BUILDING ACOUSTICS AND NOISE CONTROL

Studybook

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Foreword. I use this textbook or parts of it in lectures related to building acoustics, environmental acoustics and noise control. The textbook has no publisher and is not for sale. Unfortunately, there is no English version. I am happy to receive feedback on errors or omissions in the book and try to take it into account when I next update the textbook.

I would like to thank Docent Markku Sainio, Dr. Pekka Saarinen, M.Sc. (Tech.) Ville Rajala and Dr. Henna Maula for reviewing the figures concerning their specialty areas.

The translation of the text from Finnish to English has been made using Word translator. I am sorry about the remaining typos and contextual translation errors. Furthermore, the tables and figures have not been translated.

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Cover photo: "Riddle", Valtteri Hongisto, 2019

1 FOUNDATIONS

1.1 Basic quantities

As a rule, the volume is considered using the sound pressure *p* [Pa]. The pressure variations of normal sounds in our environment are extremely small compared to a static atmospheric pressure of 101300 Pa. As a rule, the sound pressures of sound heard in the living environment are μ between 20 Pa (0 dB) and 20 Pa (120 dB).

The particle velocity *u* [m/s] indicates the rate of oscillation of the molecules relative to the equilibrium axis. The particle velocity should be distinguished from the wave motion propagation velocity *c* [m/s], called the phase velocity.

There is an analogy with electricity, in that the sound pressure corresponds to the voltage and the particle velocity corresponds to the current.

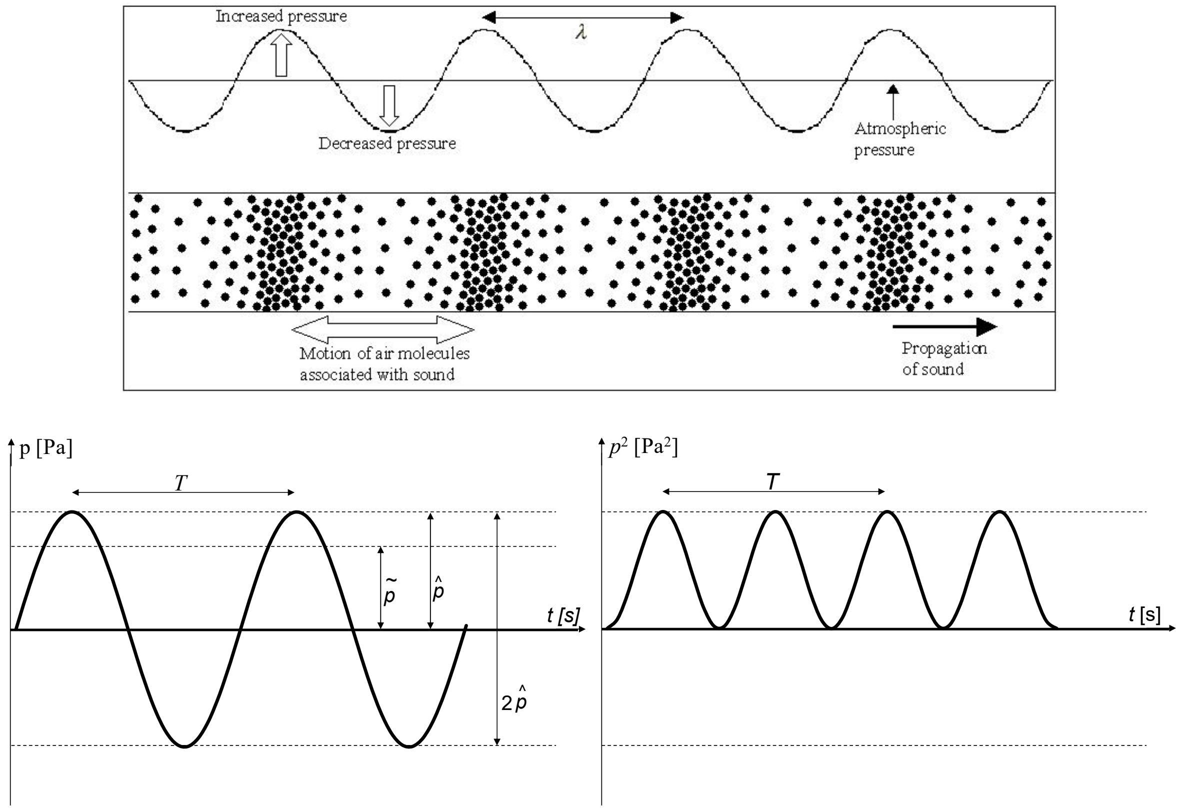


Fig. 1.1.1 (a) Airborne sound is longitudinal or compression wave (Source: medium.com). However, longitudinal wave motion is graphically represented in the same way as transverse wave (Oliva, 2005). (b) The sound pressure of a harmonic sound signal at one frequency as a function of time and the derived peak value (^), root-mean square value (~) and period T of one cycle. This course deals only with harmonic signals, i.e., signals that can be formed as the sum of blue waves. (c). Square of the previous.

Airborne sound is longitudinal wave motion. Fig. 1.1.1 shows the sinusoidal sound signal and its main indicators. The time variation of sound pressure at a given control point is presented as

(1.1.1) $$p(t) = \hat{p} \cdot \sin(\omega t + \phi)$$

where *p*(t) is the sound pressure at time *t* [s], $\hat{p}$ [Pa] is the peak sound pressure, or amplitude, ω=2πf is the angular frequency, [Hz, or 1/s], and ϕ is the phase shift.

The time-averaged sound pressure is given by:

(1.1.2) $$\overline{p} = \frac{1}{T}\int_0^T p(t)\,dt$$

where T [s] is the time period for which you want to determine the mean. When the time period under consideration is long enough, the time average sound pressure is zero.

Instead of the time average pressure, more important is the effective value of sound pressure $\tilde{p}$, i.e., the root mean square (rms), which is related to sound energy. It is defined as the time mean value of squared sound pressure

(1.1.3) $$\tilde{p} = \sqrt{\frac{1}{T}\int_0^T p^2(t)dt}$$

For a harmonic signal, the following relationship between effective and peak value applies:

(1.1.4) $$\tilde{p} = \frac{1}{\sqrt{2}}\hat{p}$$

The effective sound pressure value is used to indicate the average volume. The SPL, L_p [dB], is determined by the effective value

(1.1.5) $$\boxed{L_p = 10 \cdot \log_{10}\left(\frac{\tilde{p}^2}{p_0^2}\right)}$$

where the reference sound pressure is p_0=20 µPa. If necessary, the reference pressure shall be displayed at the end of the dB unit [dB *re* 20 µPa], but normally this does not need to be done as the use of other reference sound pressures is extremely rare. In the case of a reference sound pressure of 20 µPa, L_p = 0 dB. This corresponds fairly well to the sound pressure according to the hearing threshold of a young adult with normal hearing at 1000 Hz.

In a plane wave, the particle velocity and sound pressure are in the same phase. If we consider the wave propagating in the x-direction, then the sound pressure is given by the equation

(1.1.6) $$p(x,t) = \rho_0 c_0 u_x(x,t)$$

where ρ_0 is the density of the medium [kg/m3] and ux [m/s] is the particle velocity in the direction x.

The acoustic characteristic impedance Z (*wave resistance)* of the medium indicates the resistance produced by the medium to sound propagation. It is defined as the ratio of sound pressure and particle velocity, respectively, whereas in electrical science impedance is defined as the ratio of voltage and current (direction of propagation x):

(1.1.7) $$Z_x = \frac{p(x,t)}{u_x(x,t)}$$

Longitudinal wave motion propagates in the air, where the speed of sound is

(1.1.8) $$c_0 = \sqrt{\gamma R T_0} = \sqrt{\frac{\gamma P_0}{\rho_0}}$$

where R =8315 J/mol is the molar gas constant, γ=1.4 for an ideal gas like air, P_0=1013105 Pa is the static atmospheric pressure, ρ_0 is the air density [1.2041 kg/m^3 in 20 ºC], and T_0 [K] is the temperature. The speed of sound in air, c_0 /m/s], depends on the air temperature T [°C] by the equation

(1.1.9) $$c_0 = 331 + 0.6 \cdot T,$$

The characteristic acoustic impedance Z_0 of air at room temperature (20 C°) is $Z_0 = \rho_0 c_0$=412 kg/m^{2}s.

The sound power W [W, J/s] multiplies the total sound energy [J] produced by the sound source per unit t [s]. The measurement of the sound power of the sound source takes place in a free sound field without reflections. A hypothetical measuring surface is outlined around the sound source, inside which the sound source as a whole is located and there are no other sound sources inside the measuring surface. In this case, the sound power of the sound source is determined by means of an intensity over the measuring surface (area S [m^2]) integrated with respect to the surface:

(1.1.10) $$W = \int_S I \cdot dS$$

The intensity in direction x is the product of the particle velocity u_x *of* direction x and the sound pressure p:

(1.1.11) $$W_x = \int_S p \cdot \mathrm{u}_x dS$$

In the free field, the particle velocity is in the same phase as the sound pressure and the time mean (effective value) of the sound power is according to Eq. 1.1.6

(1.1.12) $$\tilde{W} = \frac{1}{T}\int_0^T W(t)\,dt = \frac{1}{T}\int_0^T \frac{p^2(x,t)}{\rho_0 c_0} S dt = \frac{\tilde{p}^2 S}{\rho_0 c_0}$$

Sound power level L_W [dB], **SWL**, is defined by

(1.1.13) $$L_W = 10 \cdot \log_{10} \frac{\tilde{W}}{W_0},$$

where the reference power is defined as W_0=1 pW [dB *re* 1 pW].

Intensity I [W/m^2] is defined as sound power per area S. In the free field, intensity is the product of pressure and particle velocity. The effective intensity is given by:

(1.1.14) $$\tilde{I} = \frac{\tilde{W}}{S} = \frac{\tilde{p}^2}{\rho_0 c_0}$$

Intensity level L_I [dB] is defined by

(1.1.15) $$L_I = 10 \cdot \log_{10} \frac{\tilde{I}}{I_0},$$

where the reference intensity is defined as I_0=1 pW/m^2 [dB re 1 pW/m^2].

The sound power level is determined by closing the sound source inside the measuring surface S [m^2] and determining the mean intensity level L_I [dB] perpendicular to the surface. The sound power level is given by the equation

(1.1.15) $$L_W = L_I + 10 \cdot \log_{10} S$$

If the sound field is non-reflective (free field), the intensity level may be determined from the SPL according to Eq. 1.1.14, where:

(1.1.16) $$L_W = L_p + 10 \cdot \log_{10} S$$

Since the sound power, sound pressure and intensity level have the same unit [dB], conscientious marking of subscripts (L_W, L_p, L_p) is important in written communication to avoid confusion. For example, the equivalent A-weighted SPL in the period 07–22 should be denoted as $L_{pAeq,07\text{-}22}$.

1.2 Addition of levels

Interference refers to a resultant wave that occurs when two or more waves meet each other. Suppose that N pcs of sound waves arrive at the control point at time t, each with a sound pressure of pn(t) following Eq. 1.1.1. The sound pressure interference (resultant), i.e., the total sound pressure $p_{tot}(t)$, is

(1.2.1) $$p_{tot}(t)=\sum_{n=1}^{N} p_n(t)$$

In order to determine the resulting SPL $L_{p,tot}$, the effective value of the total sound pressure shall be known.

The square of the effective value of the interference of two sound pressures 1 and 2 is

(1.2.2) $$\tilde{p}_{tot}^2 = \frac{1}{T}\int_0^T p_{tot}^2(t)\,dt = \frac{1}{T}\int_0^T \left[p_1(t)+p_2(t)\right]^2 dt = \tilde{p}_1^2 + \tilde{p}_2^2 + \frac{2}{T}\int_0^T p_1(t)\,p_2(t)\,dt$$

The first two terms can be found out by measuring the sound pressure of waves 1 and 2. The magnitude of the third term, on the other hand, depends on the phase difference of the waves. Three cases can be distinguished, depending on the correlation between sound sources.

(i) Non-correlated sources

In case (i), sound sources 1 and 2 are completely independent of each other. This is the most common and important case in noise control. On average, the product of sound pressures 1 and 2 is zero, since the phases are independent of each other. Therefore, we obtain

(1.2.3) $$\tilde{p}_{tot}^2 = \tilde{p}_1^2 + \tilde{p}_2^2$$

If the SPL of the waves is known, Eq. 1.1.5 gives the form

(1.2.4) $$\tilde{p}_{tot}^2 = \tilde{p}_1^2 + \tilde{p}_2^2 = p_0^2\left(10^{L_{p,1}/10} + 10^{L_{p,2}/10}\right)$$

and on

(1.2.5) $$\frac{\tilde{p}_{tot}^2}{p_0^2} = \left(10^{L_{p,1}/10} + 10^{L_{p,2}/10}\right)$$

from which taking the 10-base logarithm and multiplying by 10 gives the total SPL of two waves

(1.2.6) $$L_{p,tot} = 10\cdot\log_{10}\left(10^{L_{p,1}/10} + 10^{L_{p,2}/10}\right)$$

The total common sound pressure of an uncorrelated sound source is given by:

(1.2.7) $$\tilde{p}_{tot}^2 = \sum_{n=1}^{N} \tilde{p}_n^2$$

and SPL

(1.2.8) $$\boxed{L_{p,tot} = 10\cdot\log_{10}\left[\sum_{n=1}^{N} 10^{L_{p,n}/10}\right]}$$

This equation is called the **"decibel addition formula".** The equation is applied when the sum of two or more SPLs is to be calculated. The equation can be applied to any level quantity, whether it is sound pressure, intensity or power level. The equation can be applied both to individual frequency bands and to total levels with the same frequency weighting.

(ii) Identical sources, same phase

Case (ii) corresponds to a situation where sound sources are perfectly correlated. An example is two point sound sources 1 and 2, which produce exactly the same sound pressure (amplitude and phase the same). Let's consider the resultant wave of point sound sources, let's look midway between the point sources. Because

(1.2.9) $$p_1(t) = p_2(t)$$

The third term in Eq. 1.2.2 is

(1.2.10) $$\tilde{p}_{tot}^2 = 4\,\tilde{p}_1^2$$

The situation according to the equation corresponds to constructive interference, that is, the complete amplification of waves. The SPL is given by:

(1.2.11) $$L_{p,tot} = 10\log_{10}\frac{4\tilde{p}_1^2}{p_0^2} = L_{p,1} + 6 \quad \text{dB}$$

An important application of case 2 is standing wave. If sound pressure 1 hits the reflecting surface, the SPL at the very opposite surface (less than 1/10 wavelength distance from the surface) is 6 dB higher than the SPL of the wave hitting the surface at the same point without the reflective surface, because the reflecting surface produces a sound pressure 2 with the same amplitude and phase as sound pressure 1. The reflection gain of six decibels is observed the further away from the reflecting surface, the lower the frequency of the sound.

(iii) Identical sources, opposite phase

In case (iii), phase transitions φ_1 and φ_2 of sound sources 1 and 2 differ by 180 degrees. If we continue to assume that the amplitudes of sound pressures 1 and 2 are the same, then

(1.2.12) $$p_1(t) = -p_2(t).$$

The sound pressure is given by equation **1.2.2** then:

(1.2.13) $$\tilde{p}_{tot}^2 = \frac{1}{T}\int_0^T [p_1(t) + p_2(t)]^2\, dt = 0$$

and the total SPL

(1.2.14) $$L_{p,tot} = -\infty$$

This is equivalent to destructive interference, where waves 1 and 2 cancel each other out. This is the aim of active noise control applications and resonators.

1.3 Decibel addition applications

1.3.1. Average of levels

Level averages should be lowered in the following situations, for example:

- You want to specify octave band values from tersic band values
- For several sound level measurements, you want to determine the average value

The average value is determined by the equation

(1.3.1) $$L_{p,tot} = 10\cdot\log_{10}\left[\frac{1}{N}\sum_{n=1}^{N}10^{L_{p,n}/10}\right]$$

1.3.2 Background noise correction

Consider sound source 1, the measurement of which is disturbed by background noise, which is indicated as source 2. If the common SPL of sound sources 1 and 2 is $L_{p,tot}$ [dB] and the background noise pressure level is $L_{p,2}$ [dB], the SPL of sound source 1 is obtained

(1.3.2) $$L_{p,1} = 10\log_{10}\left(10^{L_{p,tot}/10} - 10^{L_{p,2}/10}\right)$$

The maximum permissible value for background noise correction, i.e. correction to $L_{p,tot}$ is -3.0 dB (engineering grade measurements) and -1.3 dB *(*precision measurements*)*.

1.3.3. Equivalent level

A significant part of the regulations, guidelines and recommendations concerning noise concerns the equivalent SPL, i.e., the time average SPL over a given period of time *T*.

The equivalent level $L_{eq,T}$ of an audio signal refers to the level of a constant signal Y that is equivalent in energy to the time mean of the level specified over period T of the sound signal. The equivalent level for sound pressure is given by the equation

$$L_{p,eq,T} = 10\log_{10}\left(\frac{1}{T}\int_0^T \frac{p^2(t)}{p_0^2}dt\right) \tag{1.3.3}$$

If we know the SPL as a function of time, we get the equivalent level from the equation

$$L_{p,eq,T} = 10\log_{10}\left(\frac{1}{T}\int_0^T 10^{L_p(t)/10}dt\right) \tag{1.3.4}$$

If the equivalent level $L_{eq,Ti}$ in time periods T_i is known, the equivalent level is given by:

$$L_{eq,T} = 10\log_{10}\left[\frac{1}{\sum_{i=1}^{N} T_i}\sum_{i=1}^{N} T_i \cdot 10^{L_{eq,T_i}/10}\right] \tag{1.3.5}$$

The equivalent level can also be determined by the intensity or sound power level.

For example, according to Finnish legislation, environmental noise is examined separately during daytime (07–22, T=15 h) and night time (22–07, T=9 h). Workplace noise, on the other hand, is usually examined over a period of 8 hours.

1.4. Frequency analysis

Consider a situation where the piston oscillates harmoniously in an inflatable tube f times per second. This causes a longitudinal plane wave in the pipe (Fig. 1.1.1a). The speed of movement of the piston corresponds to the particle velocity u of the oscillation of the plane wave and is a vector quantity. The wavelength [m] of longitudinal wave motion λ is

$$\lambda = c_0 T = \frac{c_0}{f} \tag{1.4.1}$$

where c_0 [m/s] is the speed of sound propagation in air and f [Hz] is the frequency.

The sounds that occur in our environment usually contain several frequency components. Only certain whistle sounds contain only one frequency component. All periodic signals can be represented as the sum of harmonic signals. The periodic signal repeats itself in the period T

$$p(t) = p(t+nT),\ n = 1,2,3,... \tag{1.4.2}$$

Approximately periodic signals are common in technical applications. For example, machine sounds are generated by periodic oscillations formed by rotating parts, as well as periodic pressure waves. Even variable sounds are almost periodic, when the sound is viewed in a sufficiently short time window.

According to the Fourier principle, each periodic signal can be expressed as the sum of N pcs harmonic signals in the so-called "harmonic signals". In the form of the Fourier series

$$p(t) = \sum_{n=1}^{N} \hat{p}_n \cos\left(2\pi n f_0 t + \varphi_n\right) \tag{1.4.3}$$

where $\hat{p}_n$ is the peak value of the harmonic signal component n and f_0=1/T. In the equation nf_0 are harmonic multiples of the base frequency and φ_n is the phase difference of the component. FFT analysis (*Fast Fourier Transform*) can be used to find out the amplitude of each frequency.

Fig. **1.4.1** shows examples of signal and frequency analyses of single and two sine waves. If the frequency resolution of FFT analysis is set to 1 Hz, 19980 data points are obtained when the conventional hearing range 20–20000 Hz is analyzed. This amount of information cannot be easily reported. In noise control, the results are primarily presented using percentage-band filters instead of FFT analysis (Fig. **1.4.2**) in order to have less information but sufficient for most needs. Octave and third octave filters are percentage-band filters. However, the percentage band values determined by the frequency analyzer are based on FFT analysis.

Fig. **1.4.2** shows mathematical definitions of percentage bands, medium frequencies and frequency bands. One octave band includes three third-octave bands. The level L_o of the octave band can be formed from the level of the three third octave bands $L_{t,i}$ contained in the octave band by summing up

(1.4.4) $$L_o = 10\log_{10}\sum_{i=n-1}^{n+1} 10^{L_{t,i}/10}$$

Table **1.4.1** shows standardised medium frequencies and frequency bands which differ slightly from mathematical definitions but are used in all noise abatement applications.

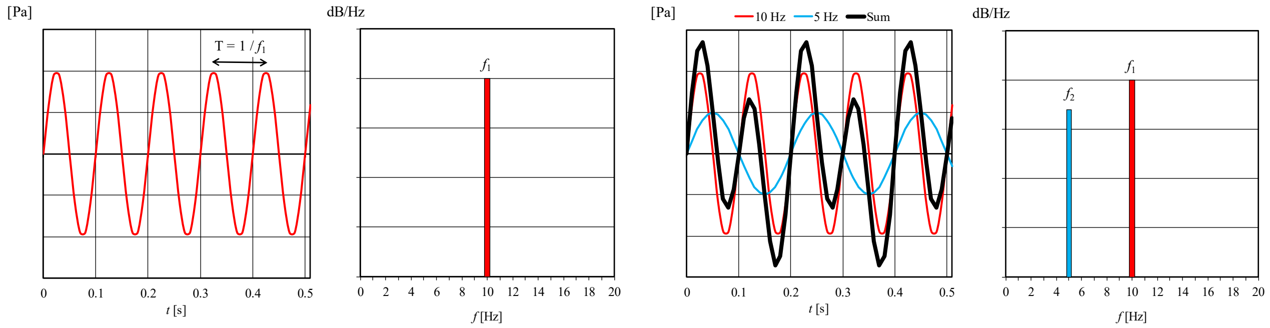


Fig. 1.4.1 **(**Top row) Sound pressure of a harmonic sine wave (10 Hz) as a function of time and SPL as a function of frequency. (Bottom row) Two sine waves of different amplitude and frequency and their sum.

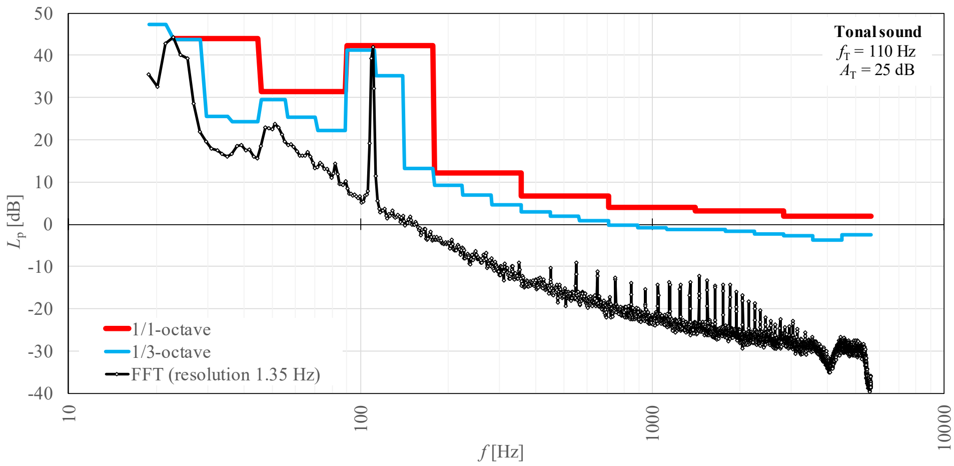
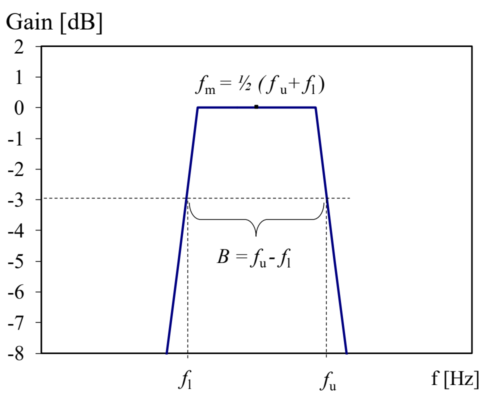


Fig. 1.4.2 FFT, one-third octave band, and octave band analysis of a tonal sound.



	1/1-octave filter	1/3-octave filter
Lower frequency limit	$f_l = f_m/\sqrt{2}$	$f_l = f_m/\sqrt[6]{2}$
Upper frequency limit	$f_u = \sqrt{2}f_m$	$f_u = \sqrt[6]{2}f_m$
Bandwidth	$B = f_u - f_l$	$B = f_u - f_l$
Middle frequency	$f_m = \sqrt{f_l f_u}$	$f_m = \sqrt{f_l f_u}$

Fig. 1.4.3 The parameters used to describe the characteristics of the frequency filter are upper limit frequency f_u, lower limit frequency f_l, bandwidth B, and medium frequency f_m.

Table 1.4.1 Nominal values of octave and terse bands. Sequential octave bands have different background colors. The mid frequencies of the octave bands are in bold.

f_m	1/3-octave range	1/1-octave range	f_m	1/3-octave range	1/1-octave range
1.6	1.41 - 1.78		200	178 - 224	
2	1.78 - 2.24	1.41 - 2.82	**250**	224 - 282	178 - 355
2.5	2.24 - 2.82		315	282 - 355	
3.15	2.82 - 3.55		400	355 - 447	
4	3.55 - 4.47	2.82 - 5.62	**500**	447 - 562	355 - 708
5	4.47 - 5.62		630	562 - 708	
6.3	5.62 - 7.08		800	708 - 891	
8	7.08 - 8.91	5.62 - 11.2	**1000**	891 - 1120	708 - 1410
10	8.91 - 11.2		1250	1120 - 1410	
12.5	11.2 - 14.1		1600	1410 - 1780	
16	14.1 - 17.8	11.2 - 22.4	**2000**	1780 - 2240	1410 - 2820
20	17.8 - 22.4		2500	2240 - 2820	
25	22.4 - 28.2		3150	2820 - 3550	
31.5	28.2 - 35.5	22.4 - 44.7	**4000**	3550 - 4470	2820 - 5620
40	35.5 - 44.7		5000	4470 - 5620	
50	44.7 - 56.2		6300	5620 - 7080	
63	56.2 - 70.8	44.7 - 89.1	**8000**	7080 - 8910	5620 - 11200
80	70.8 - 89.1		10000	8910 - 11200	
100	89.1 - 112		12500	11200 - 14100	
125	112 - 141	89.1 - 178	**16000**	14100 - 17800	11200 - 22400
160	141 - 178		20000	17800 - 22400	

1.5 Hearing threshold and constant loudness curves

Hearing threshold level (HTL) refers to the lowest SPL a person can hear. HTL has been studied the most for tone frequencies from 20 to 16 000 Hz. These results have been compiled in ISO 389-7 and ISO 226 standards. However, the HTL is also defined from the frequency range 2 to 16 Hz.

Fig. **1.5.1** shows the standardised hearing threshold and the infrasound hearing threshold. Standardised values are also given in Table **1.5.1.** The sensitivity of the sense of hearing is very different at different frequencies. Hearing is at its most sensitive at frequencies between 250 and 8 000 Hz, where speech is mainly located. Outside this range, the hearing threshold begins to rise. Here, the hearing threshold refers to the hearing threshold of a young adult (18 years old) with normal hearing. With age, the hearing threshold rises, especially at high frequencies.

