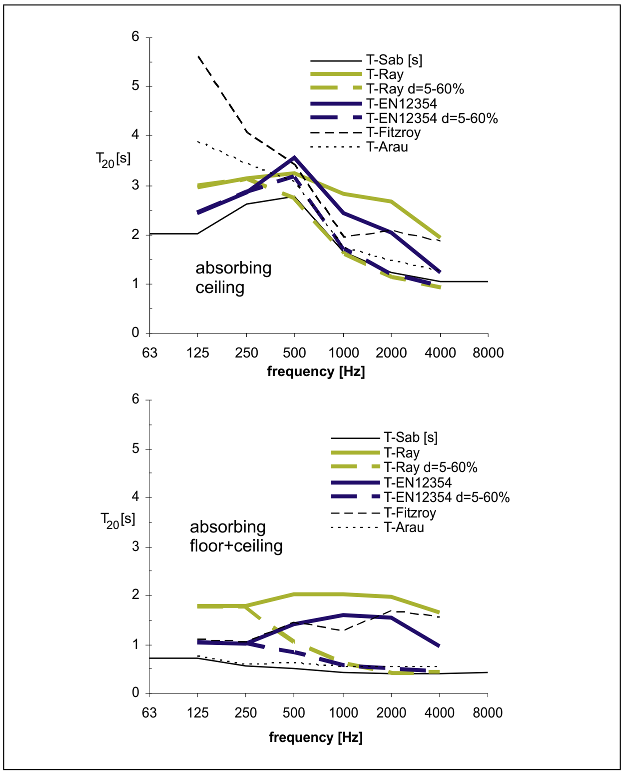

Figure 8. Flat room; absorbing ceiling and additional with absorbing floor (C1, C3); reverberation time (T_{20}) according to five different models as indicated.

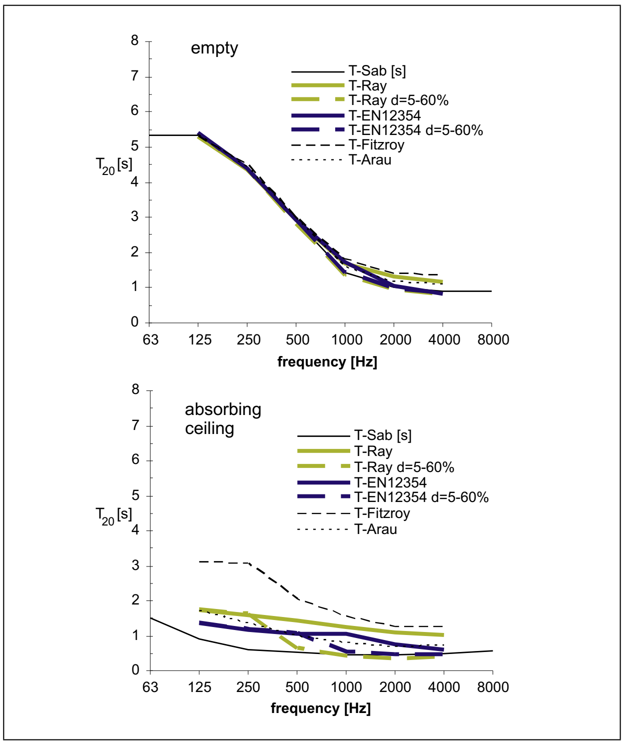

Figure 7. Normal room; empty and with absorbing material (B2, B1); reverberation time (T_{20}) according to five different models as indicated.

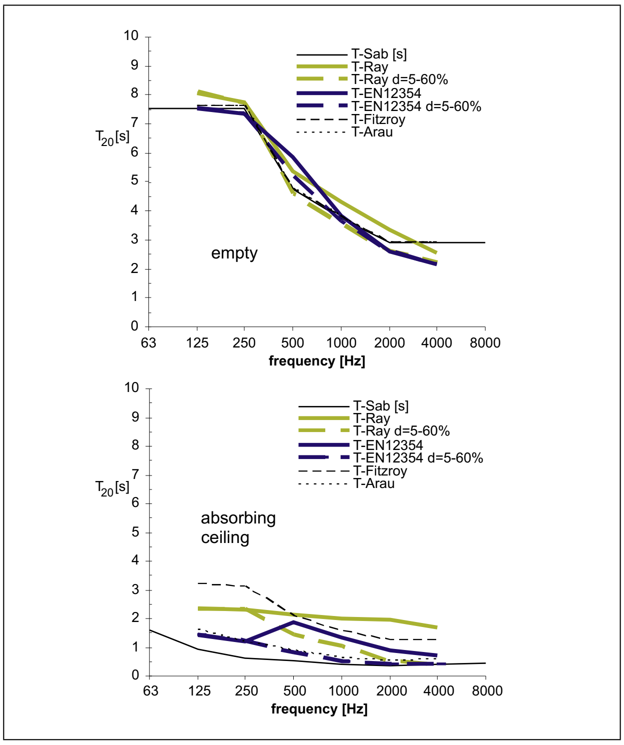

Figure 9. Long room; empty and with absorbing material (D1, D3); reverberation time (T_{20}) according to five different models as indicated.