The standardised constant noise curves are shown in Fig. **1.5.2.** A constant loudness curve means that sine tones located on the same curve are perceived as equally loud. The unit of loudness is the phon. Standard loudness curves are based on several independent psychoacoustic laboratory experiments. There has been sufficient data available in the frequency band from 20 to 12 500 Hz, so this range has been standardized. The sense of hearing is called nonlinear because constant volume curves have different shapes at different volumes, and constant loudness curves are located at different frequencies at different distances from each other. For example, an increase in SPL of five decibels by 20 Hz seems as strong as an increase in SPL of ten decibels by 1 000 Hz. The pain threshold is highly individual and there is less research data on it than on the hearing threshold or standard loudness curves.

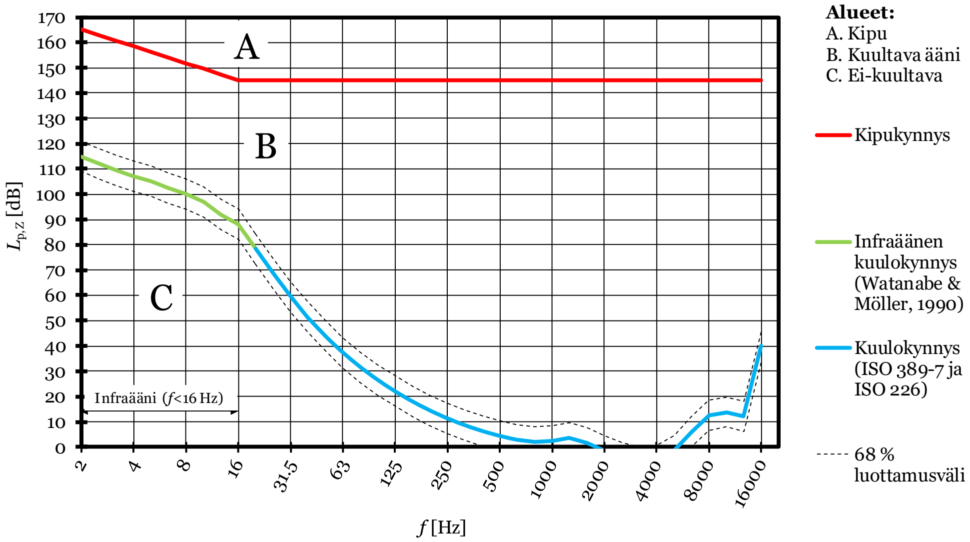


Fig. 1.5.1. Unweighted SPL $L_{p,Z}$ of the hearing threshold level and pain threshold level of a young adult with normal hearing as a function of frequency, *f*. The SPL below the blue-green line is 'non-audible' (C), the area above the blue-green area is audible up to the red line (B) and the SPL above the red-green zone is pain-inducing SPL (A).

Table 1.5.1. Standardized hearing threshold level according to ISO 389-7. The hearing threshold is set for direct frontal sine sound, which is listened to with two ears (*free field listening, frontal incidence*). If the sound is broadband, the hearing thresholds are approximately 5 dB higher than shown here.

f [Hz]	20	25	31.5	40	50	63	80	100	125	160	200	250	315	400	500
HT [dB]	78.5	68.7	59.5	51.1	44.0	37.5	31.5	26.5	22.1	17.9	14.4	11.4	8.6	6.2	4.4

f [Hz]	630	800	1000	1250	1600	2000	2500	3150	4000	5000	6300	8000	10000	12500	16000
HT [dB]	3.0	2.2	2.4	3.5	1.7	-1.3	-4.2	-6.0	-5.4	-1.5	6.0	12.6	13.9	12.3	40.2

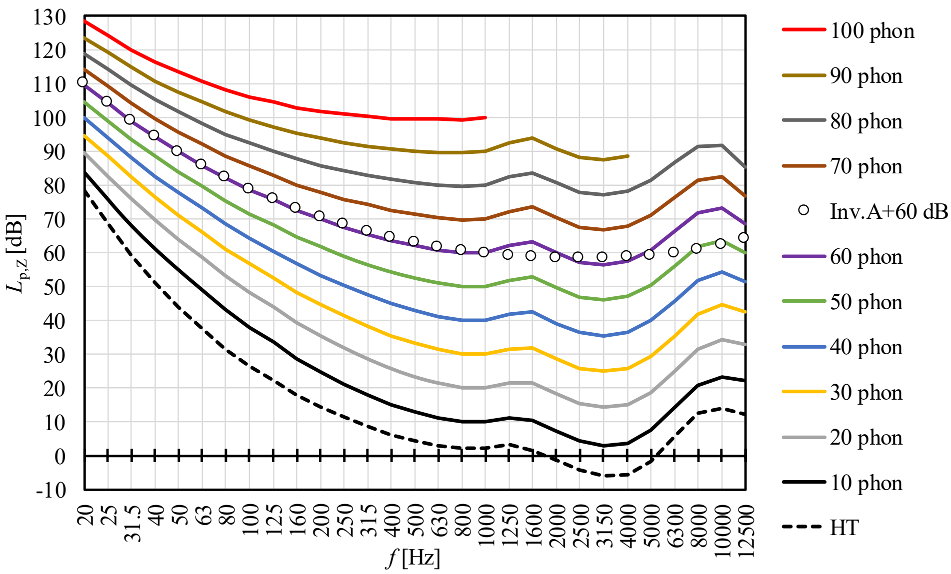


Fig. 1.5.2. Unweighted SPL $L_{p,Z}$ of constant loud curves as a function of frequency *f* according to ISO 226 (lines). In addition, the hearing threshold (HT) and the inverse A-weighting curve plus 60 dB (Inv.A+60dB) are presented. It seems to follow a curve of 60 phons.

1.6. Frequency weightings

In noise control, frequency weightings are used to represent the total SPL in a given frequency band using a single digit value. The main weightings are Z, A and C.

- Z-weighting refers to a linear or unweighted plane in the frequency range 10–20 000 Hz, so it is used in this textbook whenever an unweighted plane is meant.

- A-weighting is used to assess how loud a person perceives sound.
- C-weighting is sometimes used to assess strong impulse noise and low-frequency noise.

There are other weightings (B, D, E, F, G) but they are not used in Finnish guideline values.

The A and C weighting values are given in Table **1.6.1** by steel band. The A-weighted value of frequency band i is obtained by adding the weighting value of that band A_i to the level $L_{Z,i}$ of the linear frequency band:

(1.6.1) $$L_{A,i} = L_{Z,i} + L_{A,i}$$

The total A-weighted SPL for a frequency range containing N bands is calculated by the equation

(1.6.2) $$L_{A,tot} = 10 \cdot \log_{10}\left(\sum_{i=1}^{N} 10^{(L_{A,i})/10}\right)$$

Similarly, the C-weighted value of frequency band i is obtained by adding the weighting value C_i of that band to the level $L_{Z,i}$ of the linear frequency band:

(1.6.3) $$L_{C,i} = L_{Z,i} + L_{C,i}$$

The total C-weighted SPL for a frequency range containing N bands is calculated by the equation

(1.6.4) $$L_{C,tot} = 10 \cdot \log_{10}\left(\sum_{i=1}^{N} 10^{(L_{C,i})/10}\right)$$

A-weighting is a compromise of standard loudness curves. The A-weighting fits best on the 60 phon curve. Because of this, A-weighting may not describe perceived loudness well with sounds quieter or louder than 60 phons. C-weighting is not based on standard loudness curves.

As a rule, the A- or C-weighting of a noise meter looks at the frequency range 20□20000 Hz. Narrower frequency ranges are often used in different noise control applications and product specifications. For example

- Ventilation sound calculations: octave bands 63−8000
- Airborne and impact sound insulation measurements: measurements in the one third octave bands 50 5000 Hz
- Absorption coefficient measurements: one-third octave bands 100 to 5 000 Hz, notification values in octave bands 125 to 4 000 Hz
- speech transmission index: octave bands 125–8 000 Hz

To this end, it shall be indicated in the calculations from which frequency range the total weighted sound level has been determined. If sound occurs significantly outside the frequency band under consideration, the total sound level in the frequency band under consideration underestimates the audible SPL. Such an error may occur, for example, if the dimensioning calculations for the airborne sound insulation of the façade of the building have been made in the frequency range 50–5 000 Hz, for which airborne sound insulation values for the façade structure are available, but audible environmental noise is present to a significant extent below 50 Hz.

It should be noted that $L_{A,tot}$ may be higher than $L_{Z,tot}$ if the noise is concentrated in the frequency range 1 to 5 kHz.

It is not meaningful to consider the A-weighted value in frequency bands where the linear SPL does not exceed the hearing threshold in Table **1.5.1.** This is relevant when assessing, for example, the indication of noise emission values for silent equipment or the presentation of measurement results for low-frequency noise. There are no standard guidelines for these situations, so practices vary.

Table 1.6.1. Standardized A and C weighting factors for different steel bands *en*. When using octave bands, bold values apply. The Z-weighting factor Z_i is 0 dB for all bands.

i	f_n	A_i	C_i
	[Hz]	[dB]	[dB]
1	20	-50.4	-6.2
2	25	-44.7	-4.4
3	**31.50**	**-39.4**	**-3.0**
4	40	-34.6	-2.0
5	50	-30.2	-1.3
6	**63**	**-26.2**	**-0.8**
7	80	-22.5	-0.5
8	100	-19.1	-0.3
9	**125**	**-16.1**	**-0.2**
10	160	-13.4	-0.1
11	200	-10.9	0.0
12	**250**	**-8.6**	**0.0**
13	315	-6.6	0.0
14	400	-4.8	0.0
15	**500**	**-3.2**	**0.0**
16	630	-1.9	0.0

i	f_n	A_i	C_i
	[Hz]	[dB]	[dB]
17	800	-0.8	0.0
18	**1000**	**0.0**	**0.0**
19	1250	0.6	0.0
20	1600	1.0	-0.1
21	**2000**	**1.2**	**-0.2**
22	2500	1.3	-0.3
23	3150	1.2	-0.5
24	**4000**	**1.0**	**-0.8**
25	5000	0.5	-1.3
26	6300	-0.1	-2.0
27	**8000**	**-1.1**	**-3.0**
28	10000	-2.5	-4.4
29	12500	-4.3	-6.2
30	**16000**	**-6.6**	**-8.5**
31	20000	-9.3	-11.2

1.7 Time weightings

SPL measurements often require time weighting according to IEC 61672-1. The options are F (Fast) and S (Slow). Time weightings include ascent and descent constants of different lengths (τ=125 μσ ανδ τ=1000 μσ). The operation of time weightings is illustrated in Fig. **1.7.1.** In the case of a step rise in the SPL, it takes about 0.5 seconds with F time weighting and about 4 seconds with S time weighting to reach the maximum SPL of the step. The time weightings are symmetrical, i.e. the ascent and descent time constants are the same.

The Fast time constant is thought to correspond well to how quickly a person senses changes in SPLs. Therefore, maximum sound levels in the living environment are usually measured using Fast time weighting. The sound level meter, which follows the Fast time constant, updates the reading faster than the eye's ability to register. Slower Slow time weighting was developed to make it easier to follow the hands of analog meters in case of variable noise. Slow time weighting still has a use, for example, in assessing the sound level of short-term and non-impulsive noise events, such as passing by rolling stock.

The Fast time constant was considered too slow to assess the risk of hearing damage for impulse noise with a high rate of rise (>100 dB/s). For this purpose, I (Imp) time weighting was developed, which is asymmetrical: the rise and fall time constants are 35 and 1500 ms. The impulse time constant is no longer supported in new sound level meters. When assessing the risk of impulse noise hearing damage, the peak SPL Lpeak,max is determined over the measurement period using a sampling rate of 44 kHz and no separate time weighting is required.

Time constant and sampling interval are not the same thing. In general, sound level monitoring with the Fast time constant is updated every 125 ms, i.e. the sampling interval is the same as the rise time constant of the Fast time constant. However, this does not always have to be the case. For example, when estimating impulsive noise, NT ACOU 112 (Nordtest, 2002) states that sampling should take place every 10–20 ms in order to analyse rapid changes in SPLs in more detail.

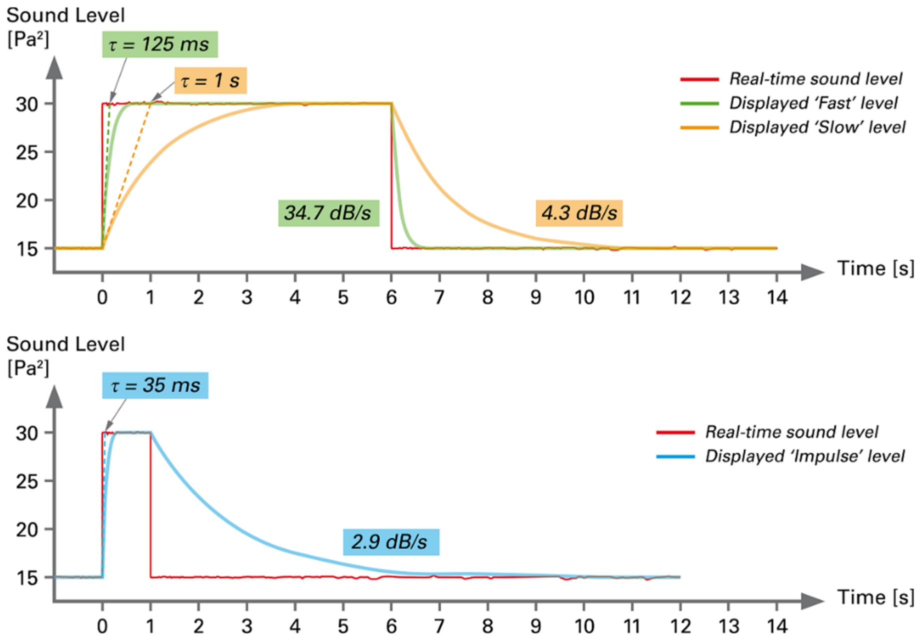


Fig. 1.7.1. Fast, Slow, and Impulse on the use of time constants for the step function. NTi-Audio AG is appreciated (Source: nti-audio.com).

LITERATURE

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2. ENVIRONMENTAL NOISE

This chapter focuses on examining the spread of environmental noise. Noise abatement measures are listed, but not discussed in detail.

2.1. Types of noise and noise control measures

Noise refers to unwanted sound. Viewing sound as noise is individual and situational. Noise is usually considered noise if it is unnecessary for you and interferes with your own ongoing activity. Ambient noise caused by the activities of people or equipment is often perceived as noise if it exceeds the guideline values set for the SPL. Or, conversely, the guideline values have been drawn up in such a way that a sound below the guideline values is rarely perceived as harmful noise.

Although exposure to environmental noise occurs at work, in leisure environments and in residential buildings, the effects of environmental noise are primarily examined in residential buildings and their yards.

Causes of environmental noise include:

- Road traffic
- Air traffic
- Rail transport
- Waterway transport
- Industry and power plants
- Infrastructure construction sites: blasting, quarrying, aggregate production
- Construction sites
- Shooting ranges and firing ranges for heavy weapons
- Motor carriageways
- Logistics centres, rail, road and air traffic stations, chartering, ports
- HVAC equipment (environmental sound) in conventional buildings
- Public events

Sound is usually transmitted to the listener by air, but vibrations on the ground can also cause vibrations in the frame of the building to be so strong that they can be heard (frame sound).

Each type of environmental noise is subject to its own legislation, measurement and modelling procedures. The regulations and guidelines are summarised in **Appendix 1**. Environmental noise must not exceed the guide values set out therein.

Noise abatement measures include:

- **Land use planning**. New sound sources shall be placed as far away from exposed areas as possible. New exposed areas shall be located as far away from sound sources as possible.
- **Placement of building masses**. Buildings with no or higher exposure limit values shall be placed between the exposed object and the noise source. These include, for example, business premises and garages.
- **Screen time limitation.** The noise emission of a sound source may be limited by setting limits relating to the time at which the noise emission is produced. These include, for example, operating bans related to evenings, nights or weekends mentioned in the environmental permit.
- **Limitation of sound emission.** In some cases, the noise emission of a sound source can be reduced directly by reducing production power. Road traffic noise can be reduced by limiting traffic volume, vehicle types and speed.
- **Technical noise control.** It is possible to reduce the noise emission of stationary sound sources with technical noise abatement solutions built on site, such as noise boxes or silencers.
- **Choosing or encouraging the choice of quieter devices.** The most effective way to achieve noise effects is to purchase quieter sound sources. For example, aircraft sound power levels have been reduced by up to 10 dB because many airports have set traffic charges according to noise emissions.

- **Limiting the spread of noise.** For example, a noise barrier (noise railing, noise wall or earth embankment) can be placed between the sound source and the exposed area, or the location of the sound source can be chosen so that existing landforms can achieve a noise barrier-like effect. The noise barrier must not cause the guideline values to be exceeded in another direction due to reflections from it.
- **Facade sound insulation.** The façade structure can have a significant impact on the SPL inside the building. An open window dampens noise by about 15 dB. The best possible facades with windows absorb sound up to more than 45 dB L_{Aeq}.
- **Balcony soundproofing.** The balcony is considered a courtyard area. Balcony glazing may be required to differ in sound levels up to ΔL=15 dB L_{Aeq} (Kovalainen and Kylliäinen, 2016).
- **Change of ownership of real estate.** Noise abatement is also achieved if the number of people exposed decreases. In some cases, the person causing the noise or the municipality may buy or expropriate the property where noise problems have been experienced because it is more expensive or impossible to apply other noise abatement measures. The sound remains, but the noise effects are eliminated.

2.2. Noise indicators

Finnish legislation on environmental noise and construction uses guideline or action values related to the following key figures:

- $L_{Aeq,T}$, i.e., the A-weighted average sound level during the period T (**Chapter 1**);
- L_{prm}, i.e., the maximum A-weighted hull noise level L_{ASmax} measured with the slow time constant (Ministry of the Environment, 2017);
- L_{AFmax}, i.e., the maximum A-weighted sound level measured with the fast time constant (Ministry of Social Affairs and Health, 2015; Ministry of the Environment, 2017);
- $L_{eq,1h,f}$, i.e., the unweighted equivalent maximum hourly SPL in the terse band f (frequencies 20–200 Hz) (Ministry of Social Affairs and Health, 2015);
- L_{Rden}, i.e., annual average sound level, which is a daily noise level subject to impulse noise correction of 15 dB and weekend correction of 5 dB (Government, 2017);
- L_{AE}, i.e., the A-weighted sound exposure level caused by a noise event (Valvira 2016);
- L_{CE}, i.e., C-weighted sound exposure level caused by heavy weapons and blasting activities (Government, 2017);
- L_{AImax}, i.e., the maximum A-weighted SPL measured with an impulse time constant (Government, 1997)

In Finland, guideline values have been set separately for night and day equivalent A-weighted SPLs. When assessing the health effects of noise, it is most advantageous to use a single figure. Noise exposure is increasingly described using daily noise levels based on A-weighted equivalent levels for different parts of the day. For example, day-evening-night noise levels L_{den} are used in noise surveys in accordance with the Environmental Noise Directive.

In the day-night noise level L_{dn} [dB] (day, night), the night-time average sound level includes a +10 dB penalty (correction, adjustment):

(2.2.1)
$$L_{dn} = 10 \cdot \log_{10}\left[\frac{1}{24}\left(15 \cdot 10^{L_{A,eq,07-22}/10} + 9 \cdot 10^{\left(L_{A,eq,22-07}+10\right)/10}\right)\right]$$

In the day-evening-night noise level L_{den} [dB] (day, evening, night) in the evening time (19–22), a penalty of +5 dB is included in addition to the above:

(2.2.2)
$$L_{den} = 10 \cdot \log_{10}\left[\frac{1}{24}\left(12 \cdot 10^{L_{A,eq,07-19}/10} + 3 \cdot 10^{\left(L_{A,eq,19-22}+5\right)/10} + 9 \cdot 10^{\left(L_{A,eq,22-07}+10\right)/10}\right)\right]$$

Perceptions of the duration of day, evening and night time vary from country to country, but the principle is the same.

The equivalent level is poorly suited for measuring the pass-by noise of trains, for example. The noise of a given train type (specific speed, length and rolling stock) is described by the sound exposure level L_{AE} [dB]:

(2.2.3) $$L_{AE} = L_{Aeq,T} + 10 \cdot \log_{10}\left(\frac{T}{t_0}\right)$$

where t_0=1 s. The sound exposure level thus indicates the A-weighted SPL that a sound event lasting one second should have in order to have the same total energy as a sound event lasting time *T* with an A-weighted equivalent level of $L_{Aeq,T}$ [dB]. The sound exposure level is also used in measurements of heavy weapons, detonations and aircraft overflights.

2.3. Geometric attenuation

This section does not take into account atmospheric absorption and reflection of the earth's surface.

(i) Point source

A sound source can be considered point-like when the maximum diameter of the sound source is 5−10 times less than the distance of the observation point from the centre of the sound source. In addition, the observation point should be located at least a wavelength away from the sound source, since there may be vortices in the nearby field. Vortices are caused by the interaction between different parts of the radiant surface.

A free sound field means that the sound can propagate freely and there are no reflections associated with the sound field. In a free sound field, the SPL caused by a point sound source is attenuated by 6 dB for geometric reasons when the distance to the sound source doubles. The SPL at the distance *r* [m] from the sound source is calculated by the equation

(2.3.1) $$L_p = L_W + D_{ge} = L_W + 10 \cdot \log_{10}\left[\frac{k}{\Omega\, r^2}\right]$$

where L_W [dB] is the sound power level of the sound source, D_{ge} *[dB]* is the geometric (geometric) damping factor, *k* is the directional coefficient of the sound source in the direction under consideration and *Ω* is the space angle to which sound from the point source radiates. The last two are defined in the following chapters.

If directivity is described by the direction index L_k [dB],

(2.3.2) $$L_k = 10 \cdot \log_{10}(k)$$

Eq. 2.3.1 can be expressed as

(2.3.3) $$L_p = L_W + L_k + 10 \log_{10}\left[\frac{1}{\Omega r^2}\right]$$

(ii) Line source

If the sound source is line-shaped, such as a road, for example, it makes no sense to determine the sound power level of the sound source, since it depends on the length of the line. For geometric reasons, the SPL of a line sound source is reduced by only 3 dB when the distance to the sound source is doubled. The SPL $L_{p,2}$ [dB] at distance r_2 [m] from the sound source is given by:

(2.3.4) $$L_{p,2} = L_{p,1} - 10 \cdot \log_{10}\left(\frac{r_2}{r_1}\right),$$

where $L_{p,1}$ [dB] is the so-called emission SPL at a distance of r_1 [m] from the line source. Emission SPL means the SPL measured in a free field relatively close to the sound source, minus the 3 dB increase in SPL due to reflection on the ground/floor.

In calculation software, the line source is modeled using nearby point sources placed on the line. The sound power of each point source shall be set so that the desired emission SPL, $L_{p,1}$, is achieved.

2.4. Coefficient of direction

The coefficient of direction x, k_x, is the ratio of the radiating intensity $I_{x,r}$ (distance to the centre of the sound source *r*) radiating in direction x and the average intensity I_r radiating in all directions

$$k_x = \frac{I_{x,r}}{I_r} \tag{2.4.1}$$

The direction factor is strongly frequency-dependent. Typically, directivity in the direction of the oscillatory axis of the oscillating body increases as the frequency increases. The direction factor is determined in an anechoic chamber or in a free field.

The direction factor is 1 for a point sound source, so that the directional pattern of sound radiation is spherical. For example, a spherical speaker consisting of 12 drivers, used in building acoustic measurements, is a point sound source below 5000 Hz. At frequencies higher than this, directivity occurs in the direction of the oscillation axis of each speaker element. Fig. 2.4.1 shows an example of how the direction factor is presented.

For sources of environmental noise, directional information on sound sources is rarely already available. It may therefore be necessary to determine the direction factor by means of on-the-spot measurements.

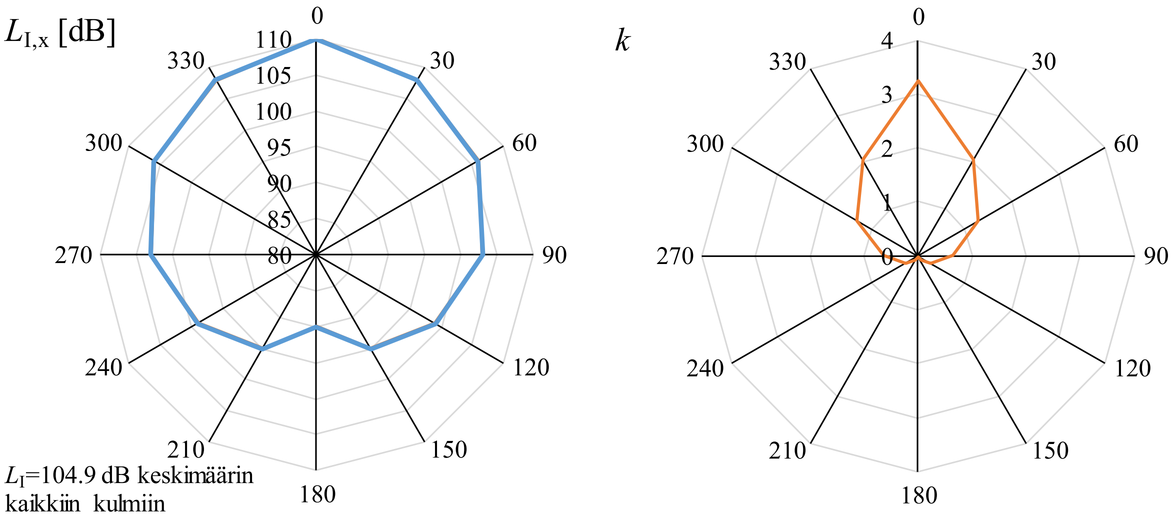


Fig. 2.4.1. The directivity properties of a sound source in the horizontal plane every 30 degrees. On the left, the intensity level $L_{I,x}$ at different angles x. On the right, the direction coefficient k derived from the previous one at different angles. The angle 0° is k=3.25 and L_k=5.1 dB.

2.5 Space angle

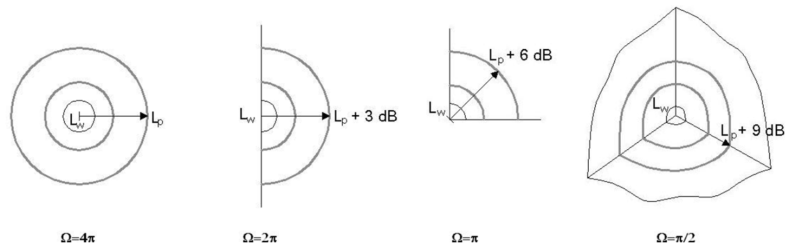
A sound source can exert very different sound pressure on its surroundings, depending on how the sound source is located in relation to reflective surfaces in its immediate vicinity. For this reason, sound pressure calculations always require information about the solid angle to which the sound source can freely radiate.

The space angle Ω is the ratio of the part of space under consideration to the whole of space, multiplied by the constant 4π. Table 2.5.1 shows the most typical values of the solid angle and the resulting increase in SPL in equation 2.3.1**.** In other words, the solid angle takes into account reflections in the near-field field of the sound source, and they no longer need to be taken into account separately in SPL calculations. The solid angle affects only direct sound. In room space, the effect of the term does not extend beyond the echo radius.

It should be noted that if the sound power level of a sound source has been determined at the same limited space angle as it is in the modelled environment, and these reflective surfaces were part of the hypothetical measurement surface of the sound power level (see Chapter 1), the confirmations in Table 2.5.1 are already included in the value of the sound power level L_W. Then the solid angle is set to 4π in Eq. **2.3.1.**

Table 2.5.1. The most commonly used values of the space angle Ω according to the location of the sound source.

4π	Source is far from surfaces, e.g. wind turbine
2π	Source is near to one surface, e.g. source on the ground
π	Source is near to two surfaces, e.g. washing machine
π/2	Source is near to three surfaces, e.g. freezer



2.6. Atmospheric absorption

The atmosphere absorbs a constant amount of sound per meter. The absorption attenuation D_{atm} [dB] is calculated by

(2.6.1) $$D_{atm} = \frac{\alpha r}{1000}$$

where r [m] is the distance to the sound source, and α [dB/km] is the tabulated absorption of air. Some values of α are shown in Table **2.6.1.** Atmospheric absorption is almost negligible at low frequencies, while at high frequencies atmospheric absorption is a more significant damping factor at large distances from the sound source than geometric attenuation.

Table 2.6.1. Atmospheric absorption values according to ISO 9613-2 α with different air temperature *T* and relative humidity *RH* values.

T	*RH*	α [dB/km]							
[°C]	[%]	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
10	70	0.1	0.4	1	1.9	3.7	9.7	32.8	117
20	70	0.1	0.3	1.1	2.8	5	9	22.9	76.6
30	70	0.1	0.3	1	3.1	7.4	12.7	23.1	59.3
15	20	0.3	0.6	1.2	2.7	8.2	28.2	88.8	202
15	50	0.1	0.5	1.2	2.2	4.2	10.8	36.2	129
15	80	0.1	0.3	1.1	2.4	4.1	8.3	23.7	82.8

2.7. Soil reflection and ground absorption

There is always at least one ground reflection between the sound source and the listener, which is taken into account in the calculations (Fig. **2.7.1**). Reflection obeys Snell's law, i.e. the angle of incidence is the same as the angle of reflection. If the sound source is high, the ground reflection usually takes place quite close to the listener, which by default is located at a height of 4 meters, unless otherwise decided. If, on the other hand, the sound source is close to the ground, the ground reflection takes place halfway between the sound source and the listener. The contours of the ground determine the reflection site, and therefore the absorption of the earth's surface must be felt throughout the studied area.

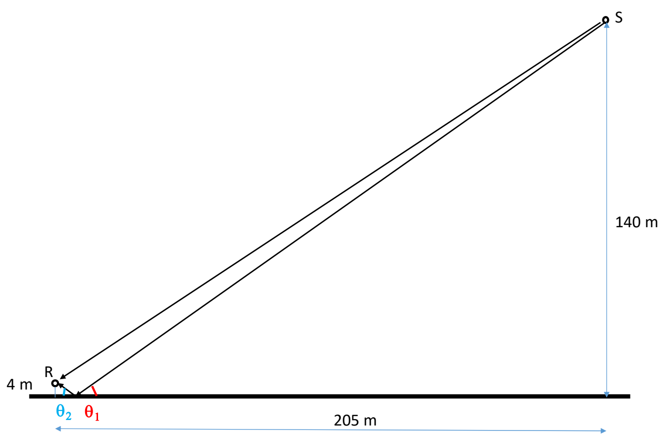


Fig. 2.7.1. An example of the location of a ground reflection when the sound source (S) is a wind turbine at an altitude of 140 m and the listener (R) is at an altitude of 4 m 205 m from the base of the wind turbine.

The absorption ratio is defined in Ch. 3. Although the absorption ratio is frequency dependent, environmental noise models do not take frequency dependence into account and often select the ground absorption ratio (α_G) from one of the following constant values:

- 0.0: hard (asphalt road, water)
- 0.3: mainly hard (sand fields, downtown areas)
- 0.7: predominantly soft (loose urban areas)
- 1.0: soft (soft ground)

In modelling programs, frequency-independent absorption ratio is often also denoted by G.

Due to absorption, the intensity level $L_{I,r}$ of the sound reflected from the ground is less than the intensity level $L_{I,i}$ of sound reflected from the ground by D_{gr} [dB]:

(2.7.1) $$D_{gr} = -10 \cdot \log_{10}\left(1-\alpha_G\right)$$

For example, the ground reflection of a wind turbine occurs close to the observation point (Fig. **2.7.1**). At very low frequencies (less than 50 Hz), the observation point is located closer to the ground than a quarter of the wavelength, so that the observation point is located at the maximum point of the standing wave hitting the ground, where the SPL is 6 dB higher (constructive interference of simultaneous phase signals). Therefore, at low frequencies, the values in Table **2.7.1** are used for ground reflection damping. The values concern a situation where modelling is carried out at a height of 4 m above the ground. However, measurements are made at a height of 1.5–2.0 m above the ground.

Table 2.7.1. Values of ground reflection damping.

	***f* [Hz]**	**20**	**25**	**31.5**	**40**	**50**	**63**	**80**	**100**	**125**	**160**	**200**	**250**	**315**	**400**
Maa	D_{gr} [dB]	5.6	5.4	5.2	5	4.7	4.3	3.7	3	1.8	0	0	0	0	0
Vesi	D_{gr} [dB]	6	6	5.9	5.9	5.8	5.7	5.5	5.2	4.7	4	3	3	3	3

	***f* [Hz]**	**500**	**630**	**800**	**1000**	**1250**	**1600**	**2000**	**2500**	**3150**	**4000**	**5000**	**6300**	**8000**	**10000**
Maa	D_{gr} [dB]	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Vesi	D_{gr} [dB]	3	3	3	3	3	3	3	3	3	3	3	3	3	3

2.8 Vegetation

Vegetation between the sound source and the control point has a significant effect on the SPL at the observation point if the vegetation is between the direct sound beam connecting the sound source and the control point. If the sound source is very high (wind turbine), then the effect of vegetation will be quite small. If the sound source is close to the ground, the effect of vegetation can be very large.

Fang and Ling (2003) experimentally determined the excess attenuation of 35 different vegetation types for sound excitation according to the spectrum of road traffic noise. Sound absorption of 20 meters per vegetation zone ranged from 1 to 9 dB. They divided the results into 3 groups:

- Group 1: Sound absorption > 6 dB/20 m. The vegetation consisted of shrubs. Visibility was less than 5 meters.
- Group 2: Sound attenuation 3–6 dB. The vegetation consisted of trees and shrubs. Visibility was 6–19 m.
- Group 3: Sound absorption less than 3 dB. The vegetation consisted of sparsely located trees and shrubs. Visibility was more than 20 meters.

Dense scrub was the best way to absorb road traffic noise. The sound-absorbing capacity of vegetation D_{veg} [dB/20m] could be predicted by what is the visibility through vegetation V_{veg} [m]:

(2.8.1) $$D_{veg} = -1.9\ln\left(V_{veg}\right) + 9$$

Visibility varies locally and it is not possible to easily determine a suitable sound attenuation group for large areas of land. Vegetation is thinned from time to time, and D_{veg} is not constant in terms of time. Therefore, it is not worth overestimating the sound absorption of vegetation, so as not to underestimate the SPLs behind the vegetation.

Sound is also refracted over vegetation, so vegetation cannot be used to suppress sound indefinitely.

2.9 Noise barrier

A noise barrier means a structure that prevents sound from being transmitted directly from the sound source to the receiver. A noise barrier cannot be given an unambiguous sound attenuation value, because the sound attenuation achieved by a noise barrier depends on the position of the sound source and listener in relation to the noise barrier. Therefore, the insertion produced by noise barrier against a fixed noise source is determined by

(2.9.1) $$D_{nb,tot} = L_{p,1} - L_{p,2}$$

where $L_{p,1}$ [dB] is the SPL in the listener's position without noise barrier and $L_{p,2}$ [dB] with the noise noise barrier is present. In both measurements, the position of the rest of the environment and the sound source, as well as the sound power and directional pattern of the sound source, must remain the same. The total sound reduction is affected by both the refractive sound (*diffraction*) and the sound passing through the noise barrier.

For a point sound source, the sound attenuation D_{diff} [dB] of the refractive sound of the noise barrier can be estimated by the formula

(2.9.2) $$D_{diff} = 10\log_{10}\left(1 + 20\frac{z}{\lambda}\right)$$

where z is the distance difference [m] and λ is the wavelength of sound [m]. The distance difference z is geometrically determined as shown in Fig. **2.9.1.** The greater the distance difference, the greater the D_{diff} (Fig. **2.9.2**). Sound attenuation is increased by raising the noise barrier, but also by shifting the sound source or control point towards the noise barrier. The maximum sound reduction of the noise barrier is achieved immediately behind the noise barrier.

The acoustic properties of the noise barrier, i.e. airborne sound insulation and sound absorption ratio, must be known so that the effects of the noise barrier on the environment as a whole can be assessed.

The total sound attenuation generated by sound refracting through and over the noise barrier shall be calculated by frequency using the equation

(2.9.3) $$D_{nb,tot} = -10 \cdot \log_{10}\left(10^{-R/10} + 10^{-D_{diff}/10}\right),$$

where R [dB] is the sound reduction index of the noise barrier. This is discussed in more detail in Ch. 3.

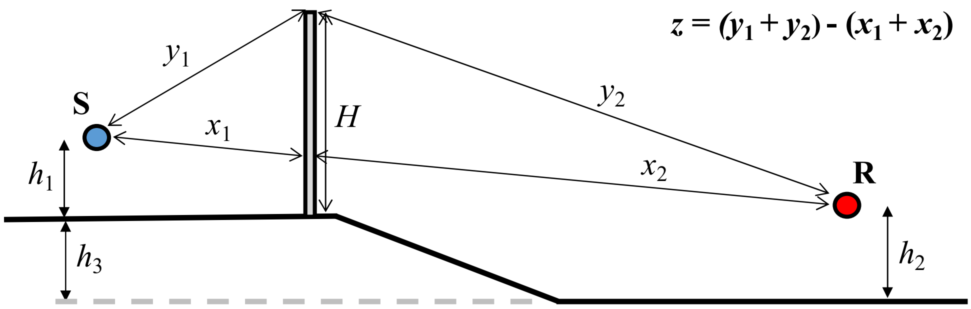


Fig. 2.9.1. The distance difference *z* of a noise barrier is the difference between the length of the sound path folding over the noise barrier (y_1+y_2) and the length of the straight sound path (x_1+x_2). The distance difference is solved with the help of trigonometry. S is the sound source and R is the receiver.

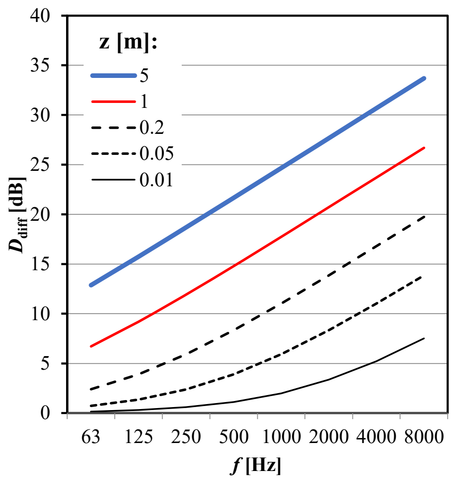


Fig. 2.9.2. The effect of the distance difference z [m] created by the noise barrier to a given control point on sound attenuation at different frequencies, assuming that no sound passes through or past the noise barrier.

2.10. Effect of weather conditions

The speed and direction of the wind, as well as the gradient of the wind speed in the direction of height, affect the propagation of sound. So does the temperature gradient. However, these factors are not taken into account. Therefore, measurements aimed at verifying the results of the models should be made when the wind speed is less than 3 m/s. An exception is wind turbine noise, where modelling is carried out using maximum noise emissions. Wind turbine noise is measured when the wind speed is sufficient to provide maximum power generation (usually 12 m/s at the height of the wind pole).

2.11. Measurement uncertainty

Environmental noise measurements, as well as SPL measurements in general, must indicate the uncertainty in decibels. Measurement uncertainty refers to the dispersion that occurs in measurements of the same phenomenon if the measurement assignment were repeated by dozens of different surveyors.

The total uncertainty U [dB] is given by:

(2.11.1) $$U = \sqrt{\sum_{i=1}^{N} u_i^2}$$

where u_i [dB] is the estimated uncertainty of component i. The measurement uncertainty of environmental noise is higher than that of indoor noise, because the weather conditions affect how sound travels from the sound source to the measuring point. Elements include:

- uncertainty related to the calibration of the measuring equipment,
- selection of the measuring point,
- the magnitude of the background noise correction,
- duration of measurement (a short measurement is more uncertain than a long one),
- Weather conditions
- distance to sound source

Uncertainties (L_{Aeq}) are at least the following:

- 1.0 dB in anechoic and anechoic laboratory rooms,
- room 1.5 dB,
- outdoors in a free anti-reflective field just outside the sound source 1.5 dB,
- outdoors in a free anti-reflective field more than 30 meters from a sound source of 2 dB.

Far from the sound source, it is possible to estimate the measurement uncertainty (L_{Aeq}) according to Table 2.11.1.

In order to reliably compare the measurement result with the guideline values, the measurement uncertainty must be known. Fig. 2.11.1 shows how the comparison could be made.

Table 2.11.1. Estimation of the measurement uncertainty of the A-weighted equivalent SPL based on the number of measurements and the measuring distance.

Etäisyys lähteeseen [m]	30	100	500	100	500
Riippumattomien mittausten lukumäärä	1	1	1	6	4
Epävarmuus ΔL [dB]	2	4	7	2	4

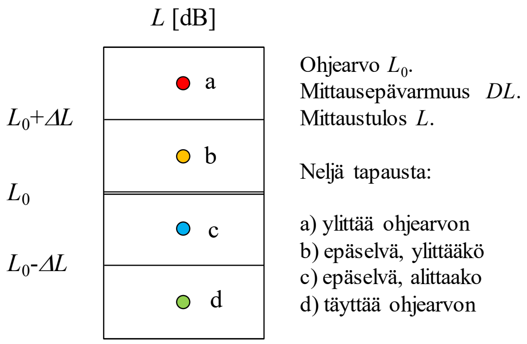


Fig. 2.11.1. Comparison of the measurement result L with the guideline value L_0. In order for the guideline value to be exceeded, the measurement result must exceed $L_0+\Delta L$. Below the guideline value, the measurement result must be below $L_0-\Delta L$.

2.13. Noise propagation modelling

Calculation of the SPL to an individual observation point is possible with moderate accuracy by applying the equations described above. The SPL of environmental noise is usually indicated using SPL maps. They are drawn up using a dedicated noise dispersion calculation programme (hereinafter: a programme) which allows flexibility in the choice of calculation parameters, standards to be used and presentations of results. Below is a concise description of how modeling proceeds with CadnaA.

To start using the program, download a base map (mapinfo, .mif) of the area to be viewed, including buildings, roads and contours. The base map is available by e-mail free of charge on the National Land Survey's website. The coordinate system of the base map must correspond to the coordinate system of the software.

On the base map, each line has a type (building, elevation, road, railway, waterline, etc.). The base map is not directly usable for noise calculations. Right from the start, the line types used by the National Land Survey of Finland are converted based on their ID number into the language of the line types used by the program, so that the road referred to by the NLS corresponds to the road understood by the program, and so on.

The floor map does not include building heights. The heights of buildings are estimated depending on the type of building (number of storeys multiplied by three meters). The National Land Survey of Finland offers freely available laser scanning data that describes the ground surface and the heights of objects on the ground. The data consists of measurement points, each of which has xyz-coortinate data. The resolution is at its densest 0.5 points per square meter, which means that elevation data can be found approximately every 1.4 m or more. It allows building heights to be set for a large number of buildings based on the coordinate point of each building. The coordinate systems must be consistent. Buildings can also be defined as sound-reflective and give an absorption ratio in reflection.

The elevation reference of 0 m is usually the height of sea level. The heights of the ground, sound sources, fairways and observation points should be checked to ensure that they are at the correct height in relation to the height reference.

The terrain is modelled using elevation points contained in discrete elevation curves. From nearby elevation points, the program forms triangles that cover the entire terrain to be modeled. The height contours are therefore in the software for illustrative purposes only. The smaller the triangles, the more accurate the terrain model will be. An accurate terrain model should be sought when the sound source is low. Special emphasis is placed on altitude resolution near the sound source and also near the viewing point.

If necessary, the elevation curves can be made denser, for example, by temporarily gluing a more accurate topographic map on top of the map and drawing new elevation curves with it. A decrease in density is necessary if it is estimated to achieve a more reliable modelling result. This may be the case, for example, if the sound source is located close to a cliff or other obstacle that significantly affects the propagation of sound.

Special attention should be paid to the description of the sound source and its ground surface, as well as the surrounding area. If the sound source is a road, the height of the road surface must correspond to its actual height, including any elevations in the road surface, and if there are rock walls or noise barriers in the vicinity of the road, they should be examined with particular care. Tunnels must be taken into account separately.

The area to be examined and its immediate surroundings are demarcated from the topographic map and other map information is excluded from the programme.

The method of calculating the SPL is selected to serve each purpose. There are options for different international standards as well as national or Nordic procedures. These also include assumptions about frequency range and resolution.

The characteristics of the sound source (location, location, sound emission) are set according to the available information. Point sound sources are assigned a sound power level by frequency band at the resolution required by the selected calculation model. If necessary, the sound source can be given the times when it is active. If only the A-weighted sound power level of the sound source is known, octave band values should be estimated according to the best available information (literature, previous measurements). If there is information about the sound source in one-third octave band, they must be converted to octave band values using the decibel addition formula.

In the case of road traffic, the total number of vehicle pass-bys at different times of the day, the share of heavy traffic from the previous one, the speed of traffic with both volumes of traffic and the type of road surface are described on an hourly basis. This information can be obtained from the Finnish Transport Infrastructure Agency or the municipality. In addition, junction and hill areas can be marked separately if the additional noise generated by them is to be taken into account in the calculations. The road is broken down into different parts as the feature changes. By default, the height of the noise source is 0.5 meters from the surface of the ground at the fairway.

In the case of rail traffic, track locations and heights are obtained from city data or laser scanning data. The shape of the embankment (platform and ramps) is shaped manually. Traffic data, which include train types, lengths and speeds, is obtained from the railway operator (VR Track Oy). Separate noise emission data are available for train types and are made available. The lower speeds used near the railway yard areas are assessed on an empirical basis. In the case of bridges, the noise emission shall be increased by the necessary amount. For example, a 3 dB increase is used for some steel bridges. Local and intermittent rail congestion is taken into account on a case-by-case basis. By default, the height of the noise source is 0.5 meters from the surface of the ground at the fairway.

Industrial noise shall be modelled by identifying significant noise sources and setting appropriate sound power levels based on literature, sound source-specific measurement data or estimates of remote field measurements. Sound sources in industrial buildings consist of point, line and surface sources. Surface sources are usually the outer walls or roofs of buildings, in which case the actual sound source is inside the building. In this case, the program can be supplied with the sound power level inside the building, as well as the absorption spectrum of the wall. Some common sources of industrial noise, such as chimneys, are directional, in which case the direction factor should be taken into account. The modeling program has extensive built-in libraries for common industrial noise sources, wall structures and directional patterns. They can be used if measurement data is not available (rarely there are and you can't or can't go to the spout of the barrel).

Air transport is not covered here, as it is significantly more complex and less common than the previous ones.

The absorption ratio on the ground is set according to the requirements of the calculation method. The program sets a default value, eg. G=1, for the whole area. Deviating areas must be separately demarcated and defined. These include, for example, sound-reflecting water surfaces.

The program sets parameters that affect the calculation, such as the maximum distance between the sound source and the calculation point and the maximum number of reflections of a single sound beam. In general, default values are acceptable, but the applicability of the assumptions must be checked.

Vegetation zones and their sound absorption are set according to the available information.

The atmospheric temperature and relative humidity are set according to the requirements of the noise type and calculation standard, regardless of what conditions actually prevail.

Noise barriers such as noise barriers, noise barriers and noise walls are placed at their correct heights and at the correct distance from the sound source. The width of the ridge is also indicated for the noise barrier, and the reflection coefficient for railings and walls.

As a result of the calculation, the desired SPL quantity (L_{den}, $L_{\mathrm{Aeq,07\text{-}22}}$, etc.) is selected according to the purpose. Because, for example, $\mathrm{L_{den}}$ is defined differently in different countries, the definition is made in the programme separately from the hour intervals for which the sanction is imposed.

If necessary, the number of occupants can be set in buildings, in which case the program can provide an analysis of how many people are exposed to different SPLs. Municipal building and dwelling register data are used for this.

When calculating constant SPL maps, the graphical representation of the results (lines, colour charts) and the density of the calculation network shall be selected. The viewing height in relation to the height of the ground is set in accordance with the guidelines of the calculation standard or selected appropriately.

The programme can produce different types of results:

- Horizontal constant SPL maps with desired resolution and height
- Vertical constant SPL maps
- Tabulated constant SPLs at desired points
- SPLs on the façade surfaces of certain buildings at different floor heights
- Number of exposed inhabitants in different SPL categories
- Color-coded elevation map of the terrain

In addition, it is possible to make simple 3D images and passing animations. In each situation, it is possible to determine separately which SPL meter is in question. This leads to the fact that the number of results is very large. When reporting results, particular attention shall therefore be paid to the fact that the outcome variable is indicated. An example of the noise map of an entire city is given in Turun kaupunki (2017).

The buildings themselves are also sources of noise, since the façade has ventilation terminals and other HVAC equipment. HVAC noise generated by a building in the environment is subject to relatively strict noise regulations, as noise is generated throughout the day. The easiest way to model the noise emissions caused by a building to the plots and façade surfaces of nearby neighbours is with the software described above.

Hongisto et al. (2017) have compared wind turbine noise modelling results with measurement results at several measurement points and in several wind power areas. According to them, the modelling results do not fundamentally differ from the measured values.

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3 SOUND ABSORPTION

3.1. Sound absorption and insulation

Each layer of material or structure both isolates and absorbs sound (Fig. 3.1.1). Good sound insulation is best achieved with a massive and dense layer of sheets. A good sound insulation reflects most of the sound back to its direction of input (impedance significantly higher without impedance) and the minimum amount of sound passes through the structure as indicated by the transmission ratio. Sound energy losses are small when the reflection ratio is high.

Good absorption is most easily achieved with porous panel layers that are directly limited to air. A good absorbent allows sound to enter almost without resistance (characteristic impedance value close to no value) and converts acoustic energy into thermal energy, so that as sound passes through the material, the amount of sound power is reduced by the absorption ratio. Sound is converted into heat because the pores of the material create friction with the particle vibration inside the material.

Absorption structures include:

1. porous materials such as wool, padding and fabrics
2. Helmholz resonator lattices (perforated panel resonators) and
3. panel resonators

Absorption materials are combined with coatings that improve appearance, durability or other properties. They should be optimised so as not to excessively impair sound absorption capacity.

This chapter presents theory and equations that allow to understand the operation of the most common absorption structures.

This chapter has attempted to bold complex numbers.

The two most important acoustic properties of a material (or structure) are sound absorption ratio α and sound reduction index, R (see Ch. 5). The sound absorption of the material is used to influence the sound field formed inside the space. In absorption, sound energy is converted into thermal energy. The sound insulation of a material is used to influence the transmission of sound to the other side of the material (or structure). In sound insulation, sound energy hitting the structure can be converted into heat (if, for example, there are absorbent materials inside the structure) or reflected back, so that energy is not lost.

Consider the situation shown in Fig. 3.1.1 where the surface of the structure is subjected to intensity *Ii*. The surface of the structure reflects back the intensity *Ir* and the structure permeates intensity *It*. The absorption ratio of sound α is determined by the equation

(3.1.1) $$\alpha = \frac{I_i - I_r}{I_i}$$

The sound absorption ratio is dimensionless. The higher the value, the better the structure absorbs sound. It can get values between 0.00... 0.99. The absorption ratio is usually strongly dependent on frequency. The attenuation caused by the absorption ratio D_I [dB] in the level of intensity of reflected sound is determined by the equation

(3.1.2) $$D_I = L_{I_i} - L_{I_r} = -10 \cdot \log_{10}\left(1-\alpha\right)$$

The airborne sound insulation [dB] is determined by the equation

(3.1.3) $$R = 10 \cdot \log_{10}\left(\frac{1}{\tau}\right) = 10 \cdot \log_{10}\left(\frac{I_i}{I_t}\right)$$

where τ is the throughput. Airborne sound insulation is usually strongly dependent on frequency. Airborne sound insulation values are usually between 0 and 70 dB. Values above 70 dB are rare, and values higher than this cannot be measured reliably either.

Airborne sound insulation is determined on a logarithmic dB scale for the reason that the values of the transmission ratio are very small, there are differences of several orders of magnitude in the values and it is not very practical to indicate them.

The typical range of absorption and penetration ratios is shown in Table **3.1.1.** Absorption usually involves power losses several orders of magnitude less than airborne sound insulation. The values of the absorption ratio of materials are usually less than α<0.90, in which case the effect on reflected sound power is less than 10 dB. In turn, the values of the airborne sound insulation of structures are usually more than 30 dB, which means that the penetration ratio is less than one thousandth.

Table 3.1.1. The typical range of absorption and penetration ratios and their corresponding decibel values.

α	D [dB]
0.00	0.0
0.10	0.5
0.20	1.0
0.30	1.5
0.40	2.2
0.50	3.0
0.60	4.0
0.70	5.2
0.80	7.0
0.90	10.0
0.99	20.0
0.999	30.0

τ	R [dB]
1	0
0.1	10
0.01	20
0.001	30
0.0001	40
0.00001	50
0.000001	60
0.0000001	70
0.00000001	80
0.000000001	90

Airborne sound insulation and sound absorption always take place simultaneously in the structures. However, the magnitude of absorption and insulation can vary significantly depending on the structural layers and their placement. Fig. 3.1.1 shows the four most typical options.

A. Hard surface and dense panel on both sides. The surface absorption ratio is very low, less than 0.10. Depending on the surface mass, the sound reduction index R can be 0–90 dB. (1 kHz)
B. A wall of sheet structure with absorbent material in the cavity. The situation is not fundamentally different from structure type A. The absorption material in the cavity does not improve the absorption ratio observed on the surface because the material is located behind the surface panel. However, the absorption material reduces cavity reverberation, which improves airborne sound insulation.
C. A wall with absorbent material on the surface. In this case, both a high absorption ratio and high airborne sound insulation are achieved. (1 kHz)
D. Absorbent material only. In this case, a high absorption ratio is achieved, but poor airborne sound insulation. Absorption materials alone are poor sound insulators (see section B) because they easily allow sound to pass through pores to one side of the structure. (1 kHz)

As a concept, airborne sound insulation shall be distinguished from sound absorption, although in cases B–D the absorption material also causes an increase in airborne sound insulation.

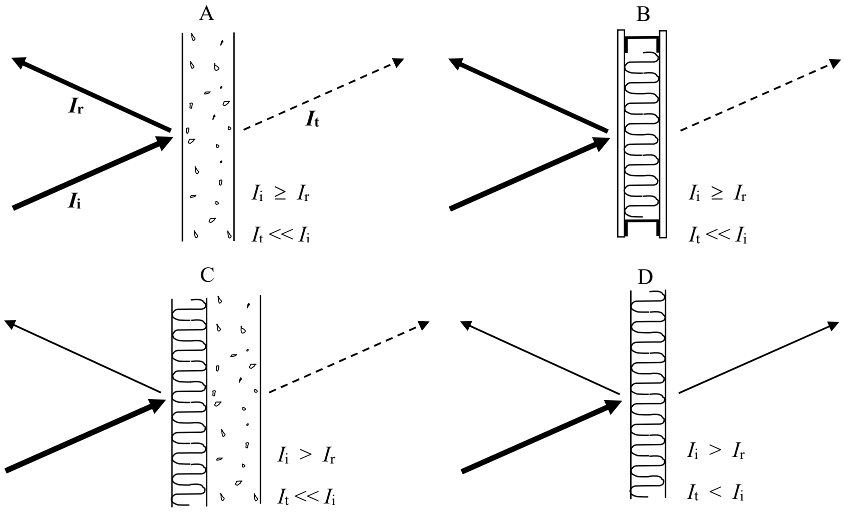


Fig. 3.1.1 Absorption and airborne sound insulation in infrastructure A, B and D and composite structure C.