The resulting curves for 8 of these situations are given for all the models in Figures 6, 7, 8 and 9. The predictions are both done for the situation with low scattering for all surfaces and for the situations with higher scattering for

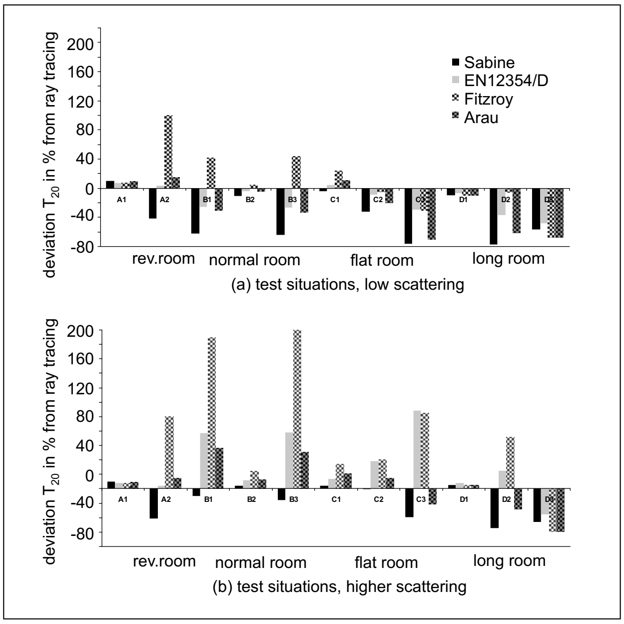

Figure 10. Average deviation in percentage over 250 Hz to 2000 Hz of T_{20} according to four different models compared to the ray-tracing results for the 11 cases considered; a. low scattering of surfaces (5%), b. higher scattering of surfaces (5%-60%).

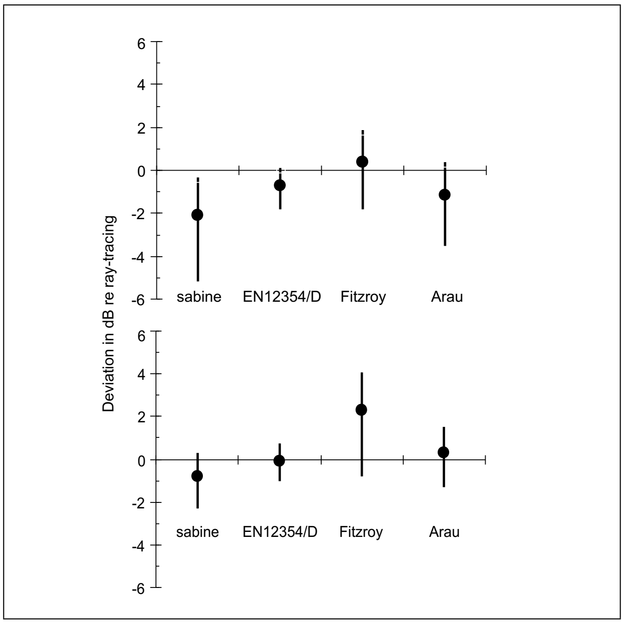

Figure 11. Average deviation in dB over 250 Hz to 2000 Hz for all cases of T_{20} according to four different models compared to the ray-tracing results; low scattering (left) and higher scattering (right).

the walls, except for the reverberation room (Figure 6). It is clear that in case of empty, hard walled spaces the different prediction models hardly differ. That is true for all the room dimensions and shapes considered. But as soon as one or more of the surfaces are absorbing, differences occur that are sometimes quite large. In those cases Sabine gives results that are always clearly lower than those from the ray-tracing model.

To make a global comparison between models easier, Figure 10 gives the average deviation from the ray-tracing results in percentages over the middle frequency range 250 Hz–2000 Hz for all 11 situations as defined in Table I. In Figure 10a this is done for the cases with low scattering and in Figure 10b for the higher scattering cases. In this comparison only the results for EN12354-6, annex D differ, since the other models (Sabine, Fitzroy, Arau) cannot take the scattering into account.

In Figure 11 the average deviation over all cases for each of the prediction models is expressed in dB, often the relevant quantity in noise situations, indicating the average and the deviation around it (+/- a standard deviation). Again, it is clear that in situations with low diffusivity both the deviation and the spread in deviations is smallest for the presented model (EN 12354) and largest for Sabine. In the more realistic scattering situations the difference between models are smaller, but still the EN 12354-model gives the smallest deviation and spread, while now the model of Fitzroy gives the largest deviations.

5. Concluding remarks

Based on this comparison it is concluded that on average the presented model, as it is published in EN 12354-6-annex D, shows the smallest deviation with the smallest standard deviation over all cases, both with and without a higher scattering coefficient for the walls. The model of Fitzroy performs the worse, while in the situation with higher scattering the differences between Sabine and the model from EN 12354-6 annex D and also Arau becomes small, as would be expected.

It would be worthwhile to verify the suggested approach for odd-shaped and occupied spaces as in chapter 3 and look for a combination of the two approaches.

Acknowledgement

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