3.2 Wavenumber and impedance

The acoustic properties of the material are described by the characteristic propagation constant $\mathbf{\Gamma}_a$ [1/m

(3.2.1) $$\mathbf{\Gamma}_a = \Gamma'_a + j\Gamma''_a$$

] and with characteristic impedance **Za** [Pa·s/m]

(3.2.2) $$\mathbf{Z}_a = Z'_a + jZ''_a$$

The characteristic impedance $\mathbf{Z_a}$ [Pa·s/m] of the material tells us the relationship between pressure and particle velocity to the plane wave propagating in the material:

(3.2.7) $$\mathbf{Z}_a = \frac{\mathbf{p}(x,t)}{\mathbf{u}(x,t)}$$

The characteristic impedance is most easily determined with an impedance tube.

In the air, the propagation constant is

(3.2.3) $$\mathbf{\Gamma} = \Gamma' + j\Gamma'' \cong jk_0$$

where k_0 is the wavenumber [1/m]

(3.2.4) $$k_0 = \frac{2\pi}{\lambda} = \frac{\omega}{c_0} = \frac{2\pi f}{c_0}$$

and f [Hz] is the frequency and c_0 [m/s] is the speed of sound in air.

Losses in room air space are very small compared to losses in room surfaces. Therefore, the real part of the propagation constant can be considered zero. In air, the sound wave propagating in the direction of the positive x-axis is represented as

(3.2.5) $$p(x,t) = \hat{p}e^{j(\omega t - k_0 x)}$$

The wave presented in this way does not fade as it progresses.

Porous material has strong losses, so the use of wavenumber alone is not enough. Losses are shown in the real part of the propagation constant. Sound pressure inside the absorbing material is formed

(3.2.6) $$p(x,t) = \hat{p}e^{-\mathbf{\Gamma}_a x}e^{j\omega t} = \hat{p}e^{-\Gamma'_a x}e^{-j\Gamma''_a x}e^{j\omega t}$$

The first exponential term is real and not oscillating. The next two terms are oscillating, causing a damping wave motion as it progresses.

The real part Γ' of the propagation constant is determined, for example, by measuring the attenuation of the SPL in a thick material sample. The imaginary part Γ'' of the propagation constant is determined by measuring the phase change as a function of distance.

3.3 Reflection from the material

(I) *PERPENDICULAR* ANGLE OF INCIDENCE

Consider the harmonic wave motion that enters material 0 perpendicular to the interface of another elastic material 1 (Fig**. 3.3.1**). Materials are considered infinitely large, in which case there are no reflections inside them other than the waves considered here. Set the coordinate system so that x=0 is located at the interface. The characteristic impedances of the materials are $\mathbf{Z}_{a0}$ and $\mathbf{Z}_{a1}$. The plane wave propagates in the direction of the positive x-axis. The sound pressure of an incident wave is

(3.3.1) $$\mathbf{p}_i(x,t)=\hat{p}_i e^{i(\omega t-k_0 x)}$$

and particle velocity (vector quantity)

(3.3.2) $$\mathbf{u}_i(x,t)=\frac{\hat{p}_i}{\mathbf{Z}_{a0}}e^{i(\omega t-k_0 x)}$$

The reflected wave is correspondingly obtained by:

(3.3.3) $$\mathbf{p}_r(x,t)=\hat{p}_r e^{i(\omega t+k_0 x)}$$

(3.3.4) $$\mathbf{u}_r(x,t)=-\frac{\hat{p}_r}{\mathbf{Z}_{a0}}e^{i(\omega t+k_0 x)}$$

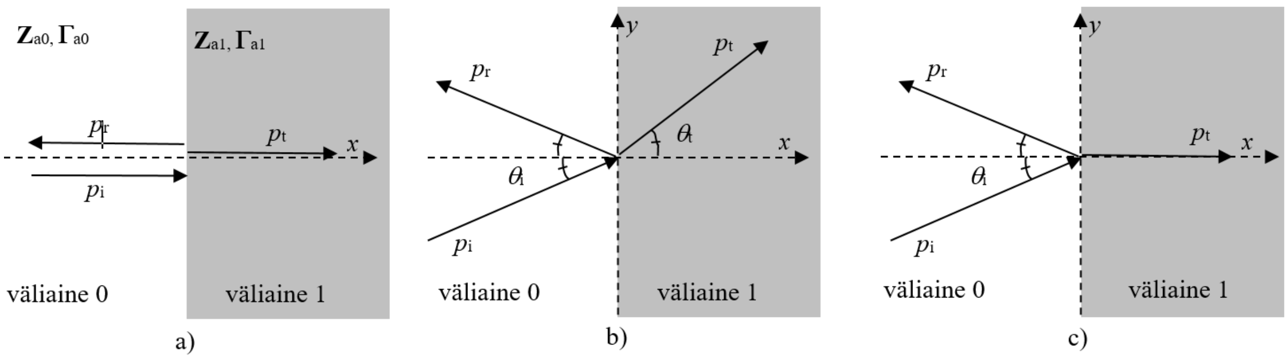


Fig. 3.3.1 Reflection and transmission at the interface of two materials. (a) Perpendicular angle of incidence, $\theta\tau$=0°. (b) Oblique angle of incidence and overall reacting material (transmission angle $\theta\tau$ depends on the speed of sound in medium 1). (c) Oblique angle of incidence and locally reactive material (always 0°).

The sound pressure and particle velocity of a transmitted sound wave are:

(3.3.5) $$\mathbf{p}_t(x,t)=\hat{p}_t e^{i(\omega t-k_1 x)}$$

(3.3.6) $$\mathbf{u}_t(x,t)=\frac{\hat{p}_t}{\mathbf{Z}_{a1}}e^{i(\omega t-k_1 x)}$$

The interface must meet two boundary conditions at all times:

(3.3.7) $$\mathbf{p}_i(x=0,t)+\mathbf{p}_r(x=0,t)=\mathbf{p}_t(x=0,t)$$

(3.3.8) $$\mathbf{u}_i(x=0,t)+\mathbf{u}_r(x=0,t)=\mathbf{u}_t(x=0,t)$$

By placing the above sound pressure equations in Eq. 3.3.7, we obtain

(3.3.9) $$\hat{p}_i+\hat{p}_r=\hat{p}_t$$

Similarly, for particle velocities, the Eq. 3.3.8 gives

(3.3.10) $$\frac{\hat{p}_i}{\mathbf{Z}_{a0}}-\frac{\hat{p}_r}{\mathbf{Z}_{a0}}=\frac{\hat{p}_t}{\mathbf{Z}_{a1}}$$

By elimination, $\hat{p}_t$ we obtain the pressure reflection coefficient R

(3.3.11) $$\boxed{\frac{\hat{p}_r}{\hat{p}_i}=\frac{\mathbf{Z}_{a1}-\mathbf{Z}_{a0}}{\mathbf{Z}_{a1}+\mathbf{Z}_{a0}}=\mathbf{R}}$$

The sound absorption ratio refers to the proportion of sound energy that falls on material 1 is absorbed by Wi:

(3.3.12) $$\alpha=\frac{W_t}{W_i}=\frac{W_i-W_r}{W_i}=1-\frac{W_r}{W_i}$$

where W_t is permeable to the interface and W_r is the sound energy reflected from the interface. Since the energy of sound depends on the intensity of sound, which in turn is proportional to the time mean value of the square of the sound pressure (number 1), the absorption ratio is obtained

(3.3.13) $$\alpha=1-\frac{I_{x,r}}{I_{x,i}}=1-\left(\frac{\hat{p}_r}{\hat{p}_i}\right)^2=1-|\mathbf{R}|^2=1-\left|\frac{\mathbf{Z}_{a1}-\mathbf{Z}_{a0}}{\mathbf{Z}_{a1}+\mathbf{Z}_{a0}}\right|^2$$

The absorption ratio is a real number between 0 and 1. If Z_{a0} and Z_{a1} differ a lot, the absorption ratio is low and the reflection is strong. A good absorption ratio is achieved when the impedance values are close to each other, allowing sound to pass through the interface.

The most commonly considered is the case when material 0 is air. Its characteristic impedance real number $Z_0=\rho_0\cdot c_0$ and we obtain

(3.3.14) $$\alpha=1-|\mathbf{R}|^2=1-\left|\frac{\mathbf{Z}_{a1}-Z_0}{\mathbf{Z}_{a1}+Z_0}\right|^2$$

If the characteristic impedance of material 1 is normalized with respect to the characteristic impedance of air

(3.3.15) $$\mathbf{z}_{a1}=\frac{\mathbf{Z}_{a1}}{Z_0}$$

Eq. 3.3.14 is reduced to:

(3.3.16) $$\alpha=1-\left|\frac{\mathbf{z}_{a1}-1}{\mathbf{z}_{a1}+1}\right|^2$$

The preceding equations require the use of complex number formulas. Eq. 3.3.14 can be reduced to a form where complex number calculation is not required:

(3.3.17) $$\boxed{\alpha=1-|\mathbf{R}|^2=\frac{4Z'_{a1}Z_0}{(Z'_{a1}+Z_0)^2+Z''^2_{a1}}}$$

It should be noted that the equation was derived in a situation where material 1 was assumed to be infinitely thick. In this case, the characteristic impedance of material 1 could be used in the equations as such. In reality, situations when material 1 has a finite thickness are of interest. In this case, Z_{a1} is replaced in the previous absorption ratio equations by the surface impedance Z_1 of the material. Calculation methods for such situations are presented in the following chapters.

(II) OBLIQUE ANGLE OF INCIDENCE

With an oblique angle of incidence of sound, a distinction must be made between materials that react locally and as a whole, because the sound that penetrates the interface is refracted in different ways.

Suppose a situation as shown in Fig. 3.3.1b where sound hits the interface of materials 0 and 1 at angle θ_i. Material 1 is *bulk reacting,* if the sound continues to travel inside the material at angle θ_t, which is determined by *Snell's* law of refraction:

(3.3.18) $$\frac{c_0}{\sin\theta_i} = \frac{c_1}{\sin\theta_t}$$

In this case, the reflection coefficient is given by:

(3.3.19) $$\mathbf{R}(\theta_i) = \frac{\mathbf{Z}_{a1}\cos\theta_i - \mathbf{Z}_{a0}\cos\theta_t}{\mathbf{Z}_{a1}\cos\theta_i + \mathbf{Z}_{a0}\cos\theta_t}$$

In locally reacting absorption materials, surface particulate oscillation is inhibited behind the interface (Fig. 3.3.1c). If material 1 is locally reacting, i.e., θ_t=0 ° regardless of the angle of incidence of sound, the reflection coefficient is given in the form

(3.3.20) $$\mathbf{R}(\theta_i) = \frac{\mathbf{Z}_{a1}\cos\theta_i - \mathbf{Z}_{a0}}{\mathbf{Z}_{a1}\cos\theta_i + \mathbf{Z}_{a0}}$$

In both cases, the absorption ratio at the angle of incidence θ_i is obtained from

(3.3.21) $$\alpha_{\theta_i} = 1 - |\mathbf{R}|^2 = 1 - \left|\frac{\mathbf{Z}_{a1} - \mathbf{Z}_{a0}/\cos\theta_i}{\mathbf{Z}_{a1} + \mathbf{Z}_{a0}/\cos\theta_i}\right|$$

Locally responsive means that the particle velocity at the interface depends only on the local sound pressure. As a whole, for reactive materials, the particle velocity at the interface depends not only on the local sound pressure, but also on the distribution of sound pressure inside the absorption material.

3.4. Porosity

Porosity is a characteristic of the absorption efficiency of materials. It is defined as the ratio of the volume V' [m^3] of air contained in the material and the total volume V [m^3] of the material:

(3.4.1) $$h = \frac{V'}{V}$$

Typically, porosity is above 0.90 for mineral wools. The porosity of a fiber-based material can be calculated from the equation

(3.4.2) $$h = 1 - \frac{\rho_m}{\rho_f}$$

where ρ_f is the density of the fibrous material (e.g., glass, stone, or wood) and ρ_m is the total density of the material.

The absorption ratio increases with increasing porosity if air can easily flow through the material. If the pores are connected, sound can easily travel inside the material and sound absorption based on friction losses is possible. When the pores are closed, even with high porosity, a high absorption ratio cannot be achieved. The particle velocity, u [m/s], of sound inside the material is proportional to porosity h and particle velocity u' in air

(3.4.3) $$u \cong s\frac{u'}{h}$$

where s [] is turtuosity. The structural factor tells you how the pores are connected to each other.

3.5 Characteristic properties of porous materials

The characteristic impedance of a material can be measured by the so-called impedance of a material. impedance tube or standing wave tube. If the specific flow resistivity of the material is known, the characteristic impedance can also be modelled fairly reliably.

Delany & Bazley (1969) presented a regression model based on empirical research to calculate the characteristic impedance $\mathbf{Z_a}$ and propagation constant $\mathbf{\Gamma}_a$ of a fibrous material when the specific flow resistivity r [Pa·s/m^2] of the material is known (Cox &; D'Antonio, 2004):

(3.5.1) $$\mathbf{\Gamma}_a = \Gamma'_a + j\Gamma''_a = \frac{\omega}{c_0}\left[1 + 0.0978E^{-0.700} - j0.189E^{-0.595}\right]$$

(3.5.2) $$\mathbf{Z}_a = Z'_a + jZ''_a = |\mathbf{Z}_a|e^{j\varphi} = \rho_0 c_0\left[1 + 0.0571E^{-0.754} - j0.087E^{-0.732}\right]$$

where φ is the phase angle and

(3.5.3) $$E = \frac{\rho_0 f}{r}$$

where ρ_0 [kg/m^3] is the density of air and f [Hz] is the frequency of sound. There are 8 coefficients in the equation, which are well suited for calculating the properties of mineral wool. Coefficients that provide good calculation accuracy have also been developed for other materials.

3.6. Calculation of the absorption ratio on a porous panel

The absorption ratio of a porous board depends on the characteristic properties of the sheet material, the thickness of the sheet and the position of the panel relative to other reflective or non-reflective interfaces. The main three cases are shown in Fig. 3.6.1. Models for calculating the absorption ratio of the normal angle of incidence are presented below.

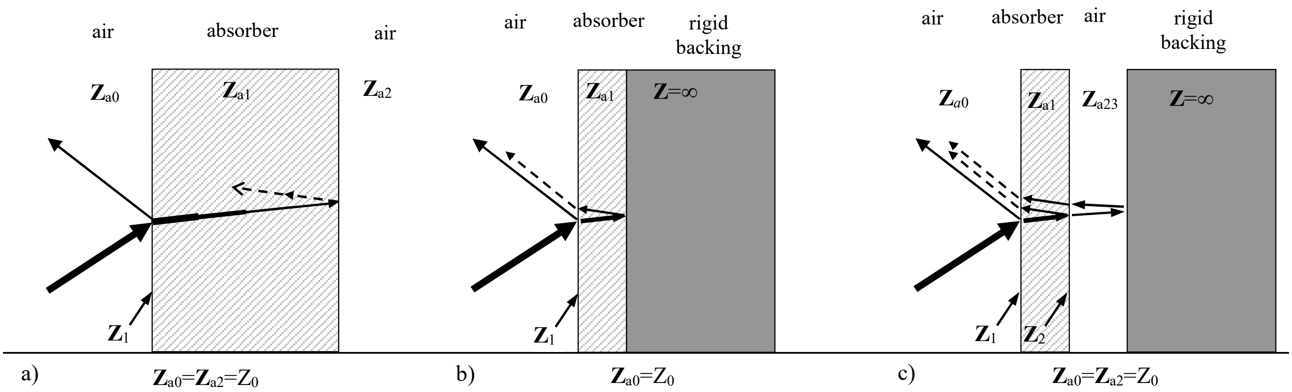


Fig. 3.6.1. Three methods of installation of porous board. The porous board is marked with light gray and the dark gray material is a high-impedance, fully reflective backwall.

(A) THICK POROUS PANEL SURROUNDED BY AIR

Consider a thick porous panel (Fig. 3.6.1a) with a characteristic impedance $\mathbf{Z_{a1}}$. The panel is surrounded by air. The panel can be viewed as "acoustically thick" in the frequency range where

(3.6.1) $$\Gamma'_{a1} d_1 = \frac{2\pi d_1}{\lambda_{a1}} > 2$$

where d_1 [m] is the thickness of the panel and λ_{a1} [m] is the wavelength inside the material. "Acoustically thick panel" means that the sound is attenuated as it travels inside the material to such an extent that the reflection from its rear surface no longer has any effect on the sound pressures perceived on the front surface. In this case, the surface impedance of the panel is equal to the characteristic impedance of the material, and the absorption ratio is determined by the equation

$$\alpha = 1 - |\mathbf{R}|^2 = \frac{4Z'_{a1} Z_0}{(Z'_{a1} + Z_0)^2 + Z''^2_{a1}} \tag{3.6.2}$$

(B) THIN POROUS PANEL AGAINST REFLECTIVE BACKGROUND

Consider the most common panel mounting method, in which the panel is placed on top of a reflective background surface (Fig. 3.6.1b). The impedance of the background surface is assumed to be fully reflective. The reflection affects the surface impedance Z_1 of the panel**,** especially if the panel is acoustically thin and the amplitude of sound reflected from the back surface is still significant when it returns to the air-panel interface. The surface impedance [N·s/m3] depends on the characteristic characteristics of the panel ($\mathbf{Z}_{a1}$, Γ_{a1}) and the panel thickness d_1 [m] as follows:

$$\mathbf{Z}_1 = -j\mathbf{Z}_{a1} \cot\left(\mathbf{\Gamma}_{a1} d_1\right) = \mathbf{Z}_{a1} \coth\left(j\mathbf{\Gamma}_{a1} d_1\right) \tag{3.6.3}$$

The hyperbolic cotangent of a complex number can be calculated by the equation

$$\coth \mathbf{\Gamma} = \frac{e^{\mathbf{\Gamma}} + e^{-\mathbf{\Gamma}}}{e^{\mathbf{\Gamma}} - e^{-\mathbf{\Gamma}}} = \frac{\cos \Gamma''\left(e^{\Gamma'} + e^{-\Gamma'}\right) + j \sin \Gamma''\left(e^{\Gamma'} - e^{-\Gamma'}\right)}{\cos \Gamma''\left(e^{\Gamma'} - e^{-\Gamma'}\right) + j \sin \Gamma''\left(e^{\Gamma'} + e^{-\Gamma'}\right)} \tag{3.6.4}$$

where the complex representation of the wavenumber is $\mathbf{\Gamma}=\Gamma'+j'\Gamma'$. The absorption ratio is calculated by the equation

$$\boxed{\alpha = 1 - |\mathbf{R}|^2 = \frac{4Z'_1 Z_0}{(Z'_1 + Z_0)^2 + Z''^2_1}} \tag{3.6.5}$$

Coth(x) approaches infinity when x approaches zero and Coth(x) approaches one when x approaches infinity. Thus, the thickness of the panel significantly affects the absorption ratio and its frequency behavior. When the panel is thin relative to the wavelength ($d_1 << \lambda_{a1}/4$), the Coth term has a high magnitude, $\mathbf{Z_1} >> \mathbf{Z_a}$**,** and the absorption ratio remains low. Because of this, for example, a thin panel does not absorb sound well as at high frequencies. When $d_1 > \lambda_{a1}/\pi$ and , moreover, Γ_{a1} is not small, the Coth term approaches one, $\mathbf{Z_1} = \mathbf{Z_a}$ and the absorption ratio behaves as in case (a). In case (b), the same absorption ratio can be achieved at panel thicknesses that are large in relation to the wavelength as if there were no reflective background surface.

Fig. 3.6.2 shows how a plane wave behaves when it hits a perpendicular interface. The sound pressure reaches a maximum on a wall surface where the boundary condition for particle velocity is zero (reactive sound field). A reflection occurs from the surface and a standing wave is formed so that the particle velocity reaches a maximum value at $\lambda/4$ distance from the surface. The exact distance depends slightly on the surface impedance and phase shift. Absorption is most effective when the particle velocity is high. At a normal angle of incidence, complete absorption can only be achieved on the panel if the panel or part of it is located at the maximum point of particle velocity.

Fig. 3.6.3 shows the effect of panel thickness on the absorption ratio. As the thickness increases, the absorption ratio improves at low frequencies.

In the room, sound comes to the surface from all directions. Due to this, almost complete absorption can be achieved already when the thickness of the panel is at least one tenth of the wavelength.

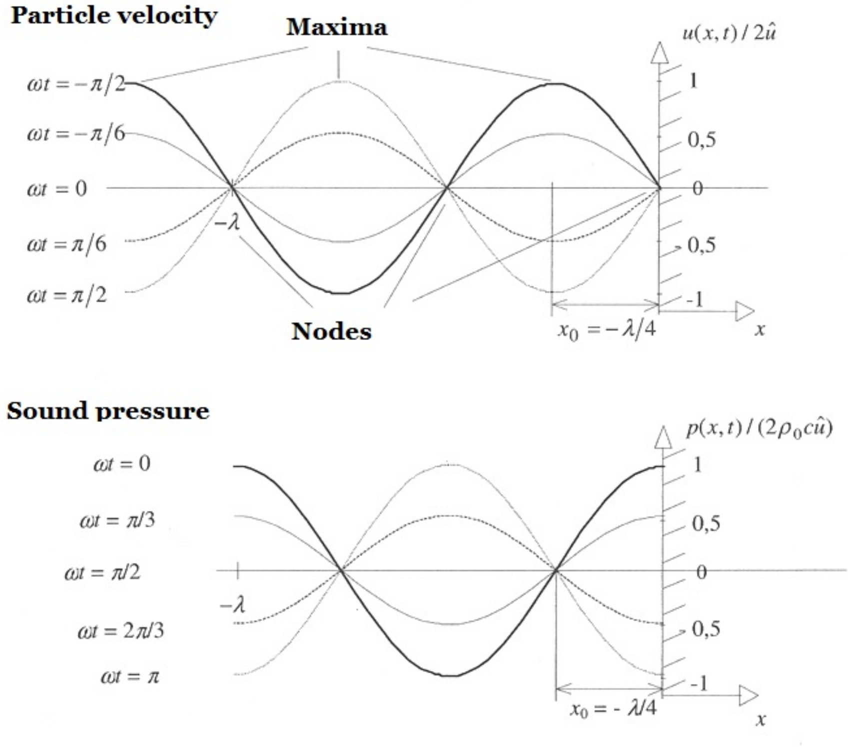


Fig. 3.6.2 Behaviour of particle velocity and sound pressure in a standing wave produced when sound encounters a reflecting surface (right). The porous panel must be located at the dome of the particle velocity in order to achieve maximum absorption. (Photo: Boden et al. 1999)

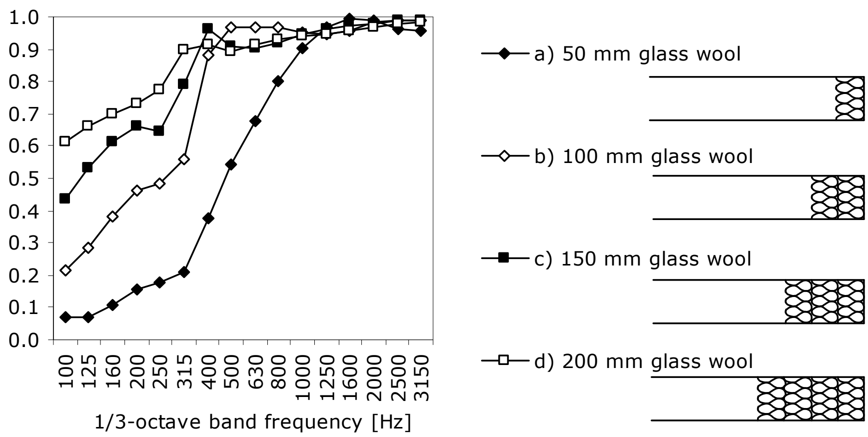


Fig. 3.6.3 Mineral wool (d=50 mm, ρ=18 kg/m^3, r=9600 Pa·s/m^2) dependence of the absorption ratio of the normal angle of incidence on thickness. (Photo: Oliva et al., 2010)

(C) THIN POROUS PANEL, AIR LAYER AND REFLECTIVE BACKING

Consider the situation shown in Fig. 3.6.1c where the porous panel (1) is detached from the hard backing surface, leaving an inflatable cavity behind the panel (2). This structure is realised, for example, in suspended ceilings, where building services are placed in the space between the porous board and the vault. The surface impedance of the cavity on the back surface of the porous panel is given by:

(3.6.6) $$\mathbf{Z}_2 = -jZ_0 \,\mathrm{co}(k_0 t)$$

where k_0 is the wavenumber in the air and t [m] is the thickness of the cavity. The reflection coefficient from the back surface of the porous panel is

(3.6.7) $$\mathbf{R} = \frac{\mathbf{Z}_2 - \mathbf{Z}_{a,1}}{\mathbf{Z}_2 + \mathbf{Z}_{a,1}}$$

The surface impedance of the front surface of the porous material layer is given by:

(3.6.8) $$\boldsymbol{Z}_1 = \boldsymbol{Z}_{a1} \frac{\boldsymbol{Z}_2 \cosh \boldsymbol{\Gamma}_{a1} d + \boldsymbol{Z}_{a1} \sinh \boldsymbol{\Gamma}_{a1} d}{\boldsymbol{Z}_{a1} \cosh \boldsymbol{\Gamma}_{a1} d + \boldsymbol{Z}_2 \sinh \boldsymbol{\Gamma}_{a1} d}$$

The absorption ratio is calculated by Eq. 3.3.17.

Fig. 3.6.4 shows the effect of the thickness of the cavity behind the panel on the absorption ratio. Increasing the thickness of the cavity significantly improves the sound absorption of low frequencies, but at higher frequencies deteriorations are observed due to standing waves generated in the cavity.

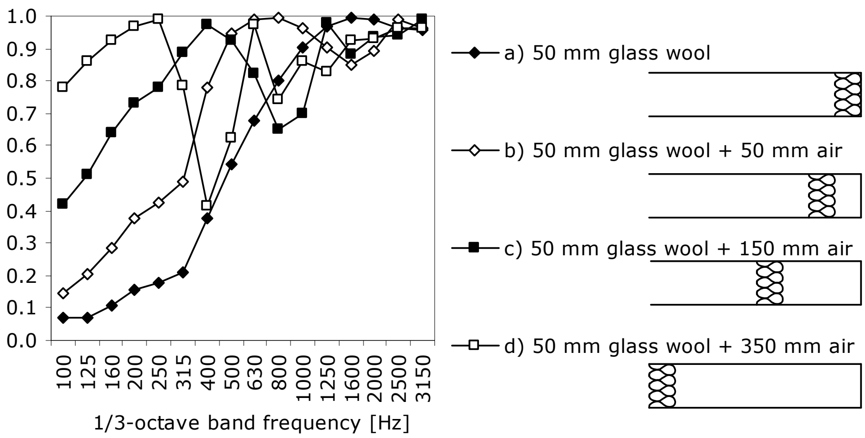


Fig. 3.6.4 Effect of cavity thickness t (0, 50, 150, 350 mm) mineral wool removed from the background (d=50 mm, ρ=18 kg/m^3, r=9600 Pa·s/m^2) to the absorption ratio of the normal angle of incidence (Source: Oliva et al. 2010).

3.7. Perforated panel resonator

The types of resonators are Helmholz resonator and panel resonator. The perforated panel resonator is the most common application of the Helmholz resonator. It refers to a layered structure as shown in Fig. 3.7.1, where the panel is perforated in the form of a regular lattice and the panel is separated by an airtight cavitation from the reflective rigid background surface. The air mass in the hole in the panel forms the cavity of the Helmholz resonator (mass-spring system) with the air spring. Thus, the perforated panel resonator is the lattice of Helmholz resonators. The surface mass of the perforated sheet is irrelevant in the case of a perforated sheet. If, on the other hand, there are no holes in the panel, the structure forms a panel resonator, i.e. the mass of the panel and the air spring of the cavity resonate.

Usually the holes are round, square or oblong slits. In commercial products, a porous layer is glued to the back surface of the perforated panel, which increases the resistive losses of the resonator. The cavity may also contain absorption material elsewhere than on the back surface of the panel. The absorption ratio of perforated panels depends on factors such as cavity thickness, hole size, hole shape, hole ratio, cavity filling method and specific flow resistivity of filling. The modeling result is extremely sensitive to the choices of the output variable. The calculation should be done at a high frequency resolution using, for example, 1/9 or 1/27 octave frequencies. The value of the tersis band is given as the average of the frequencies belonging to the tersis band.

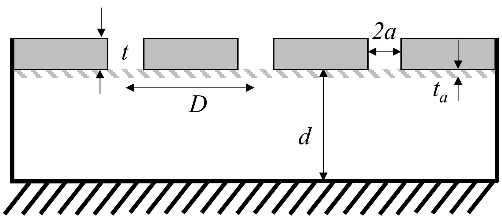


Fig. 3.7.1 Design and marking of perforated panel resonator, if the holes are round.

The surface impedance of a resonator (perforated panel or panel resonator) is

(3.7.1) $$Z_1 = Z_1' + jZ_1''$$

The absorption ratio is calculated by Eq. 3.3.17 after solving the surface impedance (normal angle of incidence).

Consider the perforated panel resonator in Fig. 3.7.1. The spacing between the holes is large relative to the diameter of the holes. In addition, the thickness of the perforated sheet and the radius of the hole are clearly smaller than the wavelength. In this case, the resistance consists of the sum of the resistance of the air holes and the resistance of any porous layer placed behind the perforated panel. The hole ratio ε [] (porosity) of the perforated panel is the proportion of the perforated area of the perforated panel:

(3.7.2) $$\varepsilon = \frac{\pi a^2}{D^2}$$

where a [m] is the radius of the circular hole and D [m] is the distance between the holes. In a round hole (diameter 2a), the resistance of oscillating air is

(3.7.3) $$Z'_{air} = \frac{\rho_0}{\varepsilon}\sqrt{8\nu\omega}\left(1+\frac{t}{2a}\right)$$

where $\nu = 15\cdot 10^{-6}$ m^2/s is the kinetic viscosity, ω [Hz] is the angular frequency, and t [m] is the thickness of the perforated panel. The equation works if the hole diameter is greater than 1 mm (not a so-called micro-perforated panel).

If the porous panel (felt) is placed in the cavity at least at a distance of the hole diameter from the perforated panel, the resistance of the porous panel will be

(3.7.4) $$Z'_{felt} = rt_a$$

where r [Pas/m2] is the specific flow resistivity of the panel and t_a [m] is the thickness of the panel. If the porous panel is placed immediately behind the perforated sheet, which is realized in almost all commercial perforated sheet products, the flow rate in the porous panel is 1/ε fold and the resistance is

(3.7.5) $$Z'_{felt} = \frac{rt_a}{\varepsilon}$$

The total resistance to be placed in equation **3.7.1** is the sum of the air resistance of the felt and the holes:

(3.7.6) $$Z'_1 = Z'_{felt} + Z'_{air}$$

The reactance of the perforated panel is of the form

(3.7.7) $$Z_1'' = \omega m' - \rho_0 c_0 \cot(kd)$$

where m' [kg/m^2] is the surface mass of the air oscillating in the hole

(3.7.8) $$m' = \frac{\rho_0 t'}{\varepsilon}$$

and t' [m] is the effective height (head-corrected height) of the air mass oscillating in the hole of the perforated panel:

(3.7.9) $$t' = t + 2\delta a + \sqrt{\frac{8\nu}{\omega}\left(1+\frac{t}{2a}\right)}$$

The square root term is relevant only for micro-perforated panels, where the hole diameter is less than 1 mm and viscous losses in the hole begin to be significant. End correction δ [-] has the shape of a round hole

(3.7.10) $$\delta = 0.8\left(1-1.4\sqrt{\varepsilon}\right)$$

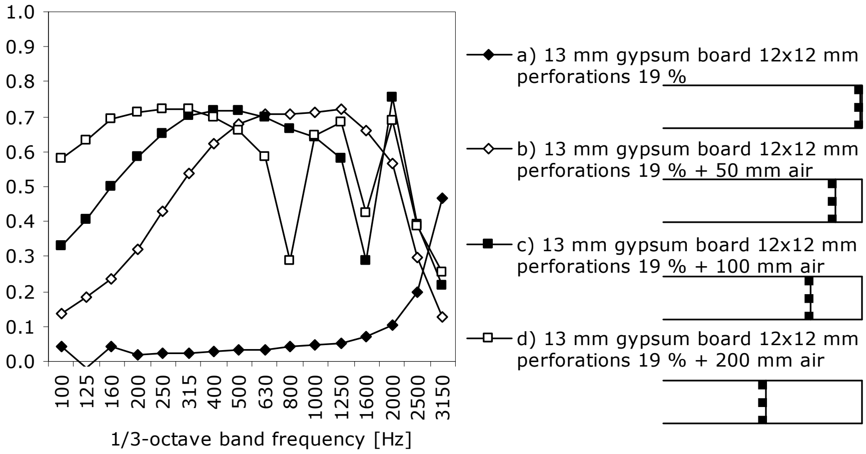
The resonant frequency of the perforated panel resonator f_0 is given by:

(3.7.11) $$f_0 = \frac{c_0}{2\pi}\sqrt{\frac{\varepsilon}{t'd}}$$

where t' is determined from Eq. 3.7.9 ignoring the square root term. The resonant frequency comes out most strongly when there is no felt or its resistivity is low.

Fig. **3.7.2a** shows the absorption ratios of one piece of perforated drywall with different cavity thicknesses. If the perforated panel is placed attached to the background surface, absorption will be minimal, since no resonator is generated. The maximum of the absorption ratio shifts towards lower frequencies as the thickness of the cavity increases. At the same time, stagnant waves in the cavity cause absorption minimas at high frequencies. If the cavity is completely filled with porous material, the absorption minima caused by standing waves are removed from the absorption curve, but the overall condition does not significantly improve (Fig. **3.7.2b**).

Thanks to felt, the absorption of perforated panel resonators is fairly broadband, and there are no significant resonant absorption peaks. Without felt, the absorption of the perforated panel alone drops to almost zero (Oliva et al. 2010).



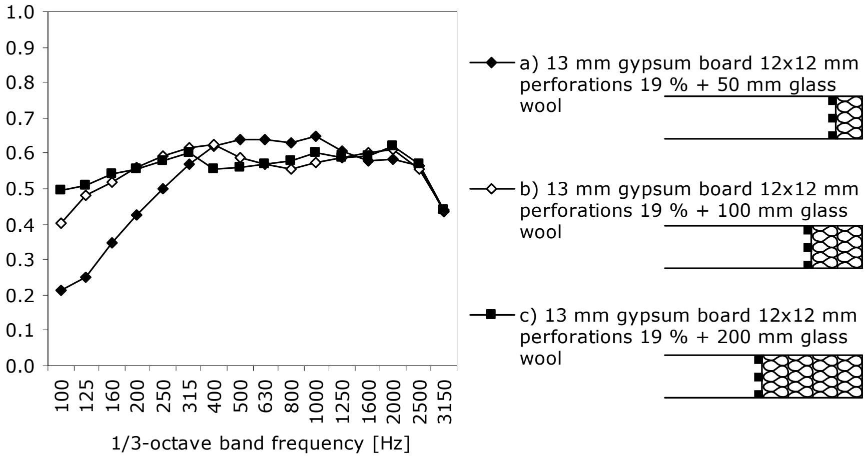


Fig. 3.7.2 Dependence of the absorption ratio of the normal angle of incidence of perforated drywall 13 mm thick on the thickness of the cavity (0, 50, 100 and 200 mm) when (a) the cavity is empty or (b) the cavity is filled with mineral wool (ρ=18 kg/m^3, r=9600 Pa·s/m^2). In both pictures, the back surface of the drywall had a felt corresponding to the product (R_s=1183 Pas/m*)*. The hole ratio is ε=0.19.

3.8 Thin perforated sheets

Porous materials are in some situations covered with a thin, 0.5-1.0 mm thick, perforated metal panel to improve appearance or wear resistance, without the panel being intended to provide a Helmholz panel resonator. The absorption ratio of the resulting layer structure can be modelled using the equations in Sec. 3.7.

Fig. 3.8.1 shows the effect of the hole ratio of a thin perforated sheet coating on the absorption ratio of the underlying 50 mm thick mineral wool. The high hole ratio has no effect on the absorption capacity of the mineral wool, as the perforated panel is acoustically transparent. With a small hole ratio, the absorption capacity of high frequencies is significantly reduced, and the properties of the Helmholz resonator come to the fore.

If there is no porous material behind the perforated panel, absorption occurs only at the Helmholz resonant frequency (Fig. 3.8.2).

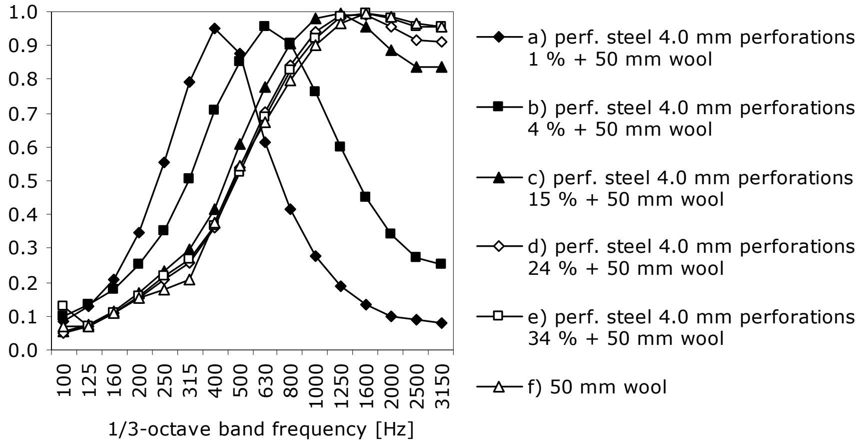


Fig. 3.8.1 Effect of hole ratio of sheet metal (thickness 0.9 mm) mineral wool behind it (d=50 mm, ρ=18 kg/m^3, r=Pa·s/m^2) to the absorption ratio of the normal angle of incidence. Wool is mounted against a rigid background. The hole ratios ε are 0.01, 0.04, 0.15, 0.24, 0.35, and 1.00. (Source: Oliva et al., 2010)

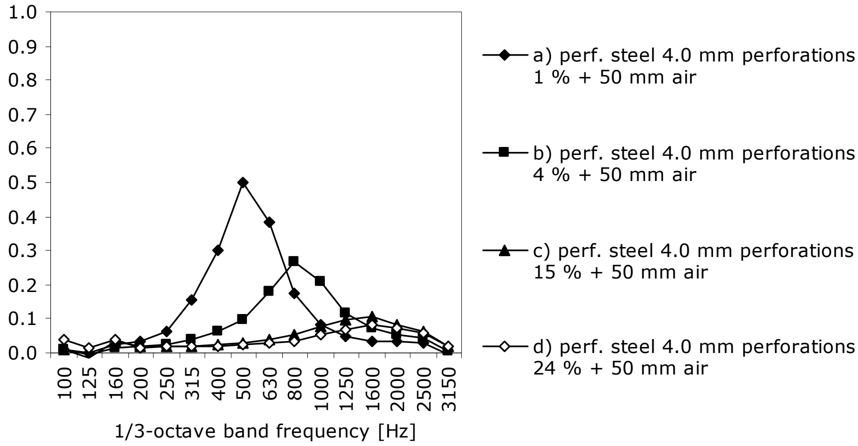


Fig. 3.8.2 Effect of hole ratio of perforated damper (thickness 0.9 mm) on normal angle of incidence absorption ratio with 50 mm inflatable cavity behind the damper. The hole ratios ε are 0.01, 0.04, 0.15, and 0.24. (Source: Oliva et al., 2010).

3.9 Micro-perforated sheets

It is possible to calculate the absorption ratio of a microperforated panel as described in section 3.7, but the resistance caused by the hole should be estimated using equations more accurate than equation 3.7.3 , which are available in the literature. In a micro-perforated panel, the diameter of the holes is less than 1 mm. Due to the viscosity of the air, the particle velocity in the centre of the hole is always higher than at the sides of the hole, but when the hole is very low, the viscosity reduces the particle velocity over almost the entire hole area (Fig. 3.9.1). The viscosity of the air causes a significant loss of friction in the hole, which means an increase in resistance. Otherwise, the principle of operation of the microperforated panel is the same as in the Helmholz resonator, that is, the air mass in the hole resonates with the air spring of the cavity. Due to microperforation, the resistive impedance is already high, eliminating the need for a separate porous layer of absorption material on the back of the panel. Due to this, the microperforated sheet can be made transparent. A microperforated panel can provide a moderately good absorption ratio in the frequency range of 50 to 1000 Hz. A single hole size-cavity sizing can provide a good absorption ratio for a frequency band only about an octave wide. To achieve broadband absorption, several different dimensions are required. For this reason, microperforated panels can never achieve high absorption ratios in relation to their surface area, but for transparency, microperforated panels can achieve good results in rooms where visible absorption surfaces are not desired or possible.

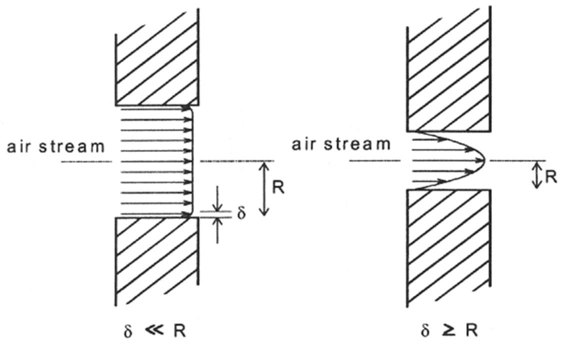


Fig. 3.9.1 Air flow in large and small holes (Source: Mechel and Ver, 1999).

3.10 Panel resonators

The panel resonator was already discussed indirectly in Sec. 3.7. The equations described here also work for a panel resonator, when the surface mass of the panel is placed in place of the surface mass *m*' and the air resistance of the holes is ignored, i.e. $Z'_{air} = 0$.

The panel resonator consists of a tight non-perforated panel, an airtight cavity at the back of the panel and a rigid reflective background on which the panel is built (Fig. 3.10.1).

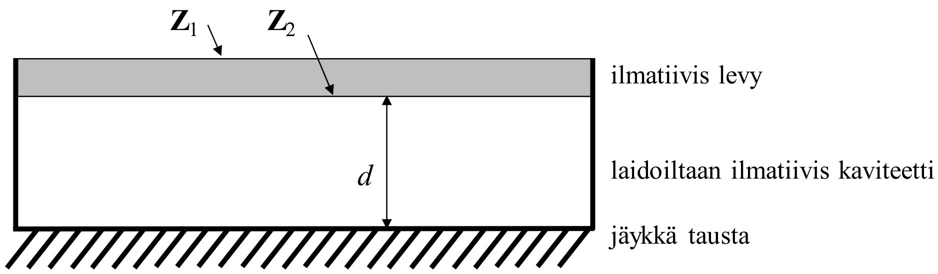


Fig. 3.10.1 Panel resonator structure. The sheet is assumed to be loose over its entire surface area. The cavity is usually inflatable or partially or fully filled with porous matter.

If the panel is loose and lossless, freely located in space, the characteristic impedance of the panel is purely imaginary and depends mainly on the surface mass *m*' [kg/m^2] of the panel at low frequencies

(3.10.1) $$\mathbf{Z}_{a1} = j\omega m'$$

The panel turns into a resonator when there is a layer of air behind it. Consider the frequency range in which the wavelength in the air is greater than the thickness of the air gap, so that no modes are formed in the air gap. In this case, the surface impedance of the air layer on the back surface of the panel is

(3.10.2) $$\mathbf{Z}_2 = -jZ_0 \,\mathrm{co}(k_0 d)$$

where *Z0* is the characteristic impedance of air and *d* [m] is the thickness of the air layer. If the cavity has a porous filling, the impedance formulas of Sec. 3.6b apply.

The surface impedance of the panel resonator on the front surface of the structure is given by:

(3.10.3) $$\mathbf{Z}_1 = \mathbf{Z}_2 + \mathbf{Z}_{a1}$$

The absorption ratio is calculated by equation **3.3.17.**

When the sum of the imaginary parts goes to zero, we are at the resonant frequency of the panel resonator, where the absorption reaches its maximum value:

(3.10.4) $$f_0 = \frac{62}{\sqrt{m'd}}$$

If there is a porous filling in the cavitation, then the resonant frequency will be

(3.10.5) $$f_0 = \frac{50}{\sqrt{m'd}}$$

The resonant frequency measured for a panel resonator usually differs from the computational one, since the panel is not loose but has a certain bending rigidity. In addition, the oscillating effective surface mass m'' is not the same as the physical surface mass *m'* of the panel, because the panel must be attached to the panels by screwing or gluing and only part of the panel can vibrate freely.

3.11 Modelling the multilayer structure

Multi-storey solutions are often used in buildings and vehicles to optimise the different surface property requirements. Requirements for the structure may be set by boundary conditions, such as vapor permeability, fire safety, thermal insulation, aesthetics, impact resistance, manufacturing costs and overall economy. Computational estimation of the absorption ratio requires knowledge of the characteristic impedance and propagation constant of each layer. A more general model for multilayer structures is presented in this chapter (Oliva et al., 2009). The model is suitable for perpendicular angles of incidence of sound.

It is assumed that the absorption material consists of an N layer. The chaining method involves recursively determining the surface impedances of the front surface of each layer of material (Fig. **3.11.1**). The sound arrives from material 0, which is assumed to be air, and is the first to arrive at the surface of material layer 1. The task of modeling is to recursively determine the surface impedance Z_1 of surface 1, from which then the absorption ratio can be determined as shown earlier. Concatenation is started from the back layer (i=N) when viewed from the direction of sound input. Modelling always begins by determining the surface impedance Z_{N+1} of the background layer N+1 of the absorption structure, which is usually a rigid wall or air.

The surface impedance of material layer i is determined recursively by the equation

(3.11.1) $$\mathbf{Z}_i = \frac{-j\mathbf{Z}_{i+1}\mathbf{Z}_{a,i}\cot(\mathbf{\Gamma}_{a,i}d_i) + \mathbf{Z}_i^2}{\mathbf{Z}_{i+1} - j\mathbf{Z}_{a,i}\cot(\mathbf{\Gamma}_{a,i}d_i)}$$

where $\mathbf{Z_{a,i}}$ [kg/m^{2}s] is the characteristic impedance of the material, $\mathbf{\Gamma}_{a,i}$ [m^{-1}] is the material propagation constant, d_i is the thickness of the material, and $\mathbf{Z_{i+1}}$ is the surface impedance of the layer behind the material (or a recursively determined combination of layers). Finally, the absorption ratio is calculated from the equation

(3.11.2) $$\alpha = 1 - |\mathbf{R}|^2 = 1 - \left|\frac{\mathbf{Z}_1 - Z_0}{\mathbf{Z}_1 + Z_0}\right|^2$$

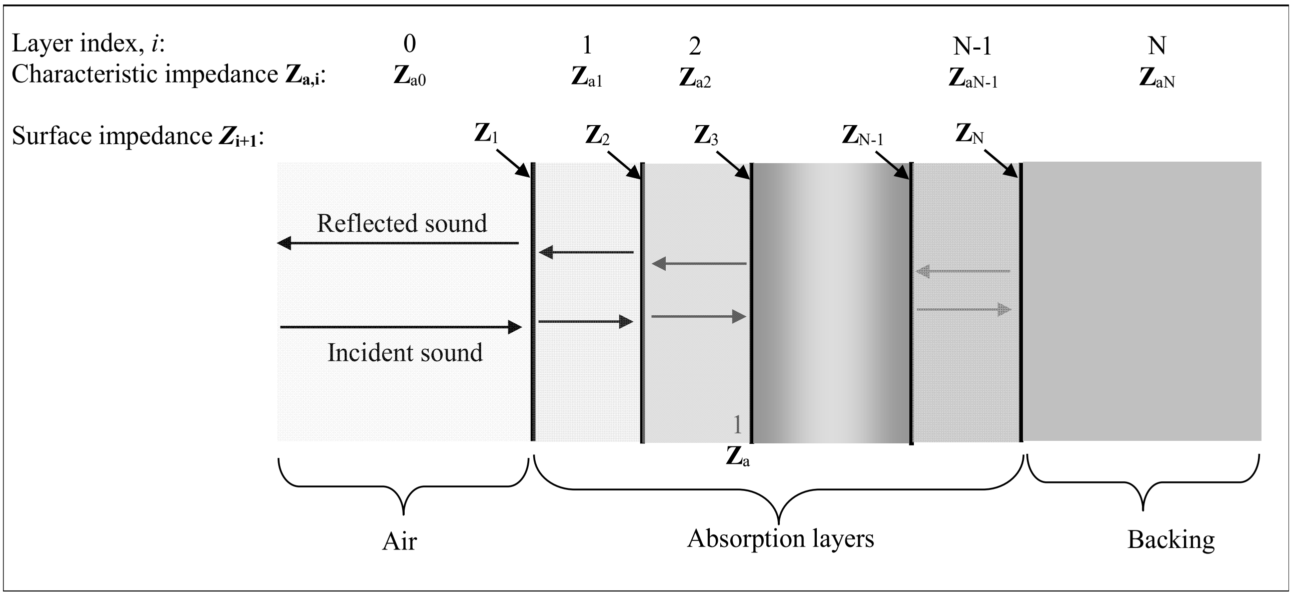


Fig. 3.11.1 Impedance modelling of multilayer absorption material. Modeling occurs from right to left. As a result, the surface impedance $\mathbf{Z}_1$ of the exterior surface is obtained, on the basis of which the absorption ratio of the entire structure is determined.

The background layer is usually a fully reflective wall (built against the wall), in which case both the real and imaginary parts of its surface impedance Z_{N+1} are marked as infinity. If the background layer is air (e.g. a screen or a hanging panel), its surface impedance is the same as the characteristic impedance of air, i.e., $Z_{N+1}=Z_0=443$ kg/m^{2}s, and the imaginary part is zero.

The simplest case is a single-layer structure (N=1) where Eq. 3.11.1 is used only once. In this case, the absorption material layer (i=1) is directly against the backing layer (i=2). An absorption structure with at least two absorption layers (N>2) is considered to be a multilayer structure.

In layers can be:

1. porous panel
2. air layer
3. perforated or non-perforated leak-proof panel

The following are the methods for determining the characteristic characteristics of these options.

(i) Porous material

The characteristic impedance of the porous panel is determined according to Ch. 3.5.

(ii) Air layer

The characteristic impedance of the air layer is a pure real number: $Z_a=Z_0=\rho_0 c_0=443$ [kg/m^{2}s]. The propagation constant is the pure imaginary number $\Gamma_a=jk_0$ [1/m].

(iii) Perforated or non-perforated leak-proof panel

The equations in Ch. 3.7 apply.

3.12. Statistical absorption coefficient

In the room, sound enters the sample from all directions, i.e. at an angle of 180 degrees. In this case, we are talking about diffuse angles of incidence. The diffuse angle of incidence absorption ratio is called the statistical absorption ratio αst. The product values of materials are always expressed as values of the statistical absorption ratio, because it gives a better picture of the behavior of the product in the room than the absorption ratio of a normal angle of incidence.

With *bulk reacting* material, the sound continues behind the interface at approximately the same angle of propagation as the sound hitting the surface. The statistical absorption ratio is determined by the Paris equation

(3.12.1) $$\alpha_{st} = 2\int_0^{\pi/2} \alpha(\theta)\cos(\theta)\sin(\theta)d\theta$$

where the angle-dependent absorption ratio $\alpha(\theta)$ is obtained, for example, from experimental observations. If the material is *locally reacting*, the sound propagates perpendicular to the surface after the interface. This happens with perforated panels, since each hole serves as a point source. The statistical absorption ratio is given by the equation (ISO 10534)

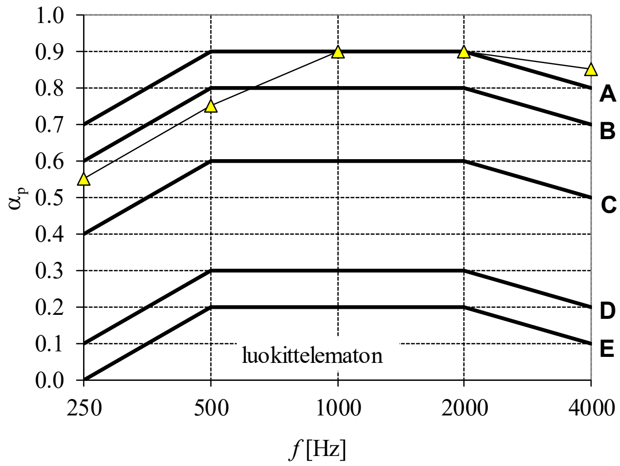
(3.12.2) $$\alpha_{st} = 8\frac{z'}{z'^2+z''^2}\left[1-\frac{z'}{z'^2+z''^2}\ln\left(1+2z'+z'^2+z''^2\right)+\frac{1}{z''}\frac{z'^2-z''^2}{z'^2+z''^2}\arctan\left(\frac{z''}{1+z'}\right)\right]$$

where z is the normalized surface impedance of the material at perpendicular angles θ=0º divided by the characteristic impedance of air

(3.12.3) $$z = z' + jz'' = \frac{Z}{\rho_0 c_0}$$

3.13. Classification of absorption materials

ISO 11654 presents a method for classifying absorption materials into classes A–E according to their absorption ratio. The method is widely used because it makes it easier to present product selection criteria, for example in building design, where frequency-dependent absorption ratio requirements are difficult to use. The classification is suitable for materials placed in normal rooms when speech is the main source of sound. The classification is based on octave bands from 250– to 4000 Hz. Octave values of the absorption ratio are first rounded to 0.05 (practical absorption coefficient, $\alpha\pi$). These are compared with the reference curves A E in Fig. 3.13.1. Class X can be achieved if the measured octave band values are less than the octave band values of the Class X reference curve by no more than 0.10. There is no compensation for exceeding the reference curve.



f [Hz]	A	B	C	D	E
250	0.7	0.6	0.4	0.1	0
500	0.9	0.8	0.6	0.3	0.2
1000	0.9	0.8	0.6	0.3	0.2
2000	0.9	0.8	0.6	0.3	0.2
4000	0.8	0.7	0.5	0.2	0.1

Fig. 3.13.1 Reference curves according to ISO 11654 for absorption classes A–E. The example result(s) gives class B.

3.14. Absorption coefficients

Fig. 3.14.1 and **Table 3.14.1** show some statistical absorption coefficients (diffuse angle of incidence).

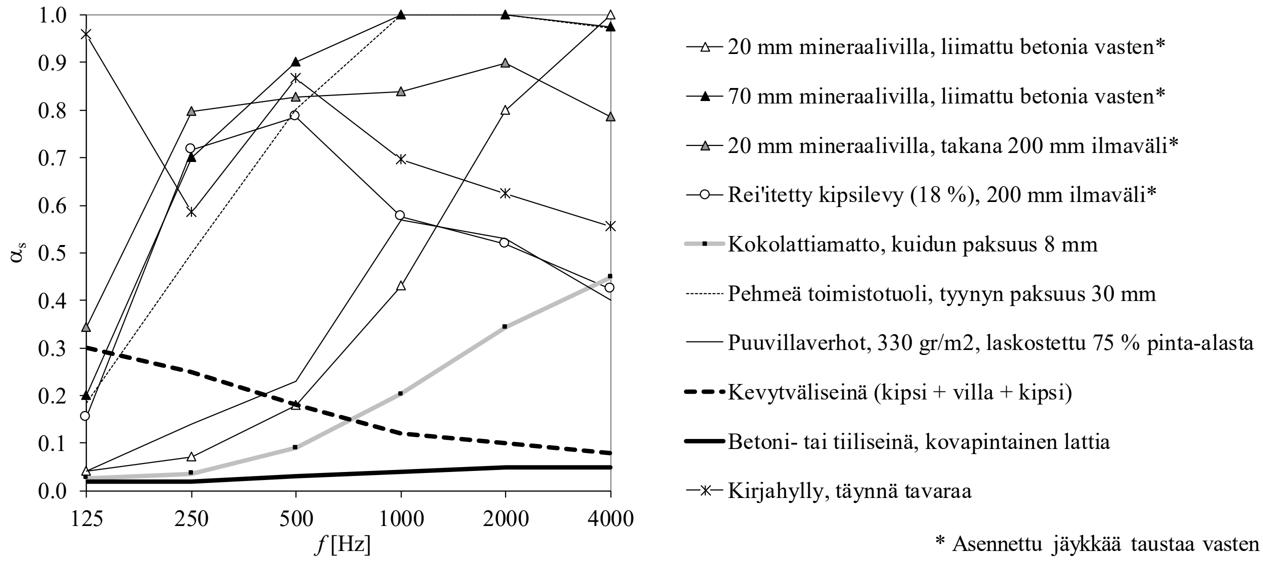


Fig. 3.14.1. Statistical absorption ratios ασ by octave frequency band.

Table 3.14.1. Statistical absorption ratios α_s in octave bands (Halme &; Seppänen, 2002).

	Taajuus [Hz]					
Rakenne	**125**	**250**	**500**	**1000**	**2000**	**4000**
tiiliseinä puhtaaksi muurattu	0.02	0.02	0.03	0.04	0.05	0.05
tiiliseinä kalkkilaastilla rapattuna	0.02	0.02	0.03	0.04	0.05	0.05
betoni, puhdasvalupinta	0.01	0.01	0.02	0.02	0.02	0.03
marmori hiottu	0.01	0.01	0.01	0.01	0.02	0.02
rapattu ja öljymaalattu tiiliseinä	0.01	0.02	0.02	0.02	0.03	0.03
rapattu ja tapetoitu tiiliseinä	0.02	0.03	0.04	0.04	0.07	0.08
kevytbetoni (solubetoni) maalaamaton	0.02	0.02	0.03	0.03	0.04	0.04
kevytbetoni (solubetoni) maalattu	0.01	0.02	0.02	0.02	0.03	0.03
kevytsorabetoni 200 mm maalaamaton	0.38	0.46	0.44	0.55	0.63	0.55
kevytsorabetonitiili 70 mm	0.05	0.20	0.66	0.38	0.52	0.47
seinälaatoitus	0.01	0.01	0.01	0.01	0.02	0.02
linoleumipäällyste suoraan betonin päälle	0.02	0.02	0.03	0.03	0.04	0.04
puulattia liimattu bitumilla suoraan betoniin	0.04	0.04	0.07	0.06	0.06	0.07
puulattia korokkeilla	0.15	0.11	0.10	0.07	0.06	0.07
5 mm:n kumimatto suoraan betonin päällä	0.04	0.04	0.08	0.12	0.13	0.10
huopapohjainen muovimatto	0.03	0.05	0.09	0.10	0.08	0.06
villanukkamatto tai vastaava tekokuitumatto + alusmatto, kokonaispaksuus 10 mm betonin päällä	0.09	0.08	0.21	0.26	0.27	0.37
mosaiikki	0.01	0.01	0.01	0.01	0.02	0.02
ikkuna kaksinkertainen	0.40	0.30	0.20	0.17	0.15	0.10
puurakenteinen laakaovi	0.14	0.10	0.08	0.07	0.06	0.05
ilmanvaihtosäleikkö puoliksi avoinna	0.30	0.40	0.50	0.50	0.50	0.50
vedenpinta	0.01	0.01	0.01	0.02	0.02	0.03
maan pinta yleensä	0.15	0.25	0.40	0.55	0.60	0.60
sora	0.25	0.60	0.65	0.70	0.75	0.80
lumi 100 mm	0.45	0.75	0.90	0.95	0.95	0.95
ruohikko	0.15	0.26	0.60	0.69	0.92	0.96
puuvillaverhot 0,33 kg/m² laskostettu puoleen alasta	0.07	0.31	0.49	0.81	0.66	0.54
kuten edellä, mutta laskostettu 3/4 alasta	0.04	0.14	0.23	0.57	0.53	0.40
vuodevaatteet	0.07	0.30	0.47	0.75	0.70	0.60
50 mm:n puupaneeli suoraan seinää vastaan	0.10	0.07	0.05	0.05	0.04	0.04
16 mm:n puupaneeli 40 mm:n rimoituksella	0.18	0.12	0.10	0.09	0.08	0.07
8 mm:n vaneri 50 mm:n rimoilla	0.28	0.22	0.17	0.09	0.10	0.11
3 mm:n vaneri 50 mm:n rimoilla	0.20	0.28	0.26	0.09	0.10	0.11
kuten edellä, mutta ilmavälissä mineraalivilla 50 mm	0.42	0.62	0.22	0.09	0.10	0.11
22 mm lautakatto 50 mm koolauksella ei pontattu	0.27	0.25	0.20	0.15	0.13	0.12
asbestirappaus 25 mm	0.20	0.34	0.62	0.78	0.75	0.74
40 mm avosoluinen vaahtomuovi (Superlon)	0.16	0.27	0.50	0.56	0.70	0.68
mineraalivilla (vuorivilla tai lasivilla) 40…100 kg/m³ kiinni taustassa 20 mm	0.05	0.10	0.32	0.55	0.85	0.92
30 mm	0.10	0.30	0.65	0.90	0.96	0.96
40 mm	0.20	0.45	0.80	0.92	0.96	0.95
50 mm	0.30	0.52	0.92	0.96	0.96	0.96
80 mm	0.50	0.90	0.96	0.96	0.96	0.96
100 mm	0.65	0.92	0.96	0.96	0.96	0.96
puuvillakuituruiskutus 12 mm betonipinnalle	0.02	0.05	0.28	0.45	0.65	0.85
puuvillakuituruiskutus 5 mm 20 mm:n lasivillalevyn päällä	0.13	0.29	0.73	0.88	0.86	0.86
kuten edellä, takana ilmaväli 200 mm	0.62	0.82	0.80	0.88	0.86	0.86
pinnoitettu mineraalivillalevy 10 mm, ilmaväli 200 mm	0.41	0.70	0.85	0.90	0.96	0.81
kuten edellä, ilmaväli 500 mm	0.70	0.85	0.90	0.96	0.95	0.95
lasivilla levy 10 mm, ilmaväli 250 mm	0.45	0.68	0.80	0.68	0.68	0.65
lasivilla levy 30 mm, ilmaväli 250 mm	0.45	0.96	0.92	0.96	0.96	0.96
metallilevy, reikäjako 15 mm, reikiä noin 12 % levyn alasta, mineraalivilla 20 mm, ilmaväli 50 mm	0.15	0.50	0.90	0.90	0.75	0.50
tiheäreikäinen metallilevy kuten edellä, mineraalivilla 20 mm, ilmaväli 500 mm	0.68	0.70	0.72	0.80	0.75	0.50
lastuvillalevy kiinnitetty suoraan betoniin, paksuus 25 mm	0.02	0.09	0.20	0.37	0.56	0.48
kuten edellä, paksuus 50 mm	0.07	0.19	0.38	0.60	0.60	0.57
lastuvillalevy kiinnitetty 25 mm:n rimoille, paksuus 25 mm	0.05	0.14	0.25	0.55	0.58	0.50
kuten edellä, paksuus 50 mm	0.12	0.24	0.50	0.70	0.60	0.78
lastuvillalevy 25 mm, takana mineraalivilla 50 mm, ilmaväli 25 mm	0.25	0.96	0.96	0.96	0.92	0.92
Huokoinen puukuitulevy 13 mm, ilmaväli 25 mm	0.20	0.55	0.32	0.26	0.28	0.28
kuten edellä, mutta rei'itetty puoleen levyn paksuudesta 5 mm:n rei'illä, reikäjako 15 mm	0.20	0.50	0.45	0.68	0.85	0.90
rei'itetty huokoinen, kuitulevy kuten edellä, mutta kiinnitetty suoraan alustaan	0.01	0.22	0.60	0.72	0.76	0.72
alas laskettu kipsilevykatto	0.25	0.20	0.10	0.05	0.05	0.10
levyseinä, 13 mm kipsilevyt, ilmaväli noin 100…150 mm	0.20	0.10	0.05	0.05	0.05	0.10
kuten edellä, mutta välissä mineraalivilla	0.30	0.20	0.10	0.10	0.10	0.10
3 mm:n rei'itetty kova puukuitulevy, reiät 5 mm, reikäjako 15mm, 25 mm:n mineraalivilla, 40 mm:n ilmaväli	0.13	0.73	0.85	0.72	0.58	0.51
10 mm:n rei'itetty levy, reiät 6 mm, reikäjako 25 mm, ilmaväli 50 mm	0.05	0.25	0.73	0.27	0.18	0.12
kuten edellä, mutta ilmavälissä mineraalivilla	0.35	0.60	0.89	0.65	0.45	0.28
kipsilevy, jossa tiheä neliörei'itys ja vaimennushuopa, ilmaväli 30 mm	0.10	0.20	0.45	0.70	0.72	0.57
kuten edellä, ilmavälissä 30 mm mineraalivilla	0.21	0.46	0.83	0.80	0.77	0.54
kipsilevy, jossa tiheä neliörei'itys ja vaimennushuopa, ilmaväli 200 mm	0.35	0.58	0.72	0.61	0.56	0.52
kuten edellä, mutta levyn takana välittömästi 30 mm mineraalivilla	0.41	0.66	0.78	0.75	0.70	0.64
urarei'itetty kipsilevy, jossa vaimennushuopa, ilmaväli 30 mm	0.08	0.21	0.46	0.68	0.61	0.48
kuten edellä, ilmavälissä 30 mm mineraalivilla	0.25	0.61	0.76	0.73	0.58	0.46
urarei'itetty kipsilevy, jossa vaimennushuopa, ilmaväli 200 mm	0.36	0.56	0.69	0.58	0.50	0.44
kuten edellä, mutta levynt takana välittömästi 30 mm mineraalivilla	0.40	0.63	0.73	0.65	0.58	0.51
mineraalivilla 50 mm peitetty 20*20 puulistoilla, listojen väli 20 mm	0.35	0.72	0.75	0.70	0.70	0.70
78-reikäinen reikätiili muurattu syrjälleen suoraan kiviseinän eteen	0.05	0.15	0.35	0.85	0.45	0.55
78-reikäinen reikätiili muurattu syrjälleen, ilmaväli 50 mm, ilmavälissä mineraalivilla	0.17	0.88	0.94	0.64	0.90	0.75
yleisön ja esiintyjien alue salissa	0.52	0.68	0.85	0.96	0.93	0.85
kangaspäällystettyjen hyvin pehmustettujen istuinten, joissa on rei'itetty pohja, alue salissa	0.44	0.60	0.77	0.89	0.82	0.70
nahkapäällysteisten istuinten alue salissa	0.40	0.50	0.58	0.62	0.58	0.50
puuistuinten alue salissa	0.05	0.05	0.06	0.08	0.08	0.07
modulimitoitettu kalkkihiekkareikätiili muurattu syrjälleen, ilmaväli 50 mm, jossa mineraalivilla	0.34	0.95	0.54	0.55	0.48	0.30
lasivilla 50 mm, tp 60 kg/m³, päällystetty Al-foliolla	0.45	0.51	0.95	0.92	0.67	0.57
lasivilla 50 mm, tp 60 kg/m³, päällystetty rypytetyllä Al-foliolla	0.44	0.54	0.82	0.92	0.57	0.35
kattorakenne, näkyvä pinta 0,8 mm:n teräspoimulevyä, poimut 45 mm, takana kova mineraalilevy	0.09	0.20	0.18	0.10	0.02	0.04
50 mm mineraalivilla, kattorakenne kuten em	0.35	0.80	0.96	0.96	0.96	0.96
100 mm:n levyinen rei'itetty alumiinisäle, rei'itys 13,8 %, mineraalivilla 20 mm, ilmaväli 40 mm	0.08	0.31	0.86	0.93	0.76	0.48
kuten edellä, mutta säleet ilman mineraalivillaa	0.05	0.05	0.04	0.05	0.10	0.05
kuten edellä, mutta taustaseinämään kiinnitetty 50 mm:n mineraalivilla	0.25	0.53	0.63	0.78	0.80	0.63
suorakaidekanava pelti+50 mm mineraalivilla + pelti	0.40	0.40	0.20	0.10	0.10	0.10

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4 ROOM ACOUSTICS

4.1. Introduction

Room acoustics is a very diverse concept, the content of which depends on the purpose of the room.

In the Building Code of Finland (Ministry of the Environment, 2017–2018), requirements directly affecting room acoustics are presented with five measurable parameters:

1. reverberation time *T*, determined in octave bands from 250 to 2000–Hz;
2. speech transmission index *STI,* defined in octave bands from 125–to 8000 Hz;
3. background sound level
   - A-weighted equivalent SPL* in MEP in frequency range 20 –20000 Hz, L_{Aeq};
   - maximum SPL with Fast time weighting in frequency range– 20 20000 Hz, L_{AFmax}.
   - A-weighted equivalent SPL* outside the building in the range 20–20000 Hz, L_{Aeq}.

* These values include the necessary impulse and narrowband correction.

Background noise can also be caused by neighbouring rooms that interfere with room acoustics. In this sense, the air and impact sound insulation of the structures bordering the room are also related to room acoustic objectives. However, this chapter focuses on quantities 1 and 2, since background sound control and soundproofing are discussed in other chapters. The exception is speech overlay audio.

Building regulations impose requirements on reverberation time in stairwells, teaching facilities (classrooms), conference rooms, dining rooms, patient rooms, doctor's offices, open-plan offices, offices, sports facilities, and care and hobby facilities. The reverberation time is controlled by absorption materials. Minimising reverberation time is also a key noise abatement measure, for example in engine rooms, manned production facilities, lobbies, libraries, restaurants, public spaces and retail spaces, although there are no actual regulations related to these spaces.

In performance spaces, room acoustic design aims to ensure that the performer's voice (speech, instrument, speaker voice) is conveyed as cleanly as possible to the listeners throughout the room. This is done by setting appropriate target levels for quantities 1–to 5, such as a high STI value, an appropriate reverberation time and the lowest possible background noise level. In addition, in medium and large spaces, the geometry of the room must be appropriately designed to support the transmission of the performer's voice to the listeners.

Building regulations do not set target levels for large performance spaces, such as cinemas, theatres, auditoriums, multipurpose facilities, ice rinks and concert halls, because the target levels are determined by the purpose or purposes. An acoustic designer is then needed both to define the target levels and to plan the implementation according to the target levels. In large farms, target levels can also be set for so-called target levels. concert hall volumes. In some situations, scattering surfaces are also used to avoid increasing absorption but avoiding mirror reflection. Large performance spaces are always accompanied by electronic audio systems. Increasingly, large performance spaces also include an electronic room acoustic modification option, in which the reverberation time of the space can be artificially extended if the presentation material so requires.

In speech and performance spaces, it is also important to take into account the room acoustics experienced by the performer, not just the audience. The performer must constantly have the experience that listeners hear the sound well so that he does not have to change the way he is presented.

In open-plan offices, group teaching rooms, libraries, healthcare facilities and public waiting rooms, it is appropriate that (confidential) conversations do not extend beyond the normal conversational distance (1-3–m). For this reason, the concept of room acoustics was expanded in the 2000s by attaching the concept of speech privacy to it. Good speech privacy is supported by a low STI value, which is achieved with a short reverberation time and an appropriate background sound level. If necessary, the level of the background sound is artificially increased by means of a so-called "resound". using a speech overlay sound system.

4.2 Diffuse field intensity for room surface

Consider a room where a single point sound source produces a steady flow of sound energy into the space and a diffuse sound field prevails in the space. This occurs if the absorption ratio α is constant and comparatively low for all room surfaces with a combined area *of S*. In addition, it is required that the room is cubic and all room dimensions are clearly larger than the wavelength of the sound being considered. In addition, absorption in air is negligible and therefore disregarded in the examination. This usually occurs when the room humidity is high, the room volume is limited (less than 1000 m3) and the sound frequency is less than 5 kHz. The room has the same SPL at all points and the sound intensity is zero except near the sound source and on room surfaces where sound is absorbed at $S\alpha Ii$ *[W*], where *Ii* [W/m2] is the intensity hitting the surface. This chapter defines this intensity.

Thinking about the image **4.2.1** volumetric embryo *dV [m3*] where energy density prevails ε [J/m3]. It is not a sound-producing point sound source, but a volumetric element to which sound has been carried from a diffuse sound field. In the following, we will find out to what extent sound is emitted from the volume embryo to the area element dS of the room surface, which is at a distance *r* volumetric embryo. In a diffuse room, the same energy density prevails at each point of the room and the total energy in the volumetric embryo *Dv* is εδς.

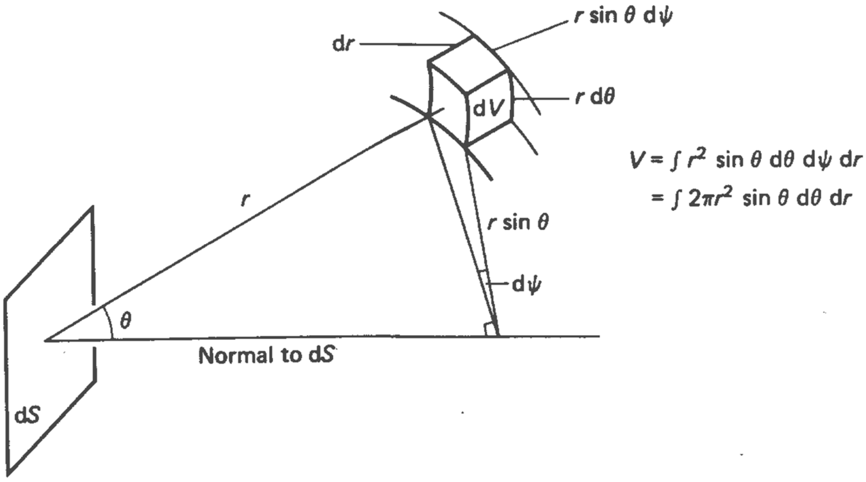


Fig. 4.2.1 The volume element *dV* of the diffuse sound field emits sound to the room surface element *dS*. The phenomenon is studied in the spherical coordinate system.

From *r* The spherical surface located from the volumetric embryo has an area of $4\pi r^2$. Since in the diffuse field sound radiates in all directions with the same power, the surface element dS is hit by the sound energy of the volume embryo εdV proportion dS·cos(θ)/4πr^2 when θ is the projection angle for the surface element *DS*. For surface embryo *dS* thus radiates from the volume embryo dV an energy germ

(4.2.1) $$\Delta E = \frac{\varepsilon dV dS \cos\theta}{4\pi r^2}$$

The total energy arriving on the surface from the entire room is obtained by integration in terms of volume. The integration unit is a hollow spherical shell with a volume as shown in Fig. 4.2.1

(4.2.2) $$dV = 2\pi r^2 \sin\theta d\theta dr$$

Since the surface element is on the room surface, the total volume of the room is covered by integrating the θ ratio of , over halfspace:

(4.2.3) $$dE = \int_0^{\pi/2} \frac{\varepsilon dS \cos\theta}{4\pi r^2} 2\pi r^2 \sin\theta d\theta dr = \frac{\varepsilon dS\, dr}{4}$$

This energy reaches the surface in the time embryo Δt, which is consumed by sound passing over the volume element dV, i.e.

(4.2.4) $$dt = dr / c_0$$

Therefore, the sound power on the surface *dS* is

(4.2.5) $$W_i = \frac{dE}{dt} = \frac{\varepsilon c_0 dS}{4}$$

Surface intensity I_i (i.e. power per unit surface) is therefore $\varepsilon c_0/4$. Thus, the room surface is hit by a quarter of the intensity that would prevail with a plane wave with energy density εc_0. Since the energy density can be calculated from the effective sound pressure by the equation

(4.2.6) $$\varepsilon = \frac{\tilde{p}^2}{\rho_0 c_0^2}$$

obtained intensity on the wall surface of the room

(4.2.7) $$I_i = \frac{\varepsilon\, c_0}{4} = \frac{\tilde{p}^2}{4\rho_0\, c_0}$$

This can be converted to decibel form by first dividing the equation by the reference intensity $I0=1\cdot10^{-12}$ W/m$_2$ and taking the bilateral logarithm

(4.2.8) $$\log_{10}\left(\frac{I_i}{I_0}\right) = \log_{10}\left(\frac{\tilde{p}^2}{I_0 4\rho_0\, c_0}\right).$$

Considering that $\rho_0 c_0$=412 kg/m^{2}s ~ 400 kg/m^{2}s *gives*

(4.2.9) $$\log_{10}\left(\frac{I_i}{I_0}\right) = \log_{10}\left(\frac{\tilde{p}^2}{4\cdot 10^{-10}}\right) - \log_{10}(4)$$

The second term is constructed as the square of the reference SPL, p_0 = 20 μPa. Multiplying the equation by ten gives

(4.2.10) $$10\cdot\log_{10}\left(\frac{I_i}{I_0}\right) = 10\cdot\log_{10}\left(\frac{\tilde{p}^2}{p_0^2}\right) - 10\cdot\log_{10}(4)$$

can be obtained from this by simplifying

(4.2.11) $$L_{Ii} = L_p - 6$$

L_p stands for diffuse sound field SPL. In the diffuse sound field, the SPL is constant. Since no room is completely diffuse, the SPL is determined as the average volume of a room outside the reverberation radius measured from the sound source. The equation is applied very extensively in building and room acoustics. Examples include reverberation time and different definitions of sound insulation.

The previous equation can also be used to determine the sound power level $L_{W,e}$ [dB] directed at a specific structural element *e* of the room surface when the sound level of the room is known:

(4.2.12) $$\boxed{L_{W,e} = L_{I,i} + 10\cdot\log_{10}(S_e) = L_p + 10\cdot\log_{10}(S_e) - 6}$$

where *Se* [m2] is the area of the structural element under consideration, and L_p [dB] is the SPL of the diffuse sound field.

4.3. SPL in diffuse sound field

Make the same assumptions as **in Section 4.2.** The intensity reaching the room surface from the diffuse sound field is a quarter of the intensity of a level wave with the corresponding SPL:

(4.3.1) $$\frac{\tilde{p}_r^{\,2}}{\rho\, c} = 4I_i = 4\frac{W}{A}$$

where W [W] is the power of the sound source. In the following, consider the intensity I_i, which enters the absorbent material, which is present in the room in the amount A [m^2] and is evenly distributed over the room surfaces. The sound pressure p_r [Pa] refers to the sound pressure of the diffuse sound field (*r=reverberant*).

When you get close enough to a point sound source (reverberation radius), the relative proportion of sound pressure emitted directly from the sound source to the total sound pressure begins to be significant. The intensity radiated by a point sound source I_d [W/m^2] is inversely proportional to the square of the distance r between the point source and the point of observation

(4.3.2) $$I_d = \frac{\tilde{p}_d^{\,2}}{\rho_0 c_0} = \frac{kW}{\Omega r^2}$$

where p_d [Pa] is the sound pressure caused by direct sound, *k [-]* is the direction coefficient of the sound source in the direction of view, and [-] Ω must be considered the solid angle (**Chapter 2**).

The sum of direct (d) and anechoic (r) sound pressure, p_{tot} [Pa], is

(4.3.3) $$\tilde{p}_{tot}^2 = \tilde{p}_d^2 + \tilde{p}_r^2$$

and thus

(4.3.4) $$\frac{\tilde{p}_{tot}^2}{\rho_0 c_0} = W\left(\frac{k}{\Omega r^2} + \frac{4}{A}\right)$$

Taking the decimal logarithm and multiplying by ten, we get an equation that multiplies the SPL at different distances *r* [m] from the point sound source in the diffuse field:

(4.3.5) $$\boxed{L_{p,tot} = L_W + 10 \cdot \log_{10}\left[\frac{k}{\Omega r^2} + \frac{4}{A}\right]}$$

Fig. **4.3.1a** shows the behavior of the equation, taking into account only the first term in brackets (direct sound), only the second term (reverberant sound), or both terms (resultant). The first term is ignored if the measurement or calculation is carried out in an echo field, i.e. outside the echo radius.

The echo radius *rR* is called the distance where the direct sound and the echoic sound are equally strong. At the distance corresponding to the echo radius, the terms in the previous parenthesis should be equal:

(4.3.6) $$\frac{k}{\Omega r_R^2} = \frac{4}{A}$$

and we get

(4.3.7) $$r_R = \sqrt{\frac{kA}{4\Omega}}$$

In the case of a surface non-directional sound source (k=1, Ω=2π), the equation takes the form:

(4.3.8) $$r_R \approx \frac{\sqrt{A}}{5}$$

Fig. **4.3.1b** illustrates the value of the echo beam in different absorption sectors. The echo radius is long when *A* is large. Diffuse sound field calculations and measurements are usually made outside the echo radius. Therefore, there must be a good idea of the size range of the echo beam. With a directional sound source, the echo radius can be significantly increased in the maximum directions of directivity.

Equation **4.3.5** can also be used to determine the sound power level of a sound source in an echoic room by SPL measurements. The measurement shall be made on an imaginary measuring surface with an area *of S* [m^2]. First, suppose that the measuring surface is a sphere, in which case its area is $S = 4\pi r^2$, where are is the distance of the measuring surface to the center of the sound source. In this case, equation **4.3.5** gives the sound power level

(4.3.9) $$L_W = L_{p,tot} + 10\log_{10}(S) - 10 \cdot \log_{10}\left[1 + \frac{4S}{A}\right]$$

where $L_{p,tot}$ [dB] is the mean SPL on the measuring surface. The direction factor was set to *k=1* because the

SPL is determined in all directions. In practice, the shape of the measuring surface does not have to be a sphere, as long as the sound source is contained inside the measuring surface. If the sound source is on a hard surface, the surface area of the substrate is not taken into account in the calculation of S. Equation **4.3.9** means that the intensity level of the sound source towards the surface, even in an echoic environment, is the same as if it were in a free field: the third term is zero when there is no echo. The SPL on the measuring surface increases by a third term due to echoing.

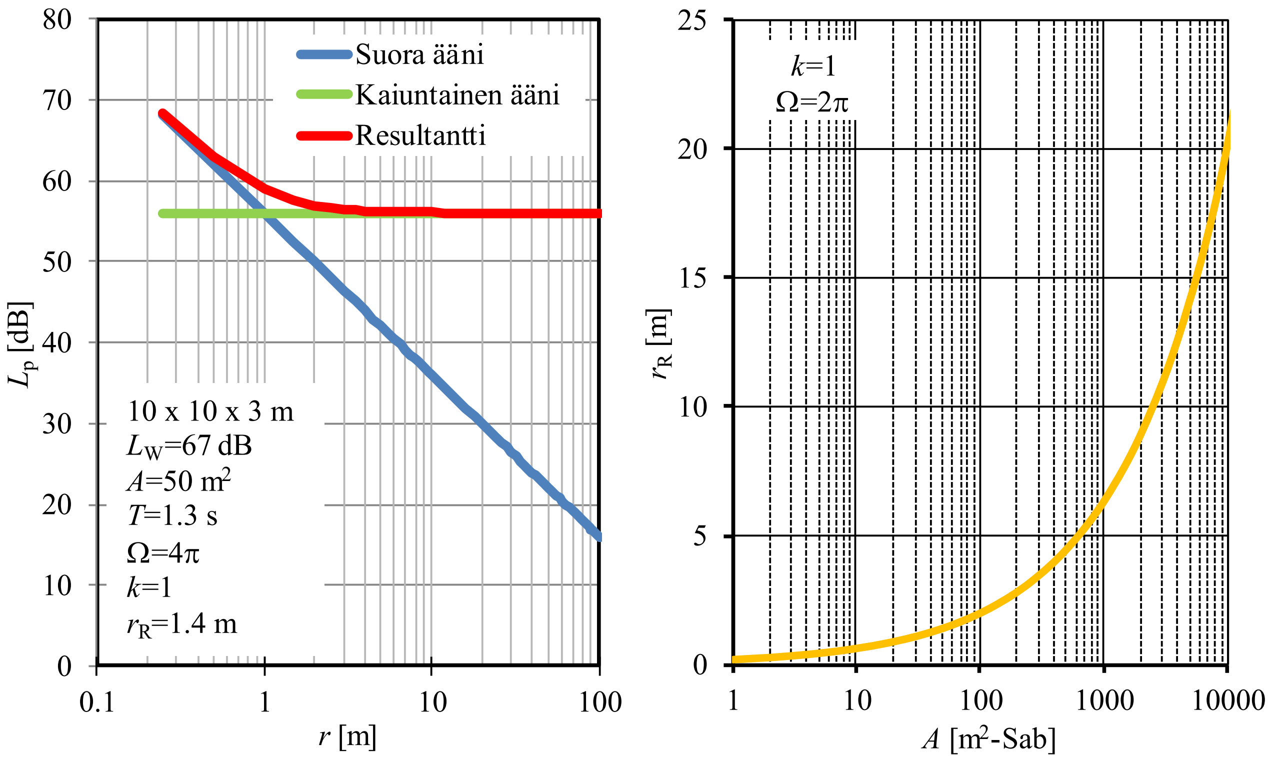


Fig. 4.3.1 (a) SPL Lp of direct sound, reverberant sound and the preceding resultants as a function of the distance r from the sound source in a room. (b) The echo radius rR of the non-directional sound source placed on the room surface as a function of the absorption area A of the room. Both images depend on frequency, because A depends on frequency.

4.4 Reverberation time in diffuse sound field

Assume the diffuse sound field according to section **4.2.** The total acoustic energy E [J] of a room is the energy density ε [J/m^3] multiplied by the room volume V [m^3] :

(4.4.1) $$E = \varepsilon V$$

Feed the room with sound that is silenced at $t=0$. Subsequently, the change in acoustic energy as a function of time depends on the absorption power at room surfaces, $S\alpha I$, by the equation

(4.4.2) $$-V\frac{d\varepsilon}{dt} = S\alpha I = AI$$

where $A=\alpha S$ is the total absorption area of the room. The intensity reaching room surfaces from diffuse sound fields depends on the energy density by the formula (derived from **Chapter 4.2**)

(4.4.3) $$I = \frac{\varepsilon c_0}{4}$$

When this is differentiated over time, we get

(4.4.4) $$\frac{\partial \varepsilon}{\partial t} = \frac{4}{c_0}\frac{\partial I}{\partial t}$$

Equation **4.4.2** then takes the form

(4.4.5) $$\frac{dI}{dt} = -\frac{A c_0}{4V} I$$

and on

(4.4.6) $$\frac{dI}{I} = -\frac{A c_0}{4V} dt$$

Integration on both sides with respect to time yields by applying a formula $\int dx/x = \ln x$

(4.4.7) $$\ln I = -\frac{A c_0}{4V} t + \text{constant}$$

The initial intensity value lnI0 at t0 is recorded as the constant, giving

(4.4.8) $$\ln I - \ln I_0 = \ln \frac{I}{I_0} = -\frac{A c_0}{4V} t$$

which gives a time dependence of the intensity that hits the surface

(4.4.9) $$\frac{I}{I_0} = e^{-\frac{A c_0}{4V} t}$$

The absorption area A of the room can be determined by determining the attenuation of intensity as a function of time after turning off the sound. This is done by determining the reverberation time. The reverberation time T [s] is defined as the time during which intensity I decreases to one millionth of the initial intensity I_0, i.e.

(4.4.10) $$\frac{I}{I_0} = 10^{-6}$$

By taking the decimal logarithm and multiplying by ten, we get a change in the intensity level

(4.4.11) $$10 \cdot \lg \frac{I}{I_0} = L_I - L_{I_0} = -60$$

that is, the sound intensity has been reduced by 60 dB from the original. Fig. **4.4.1** shows the decrease in intensity and intensity level as a function of time. Since $\log_a N = \log_b N / \log_b a$, we get the left-hand term as

(4.4.12) $$10 \cdot \lg\left(\frac{I}{I_0}\right) = \frac{10}{\ln(10)} \cdot \ln\left(\frac{I}{I_0}\right) = -\frac{10}{2,3} \frac{A c_0}{4V} T$$

This solves reverberation time and results in ***Sabine's equation***

(4.4.13) $$T = \frac{55.3V}{c_0 A}$$

which is at room temperature 20°C roughly

(4.4.15) $$\boxed{T = \frac{0.16V}{A}}$$

The Sabine equation is used to estimate reverberation time and the amount of absorption material. In addition, it is used to determine the statistical absorption ratio of absorption materials: by measuring the reverberation time of a room without material and with a material with a known surface area, the change in absorption area and absorption ratio can be determined.

Mm. Eyring, Millington, Cremer, Kuttruff, Fitzroy and Arau-Puchades present well-founded alternative reverberation time equations to equation 4.4.5 (Keränen and Hongisto, 2010). However, these are not presented because they generally do not achieve a significant improvement in accuracy compared to the Sabine equation, taking into account the uncertainties of the absorption ratios and reverberation time measurements reported for the materials.

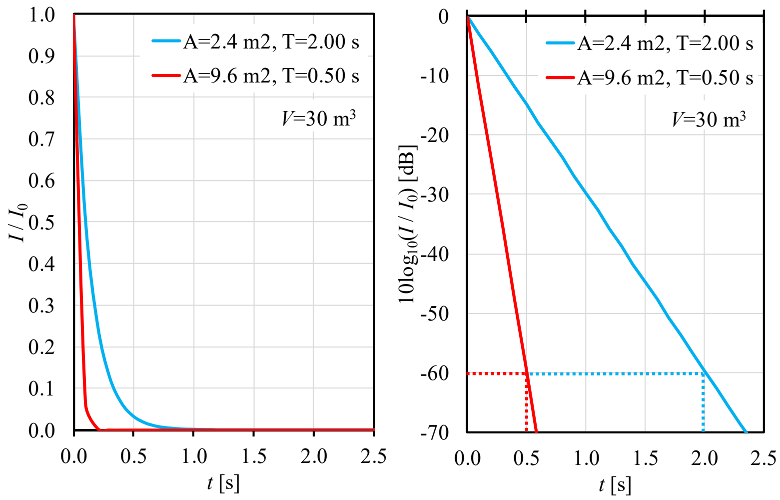


Fig. 4.4.1 Attenuation of the impulse produced at t=0.00 as a function of time t in a room of 30 m3 with an ideal diffuse sound field. The time profile of absolute intensity [dB] is shown on the left, and the time profile of the intensity level [dB] is shown on the right. On the right, dotted lines show the determination of reverberation time T.

4.5 Impulse response of a room

The state impulse response function r (later: impulse response) tells us how signal a changes when it moves from the sound source from point A to point B. The impulse response is obtained by measuring the impulse produced at point A at point B. The impulse is the dirac delta function δ, which is zero everywhere except zero at point in time, where it is theoretically infinite:

(4.5.1) $$\delta(t)=\begin{cases}+\infty, & t=0\\ 0, & t\neq 0\end{cases} \quad ja \quad \int_{-\infty}^{+\infty}\delta(t)\,dt=1$$

The Fourier transform of the delta function is constant, that is, it includes all frequencies. In practical measurements, the time is discretised and the impulse length is then the inverse of the sampling rate, i.e. of the order of 23 microseconds if the sampling rate is 44 kHz. The delta function of this length includes all frequencies up to 20 kHz. It is difficult to produce such a short impulse strong enough with a speaker. Therefore, impulse response used to be determined using impulse excitations such as gunshot sound and nowadays using loudspeaker assistive methods of digital signal processing, such as maximum length sequence and sine sweep. They are not covered in this course.

In room acoustics, the impulse response is determined using a microphone. The impulse response describes the sound pressure as a function of *p time* t at B when the excitation is the impulse at A. Since the impulse response is a discrete function (sound samples are taken at finite time resolution, usually 44 kHz or less), the impulse response appears unnatural up close. When the impulse response is viewed in a broader perspective (Fig. **4.5.1a**), it sees sound pressure damping as a function of time.

From the point of view of room acoustic interpretation, a more useful representation than the impulse response is the amplitude of the impulse response or the squared impulse response, i.e. *p2*. It can be used to examine, for example, when direct sound and first reflections reach the listener. Fig. **4.5.2** shows a simplified representation of the squared impulse response. The measured responses differ from this simplified representation in that between the lines *p2* does not fall anywhere near zero and the maxima stand out less well.

The most common representation is the logarithm of the squared impulse response (Fig. 4.5.1c), which is often referred to as either an echo curve (or, erroneously, an impulse response). It is obtained from the impulse response by applying to all observation points $p(t)$ of the signal, for example, the SPL equation if the sound pressure is calibrated:

(4.5.2) $$L_p = 10 \cdot \log_{10}\left(\frac{p^2}{p_0^2}\right)$$

In impulse response measurement, sound pressure does not always have to be calibrated. However, some quantities derived from the level representation of the imputation response require the use of a calibrated sound source, in which case calibration is necessary.

From the reverberation curve, reverberation time can be determined by matching it directly to the section straight. However, the determination should avoid the upstream (first reflections) and downstream curved area where the reverberation curve reaches the background noise. In general, reverberation time is determined by measuring -5 and -25 dB of attenuation time and multiplying it by three. The subentry for this method of determination is T_{20}.

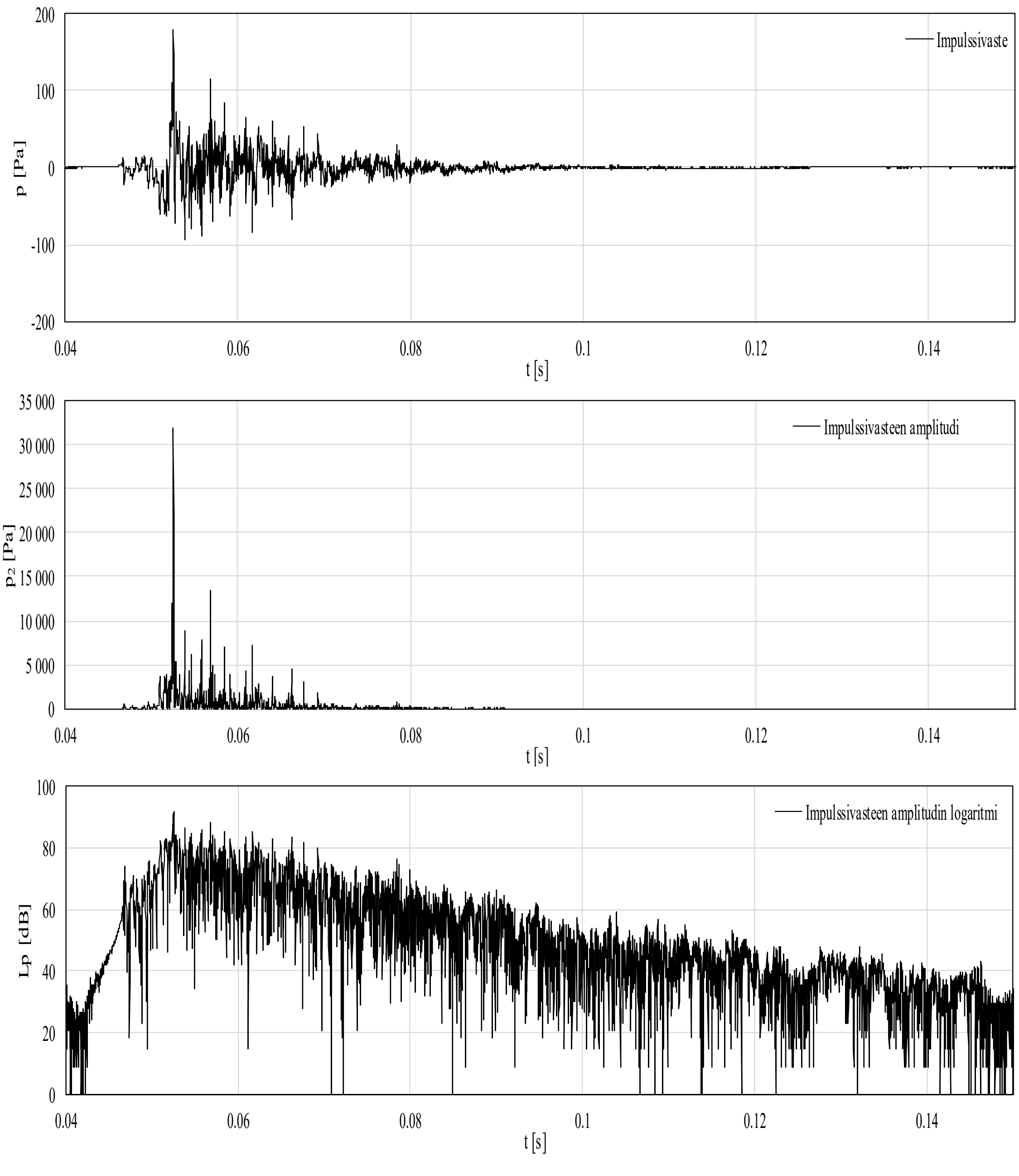
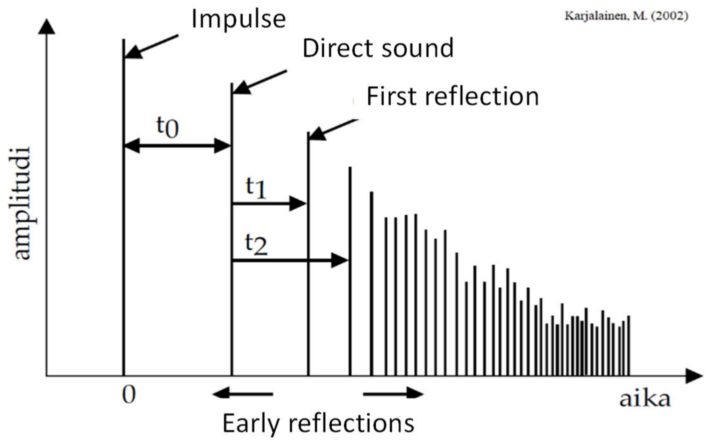


Fig. 4.5.1 Impulse response and derived impulse response amplitude and logarithm of amplitude. The measurement was made in a semi-anechoic room with little furnishings. The sound source A is 3 m from measuring point B. The sampling rate is 44100 Hz.



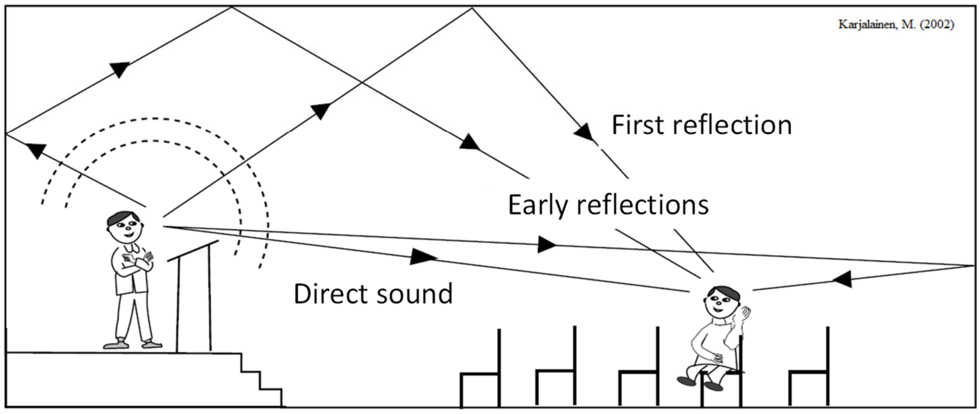


Fig. 4.5.2 A simplified representation of the squared sound pressure in the space of the lower image. The listener receives direct sound first, followed by first-degree reflections of the room surface, and finally reflections of several degrees, which blend together and cause reverberation of the space.

4.6. Reverberation time calculation

The Sabine equation is used to determine the required absorption area A [m2] knowing the room volume V [m3] and the target reverberation time T [s]:

(4.6.1) $$A = \sum_{i=1}^{n} A_i = \sum_{i=1}^{n} \alpha_i S_i = \frac{0.16V}{T}$$

The example in Fig. **4.6.1** shows a calculation for an unfurnished room. For example, if the floor area S_1 = 69.1 m^2 and its absorption ratio in the 125 Hz octave band α_{125} = 0.03, we get the floor absorption area $A_{1.125}$ = $\alpha \cdot S$ = 2.1 m^2. In the case of Fig. **4.6.1**, the roof and rear wall are most affected by reverberation time. Taking these two surfaces into account alone would achieve almost the same result as taking into account all 7 surfaces.

The accuracy of the calculation method is acceptable if the absorption ratios and surface areas have been correctly determined and the absorption material is distributed over more than one surface. The accuracy is good, less than ±0.2 seconds, when the space is small, cubic, open, echoing, and the absorption is evenly distributed over different surfaces. In non-cubic strongly furnished spaces, the error is greater than this. Similarly, if all absorption material is placed on a single room surface (e.g., ceiling) and the room is otherwise unfurnished, Sabine's formula provides a shorter reverberation time than reality, because the material does not effectively absorb the horizontal sound field relative to the material surface (e.g., between walls).

Reverberation time is also heavily influenced by furniture and people. The absorption area of a person normally dressed indoors can be 0.5 m^2 at all frequencies.

HUONEMITAT

Pituus x:	10.0	m	Pohjapinta-ala:	70.0	m^2
Leveys y:	7.0	m	Tilavuus *V*:	210.0	m^3
korkeus z:	3.0	m			

Pintojen i pinta-alat S_i ja absorptiosuhteet α_i oktaaveittain **f [Hz]**

i		S_i [m^2]	125	250	500	1000	2000	4000
1	Betonilattia (ei kalusteita)	70.0	0.03	0.03	0.03	0.03	0.03	0.04
2	Katon alaslaskettu absorptiolevy	70.0	0.60	0.71	0.75	0.58	0.51	0.42
3	Ikkunaseinä (lasi)	21.0	0.30	0.30	0.20	0.17	0.10	0.10
4	Takaseinä (betoni)	21.0	0.03	0.03	0.03	0.03	0.03	0.03
5	Käytävänpuol. seinä (kipsilevyseinä)	21.0	0.30	0.20	0.10	0.10	0.10	0.10
6	Etuseinä (betoni)	21.0	0.03	0.03	0.03	0.03	0.03	0.03
7	Takaseinän lisäabsorptiolevy	8.4	0.15	0.75	0.97	0.99	0.99	0.96

Pintojen i absorptioalat, $A_i=\alpha_i \cdot S_i$ [m^2]

			125	250	500	1000	2000	4000
1	Betonilattia (ei kalusteita)	A_1	2.1	2.1	2.1	2.1	2.1	2.8
2	Katon alaslaskettu absorptiolevy	A_2	42.0	49.7	52.5	40.6	35.7	29.4
3	Ikkunaseinä (lasi)	A_3	6.3	6.3	4.2	3.6	2.1	2.1
4	Takaseinä (betoni)	A_4	0.6	0.6	0.6	0.6	0.6	0.6
5	Käytävänpuol. seinä (kipsilevyseinä)	A_5	6.3	4.2	2.1	2.1	2.1	2.1
6	Etuseinä (betoni)	A_6	0.6	0.6	0.6	0.6	0.6	0.6
7	Takaseinän lisäabsorptiolevy	A_7	1.3	6.3	8.1	8.3	8.3	8.1
	Kokonaisabsorptioala (edellisten rivien summa)	A	59.2	69.9	70.3	57.9	51.6	45.7

Jälkikaiunta-aika, T=0.16·V/A [s]

	125	250	500	1000	2000	4000
T	0.57	0.48	0.48	0.58	0.66	0.74

Fig. 4.6.1 An example of applying the Sabine equation in a classroom, considering 7 room surfaces.

4.7. Relative humidity and reverberation time

Room air absorbs sound. The drier the air in the room and the higher the frequency, the more absorption occurs. In Sabine's equation, this can be accounted for by an additional term

(4.7.1) $$T=\frac{55.3V}{c_0\left(A+4mV\right)}$$

where *m [1* /m] is the room air attenuation coefficient (Table **4.7.1**). The relative humidity determines the maximum reverberation time of the room. The longest possible reverberation time for a diffuse room at room temperature can also be determined from the equation (Cremer &; Müller, 1978)

(4.7.1) $$T=\frac{2.4\cdot RH}{f_k^2}$$

where *RH* [%] is the relative humidity and *fk* [kHz] is the frequency in kilohertz.

Table 4.7.1 Room air attenuation coefficient m [1/m] at room temperature at different relative humidity *RH.* At lower frequencies, the coefficient does not matter. In sub-zero temperatures*, RH* can fall below 10% indoors (Rindel, 2018).

RH [%]	*f* [kHz]			
	1	2	4	8
40	0.0011	0.0026	0.0072	0.0237
50	0.001	0.0024	0.0061	0.0192
60	0.0009	0.0023	0.0056	0.0162
70	0.0009	0.0021	0.0053	0.0143
80	0.0008	0.002	0.0051	0.0133

4.8. Specific frequencies

Room echoes arise from the fact that the space "rings" with its characteristic frequencies. Rooms of different sizes and shapes sound different, even if their absorption area is the same, because they have different characteristic frequencies. At frequencies other than characteristic frequencies, echoes do not occur. Since characteristic frequencies have non-zero bandwidth, non-zero reverberation times can also be measured at frequencies between characteristic frequencies.

The specific frequencies of a cubic room are calculated by the equation

(4.8.1) $$f(n_x, n_y, n_z) = \frac{c_0}{2}\sqrt{\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2} \qquad n_x, n_y, n_z = 0, 1, 2, \ldots$$

where l_x, l_y, and l_z [m] are the dimensions of the room in the directions x, y and z. Characteristic frequencies are also called modes, room modes, room resonances or characteristic frequencies. A standing wave does not require a closed volume to form, so it is not the best designation. A standing wave field is also formed in a free field when sound hits a single reflective surface. Moreover, a standing wave is always generated at all frequencies, while the characteristic frequency occurs only at certain frequencies.

Axial, tangential and oblique modes can be distinguished (Fig. **4.8.1**). In the spectrum, the most distinctive modes are axial, since they occur at the lowest frequencies and are usually separate from tangential and especially oblique modes.

The pressure maximum of the lowest axial mode occurs at the room surface and the pressure minimum occurs at the centre of the room (Fig. **4.8.2**). The lowest axial mode can detect fluctuations in SPLs of up to 20 dB between different points in the room at that frequency, if sound is available at that frequency.

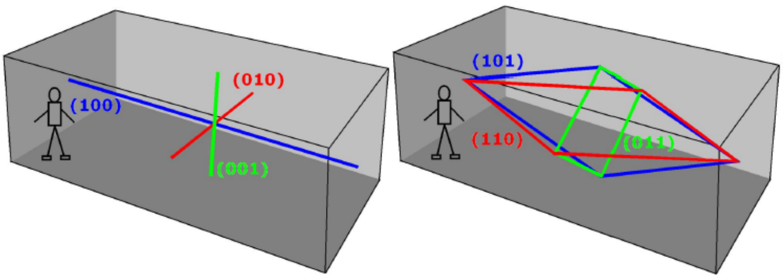


Fig. 4.8.1 Axial modes (100), (010) and (001) and tangential modes (110), (011) and (101). The lowest oblique mode (111) is difficult to describe understandably with the help of a level drawing. (Oliva et al., 2011)

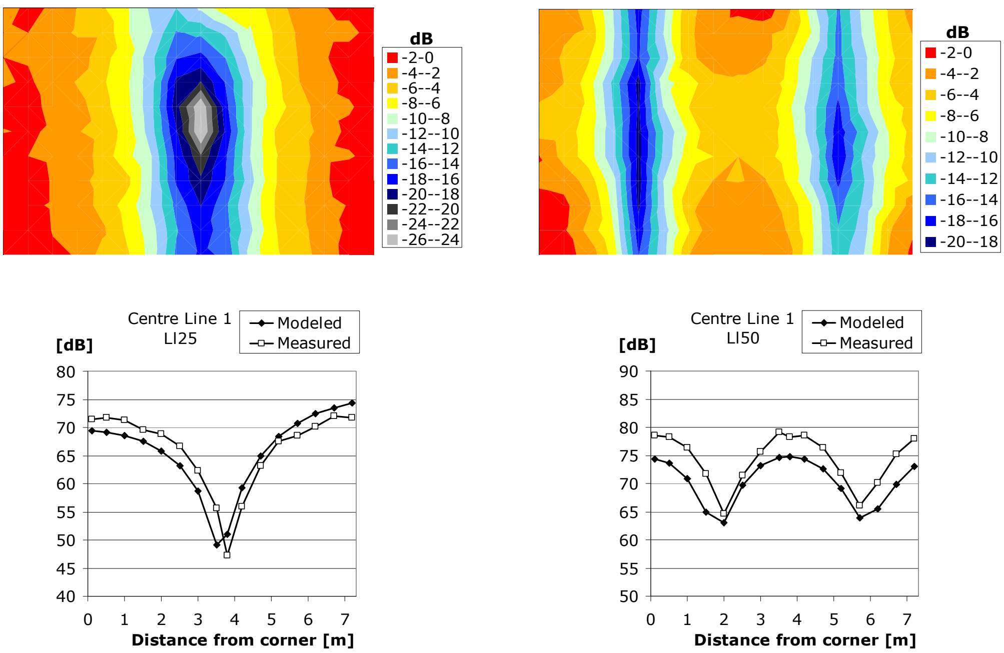


Fig. 4.8.2 Behaviour of the SPL in the 25 and 50 Hz bands in an echo chamber of 7.5 m, 4.9 m and 4.2 m (155 m3) in length, width and height. The top row has a distribution of SPL on the horizontal surface at a height of 1.5 m from the floor. The lowest SPLs are in dark blue and the highest in red. Modeling was done using COMSOL. The left image shows the mode (100) and the right mode (200). The bottom row shows the modelled and measured SPL of the terse band on a longitudinal measuring line located 1.5 m from the floor. The sound entered the room from a 0.6x1.0 m wall opening (red spot in the middle at the top). (Oliva et al., 2011).

The room does not resonate, i.e. echo at frequencies lower than the lowest axial mode. In this case, the mode is the so-called. constant pressure chamber. In this case, the sound pressure that forms in the room is forced vibration and the room does not amplify the sound pressure due to the absence of characteristic frequencies. In a constant pressure chamber, the SPL is the same between different measuring points.

The modes can be thought of as being located in the mode lattice, as shown in Fig. 4.8.3. In this case, one mode occupies the volume in that space

(4.8.2) $$V_m = \frac{c_0}{2l_x}\frac{c_0}{2l_y}\frac{c_0}{2l_z}$$

The number of characteristic frequencies N below any frequency f is obtained by dividing the volume according to that frequency by the volume of the mode

(4.8.3) $$N = \frac{\frac{1}{8}\frac{4\pi f^3}{3}}{V_m} = \frac{4\pi f^3}{3c_0^3} l_x l_y l_z$$

The formula ignores modes at the edges of the mode space where one of the integer coefficients nx, ny, or nz is zero. Similarly, axisal modes are ignored, where two of the three integer coefficients are zero. A more accurate equation is

(4.8.4) $$N \approx \frac{4\pi f^3}{3c_0^3}V + \frac{\pi f^2}{4c_0^2}S + \frac{f}{8c_0}L$$

where V [m^3] is the room volume, S [m^2] is the area of the room surfaces, and L [m] is the total length of the edges of the room.

The diffuse sound field assumes that the room has a large number of characteristic frequencies (at least ten discrete) for each frequency band under consideration. This only occurs above the Schröder limit frequency *fs*

(4.8.5) $$f_S = 2000\sqrt{\frac{T}{V}}$$

where V [m^3] is the room volume and T [s] is the reverberation time. Above the limit frequency, discrete modes are no longer observed, but overlap with each other, and the statistical assumption of the diffuse field can be realized.

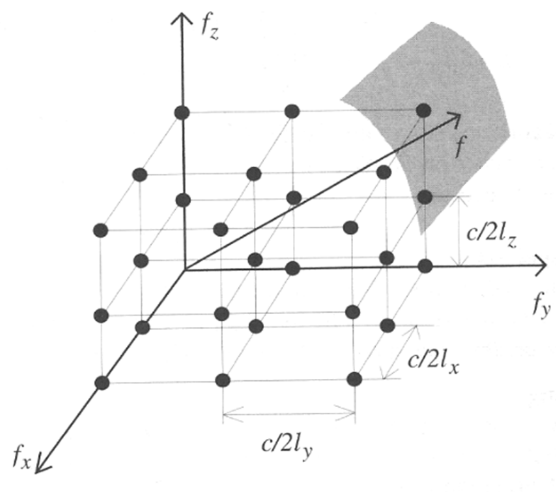


Fig. 4.8.3 The mode lattice is located at the positive values of the frequency space.

4.9. Spatial attenuation

The reverberation time T describes the temporal attenuation rate of sound locally. Reverberation time is related to the subjective experience of sound quality (speech separation, music quality) in spaces where the purpose is to hear sound well. The reverberation time is used to determine the absorption ratio of materials and the sound insulation of structures. Reverberation time has spread to almost all room acoustic applications because the equations of Sabine and room attenuation are easy to apply. The reverberation time describes the basic acoustic properties of a space well if the room is sufficiently diffuse and the measurement results correspond with sufficient accuracy to the prediction models of the squared equations of 4.2–4.4. These conditions are realised with reasonable precision in empty living rooms, school classrooms, small auditoriums, gymnasiums and also in larger premises with little furnishings.

In a diffuse sound field, the SPL is constant outside the reverberation radius. However, in a non-diffuse sound field, the SPL continues to decrease sharply as the distance to the sound source increases, and does not stabilize at the echo beam or any other distance. Diffuse field conditions typically do not apply when at least one of the following factors is true:

- at least one dimension of the room is many times the height of the room or the room is not rectangular,
- there is a large amount of absorption material in the room,
- The room has high and dense furnishings that prevent direct sound from being partially or completely transmitted from the sound source to the listener.

In large workspaces such as open-plan offices and industrial halls, all of the above factors usually apply. Acoustic goodness is assessed subjectively on the basis of how quickly sound attenuates with increasing distance to the noise source (spatial decay) and not according to how quickly the noise attenuates over time at a given point (reverberation time). The reverberation time is not suitable for describing the room acoustics of large workspaces, because the space no longer behaves as a single entity as required by the diffuse field. The SPL strongly depends on the distance to the sound source. For example, the impulse response and the reverberation time determined from it depend on the distance from the sound source at which the measuring microphone is placed. The shape of the echo curve is also different if the measurement is made near or far from the sound source.

In clearly non-diffuse spaces, a more suitable parameter for describing the quality of room acoustics is spatial decay, i.e. the dependence of the SPL on the distance to the sound source. According to studies, sound is linearly attenuated in such states as a function of logarithmic distance. From the spread attenuation curve, we can determine the spatial decay rate, D_2 [dB], which indicates the decrease in SPL as the distance is doubled:

(4.9.1) $$D_2 = L_p(r_1) - L_p(r_2)$$

where $r_2=2r_1$. D_2 is always a positive number and is determined from observations by making a linear fit between the SPL (y: Lp) and the logarithmic distance (x: lnr). Observations less than 2 metres from the sound source are ignored because closer to this, the sound is attenuated in a free-field manner.

Examples of spatial decay rates are shown in Fig. **4.9.1.** The spatial decay rate can be predicted either with regression equation models based on empirical data or with a room acoustic computer model.

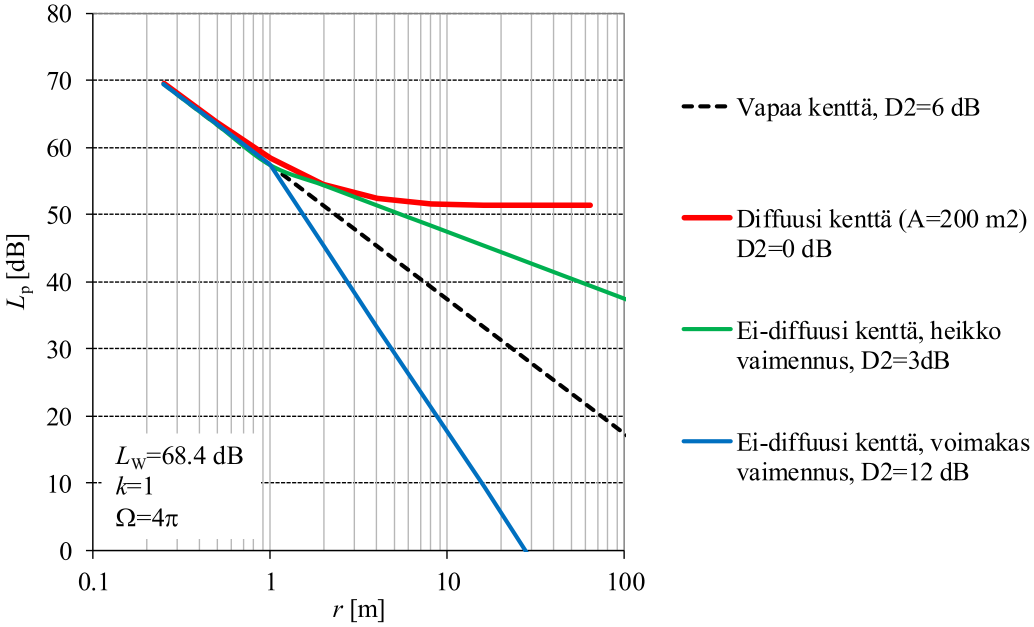


Fig. 4.9.1 Spatial decay curves in different environments. The diffuse room and free field are calculated by equation **4.3.5.** Other decay curves are from environments with high damping and diffuse field assumptions do not apply. Very high spatial decay rates (D_2>8 dB) occur in open-plan offices, where near the sound source the attenuation follows a free field behind the first high screen, the slope stabilizes at its final D2 value.

4.10 Voice

Male speech contains frequencies 100−10000 Hz and female speech frequencies 200−10000 Hz, because the basic voice of a man is about an octave lower than that of a woman. Vowels are located in the middle frequencies of 500–2000 Hz, and consonants mainly at high frequencies, above 2 kHz. The volume of vowels is higher than that of consonants or the base sound. A prerequisite for the distinguishability of good speech is the correct hearing of vowels and consonants.

In room acoustics design, standard speech with a certain octave spectrum is used. Different standards provide slightly different spectra of speech. In addition, elevated and powerful speech spectra are provided for special situations. The most significant for this textbook is the spectrum of ISO 3382-3 in Fig. **4.10.1.** The standard presents two options for the SPL at 1 m in the free field:

- in front of the normally directional mouth, $L_{p,S,1m,normal}$, and
- omnidirectional in front of the mouth, $L_{p,S,1m,omni}$.

In both cases, the sound power level LW,S,A is the same. In the case of undirected speech, the SPLs are lower than in the case of directional speech, because behind the directional speaker, in turn, the levels are lower. Directional speech is used in speech modes when the direction of the speaker is known. Non-directional speech is used in open-plan offices and group teaching spaces when the direction of the speaker is unknown. Fig. **4.10.1** also shows the speech hearing threshold level ART. The values are 5–20 dB higher than the audiological sine sound hearing threshold levels in Chapter 1. At SPLs lower than ART, speech is no longer considered relevant to speech intelligibility. The significance of the octave band 125 Hz for speech intelligibility decreases already when speech is attenuated from the normal level by 5 dB. In addition, 8 kHz sounds are easily attenuated more strongly than other frequencies and have the lowest level. The most significant for speech separation are therefore octave bands 250–4000 Hz.

The directivity of the A-weighted voice is shown in Fig. **4.10.2.** Speech is about 10 dB quieter behind the speaker than in front. Directivity is strongest at high frequencies. At low frequencies, directivity is non-existent, since the mouth begins to resemble a point source and the head does not significantly overshadow the passage of sound. In room spaces, the importance of directivity of speech increases as reverberation becomes shorter. The more space echoes, the less influence the speaker's direction is perceived to have.

f [Hz]	$L_{p,S,1m,omni}$ [dB]	$L_{p,S,1m,normal}$ [dB]	$L_{W,S}$ [dB]	ART [dB]
125	49.9	51.2	60.9	46.0
250	54.3	57.2	65.3	27.0
500	58.0	59.8	69.0	12.0
1000	52.0	53.5	63.0	6.5
2000	44.8	48.8	55.8	7.5
4000	38.8	43.8	49.8	8.0
8000	33.5	38.6	44.5	12.0
A	57.4	59.5	68.4	30.4

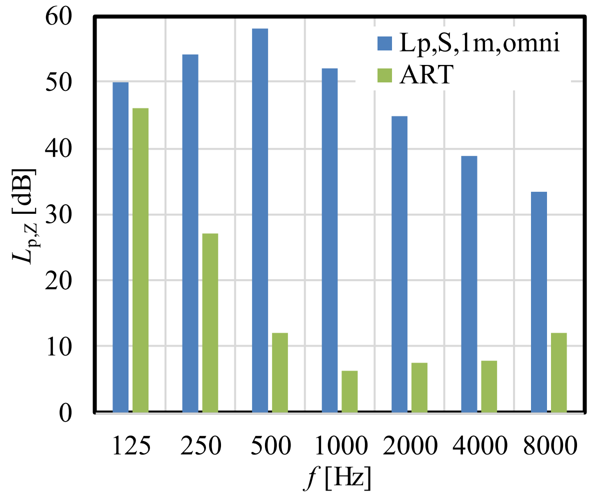


Fig. 4.10.1 Speech SPL according to ISO 3382-3:2012 for normal or omnidirectional speech. In addition, the sound power level $L_{W,S}$ corresponding to both situations and the absolute speech reception threshold (ART) according to IEC 60268-16:2011 are presented.

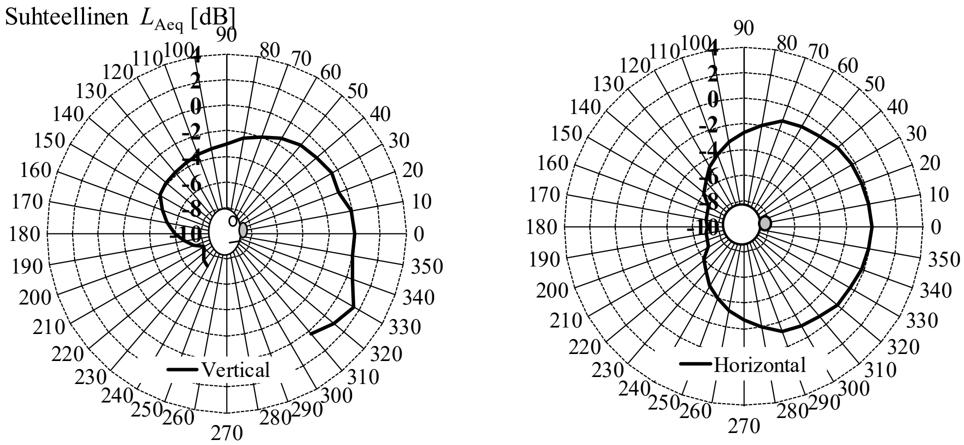


Fig. 4.10.2 Directivity of speech in the horizontal and vertical planes. The measurements have been taken in an anechoic chamber 1 m from the mouth. (Chu et al., 2001)

4.11. Speech transmission index STI

The subjective intelligibility of speech presented in the mother tongue in the room depends on several factors in a practical situation:

- the distance between speaker and listener,
- on the volume of the speaking voice,
- the direction of the speaker in relation to the listener,
- seeing the speaker's mouth and gestures,
- on the manner of speech and its familiarity,
- spatial reverberation,
- space geometry and sound reflectors,
- background noise level.

The subjective speech intelligibility between speaker and listener can be objectively assessed by measuring the *Speech Transmission Index* (*STI*). It is determined by acoustic instruments between speaker and listener according to IEC 60268-16. The numerical value of STI can vary between 0.00–1.00. The higher the *STI* value, the better the speech intelligibility. A value of 1.00 indicates complete byte separability. It can usually be achieved in rooms only in close proximity to the speaker. A value of 0.00 indicates that not a single syllable of speech can be understood. *STI* correlates quite linearly with subjective syllable separability.

The speech transmission index is used to describe the acoustic quality of spaces in teaching and speaking rooms and between open-plan workstations. Table **4.11.1** shows the subjective significance of the speech transmission index. In speech spaces, the aim is to achieve the highest possible value of the speech transmission index, i.e. good speech intelligibility. Complete speech intelligibility prevails when *STI>0.75.*

Between office workstations, meeting rooms or patient rooms, the aim is to achieve the lowest possible *STI* value in order to concentrate on work, have confidential conversations or sleep. A low *STI* is synonymous with speech *privacy*. If completely confidential discussions are to take place, the *STI* value should be 0.00 where other people are present.

Table 4.11.1 Subjective significance of the speech transmission index *STI* for speech intelligibility (communication spaces) and speech privacy (workspaces requiring concentration).

STI range	Speech intelligibility
<0.30	Useless
0.30-0.45	Poor
0.45-0.60	Reasonable
0.60-0.75	Good
>0.75	Very good

STI range	Speech privacy
0.00	Very good
0.00-0.10	Good
0.10-0.20	Reasonable
0.20-0.35	Poor
>0.35	Useless

Although the subjective intelligibility of speech depends on several factors, as described above, its objective counterpart, i.e. *STI,* can be computationally estimated with good accuracy using the method of Houtgast and Steeneken (1985), given the known reverberation time and speech-to-noise ratio in the octave bands of 125□8000 Hz. The method is suitable for diffuse sound fields and does not take into account the hearing threshold of speech or auditory coverage.

The signal-to-noise ratio of speech (speech-to-noise ratio) is defined by the equation

(4.11.1) $$L_{SN} = L_S - L_N$$

where L_S [dB] is the SPL of speech and L_N [dB] is the SPL of stationary background noise. The calculation of the STI begins by defining the modulation transfer function (MTF). MTF consists of a modulation reduction factor m(F,f) at 14 modulation frequencies F_i (0.63, 0.80, 1.00, 1.25, 1.60, 2.00, 2.50, 3.15, 4.00, 5.00, 6.30, 8.00, 10.00 and 12.5 Hz) and seven octave bands fj (125, 250, 500, 1000, 2000, 4000 and 8000 Hz)

(4.11.2) $$m(F,f) = \frac{1}{\sqrt{1+\left(T(f)\cdot 2\pi F/13.8\right)^2}} \cdot \frac{1}{1+10^{-L_{SN}(f)/10}}$$

where *T*(*f*) is reverberation time. The apparent signal-to-noise ratio is calculated for the 98 *m* values obtained above from the equation

(4.11.3) $$SN_{app} = 10\log_{10}\frac{m}{1-m}$$

If SN_{app}>15 dB, SN_{app}=15 dB is used. Similarly, if SN_{app}< -15 dB, SN_{app}= -15 dB is used. The STI is then calculated from the equation

(4.11.4) $$STI = \frac{1}{30}\left\{15 + \sum_{j=1}^{7} k_j \cdot \left(\frac{1}{14}\sum_{i=1}^{14} SN_{app}\left(F_i, f_j\right)\right)\right\}$$

where the weighting factors k_j of the octave bands 125–8000 Hz are 0.13, 0.14, 0.11, 0.12, 0.19, 0.17, and 0.14.

The dependence of STI on speech signal-to-noise ratio is illustrated in Fig. **4.11.1a.** Fig. **4.11.1b** shows the coarse dependence of STI on both reverberation time and signal-to-noise ratio. Both involve drastic assumptions, but with them it is already possible to roughly estimate the end result even with limited initial data.

The latest version of IEC 60268–16:2011 takes into account several factors compared to the simple model presented above. They have the greatest impact on the value of the STI when the speech level is low, the signal-to-noise ratio is low, or the speech/background noise spectra have an abnormal shape. To determine the STI, L_S is further measured using the appropriate speech power level and directivity at the speaker position and background noise L_N in the octave bands f_j. Instead of the early reverberation time, the modulation transfer function between the speaker and the listener is measured by means of the impulse response, taking into account the above. modulation frequencies $_{Fi}$ and octave bands f_j. In addition to background noise L_N, auditory speech masking (low frequencies mask higher ones) and speech hearing threshold (ART in Fig. **4.10.1**) are taken into account as background sounds masking speech. It is possible to calculate STI separately for male and female speech, which are subject to different octave bands weighting and redundancy factor. The coefficients α_j correspond to, but differ from, the kj values shown above. In addition, the female octave weighting factors lack the value of the octave band 125 Hz. An *STI* is recommended to be assigned to male speech, as it is usually worse distinguishable than female speech. The coefficients β_j take into account that if speech in the octave band j does not stand out (large echo and/or coverage), nonzero speech separation in adjacent octave bands j-1 or j+1 compensates for the lack of speech separation in band j to some extent.

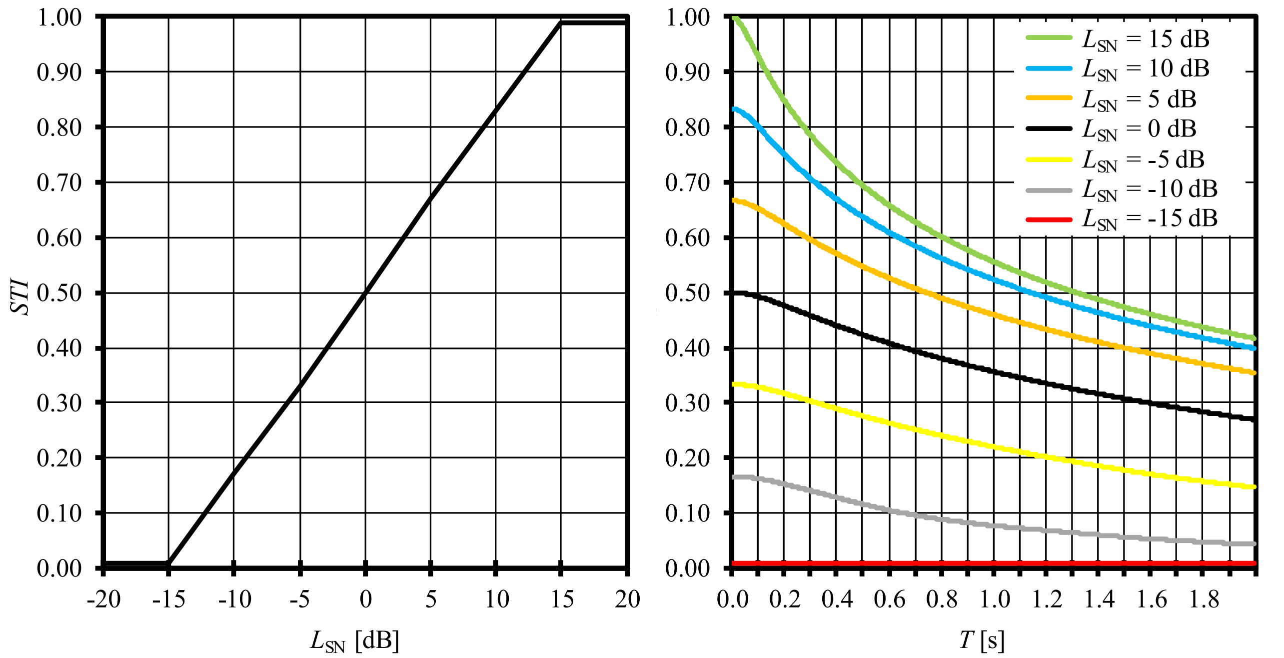


Fig. 4.11.1 (a) The dependence of the speech transmission index STI on the signal-to-noise ratio of speech assuming free field (T=0 s) and frequency-independent signal-to-noise ratio. In practice, speech peaks can still be heard when L_{SN}=-15 dB, but nothing is clear from it anymore. Speech is no longer heard when L_{SN}*<-22 dB.* **(b)** The dependence of the speech transmission index STI on reverberation time *T* and the signal-to-noise ratio of speech *LSN*, assuming that both are frequency independent in the octave bands of 125–8000 Hz.

4.12 Room acoustics in open-plan offices

An open-plan office refers to a single room space with several workstations without partitions separating them. There is no exact definition of an open-plan office, but premises with more than 5 people are generally considered open-plan offices because they are usually specially designed for this purpose. Workstations can be separated by a screen or cabinet. Cabinets and screens can reach up to 2.1 m, but usually not to the ceiling. However, light elements that absorb sound or create visual obstructions may hang from the ceiling.

A common problem in open-plan offices is insufficient speech privacy and distraction caused by unnecessary voices. In poorly executed open-plan offices, speech stands out clearly (STI>0.50) up to 20 meters away from the speaker. In a very well-executed open-plan office, STI=0.50 is already 2–3 meters away from the speaker and becomes completely unclear (STI=0.00) 6–8 meters from the speaker.

Room acoustic characterization is carried out according to ISO 3382-3, which is based on Finnish research (Virjonen et al., 2009). It determines the sound level and STI of the voice at different distances from the speaker on a measuring line extending over several workstations. Both voice production and measurement take place at workstations at a height of 1.20 m from the floor. The measurement is carried out according to the principle shown in Fig. 4.12.1.

The acoustic goodness of an open-plan office is determined by the A-weighted speech attenuation degree $D_{2,S}$ and the distraction distance r_D (Fig. 4.12.2). In an open-plan office, $D_{2,S}$ is determined from A-weighted speech levels by matching a straight line to the values of the distance interval 2–16 m. $L_{p,A,S,4m}$ [dB] is the A-weighted SPL 4 m away from the speaker. Distraction distance (rD) is the distance outside which the STI falls below 0.50. Conditions in open-plan offices vary widely: $D_{2,S}$ values have been measured between 2 and 14 dB, $L_{p,A,S,4m}$, values between 40 and 54 dB and r_D values between 2 and 16 m. However, most observations are in the middle of the ranges listed above.

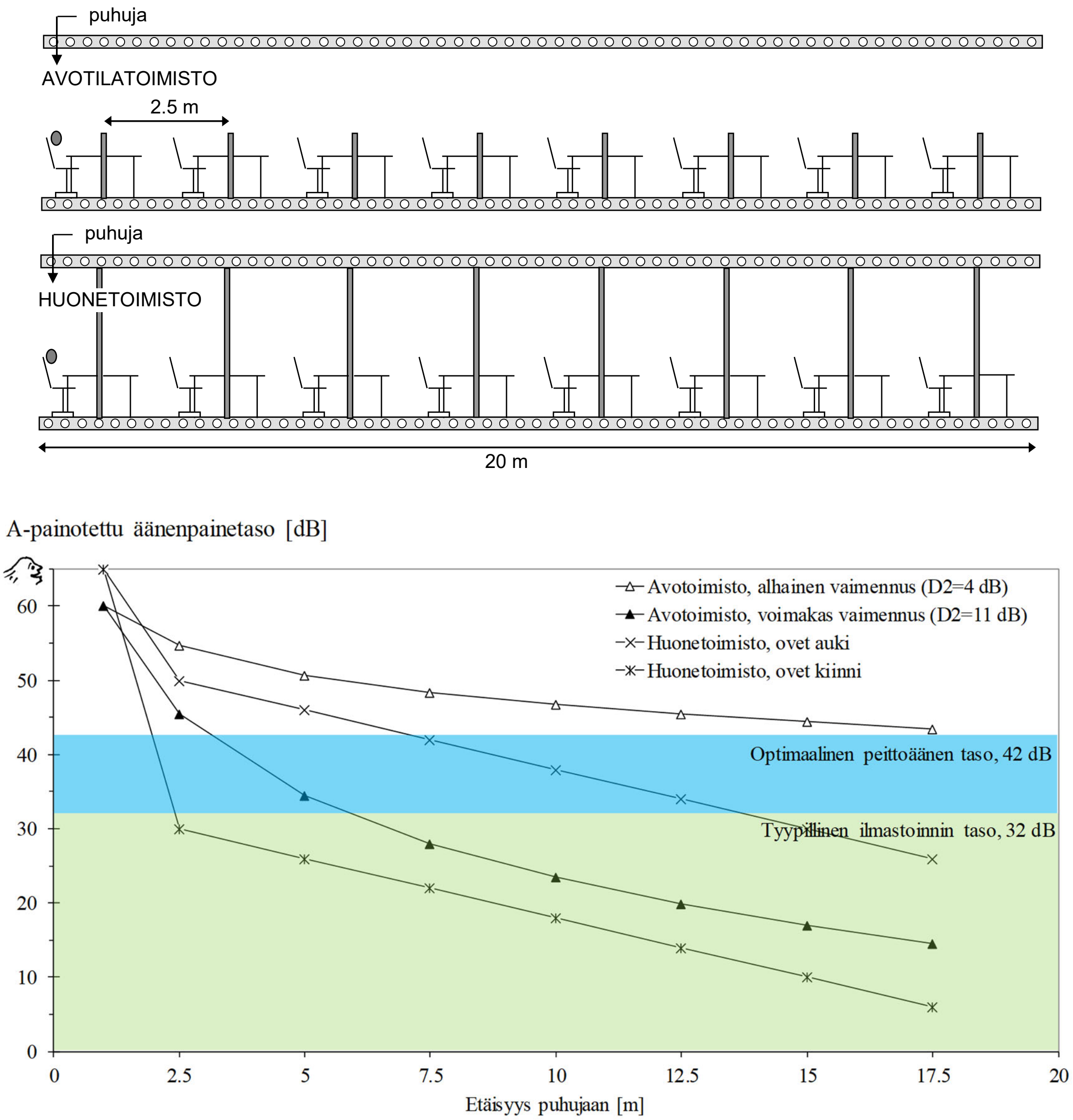


Fig. 4.12.1 A-weighted SPL of speech as a function of distance. The voice is produced from the workstation on the left. The open-plan office has no partitions and the room office does. 8 measuring points are located at workstations at distances 1.0, 2.5, 5.0, 7.5, 10.0, 12.5, 15.0 and 17.5 m.

According to numerous laboratory studies, a person's performance in tasks requiring working memory improves when the STI value of speech decreases (Haapakangas et al., 2020). According to studies conducted in open-plan offices, the proportion of people who find noise very disturbing decreases when the annoyance distance r_D decreases. For these reasons, the acoustics of open-plan offices have been regulated by building regulations since 2018. According to the Ministry of the Environment (2019), open-plan offices in Finland should be designed so that:

- STI < 0.50 more than 8 m from the speaker in an unfurnished space (more than 5 m from the speaker in a furnished space), and
- reverberation time is less than 0.60 seconds in the octave bands at 250–2000 Hz.

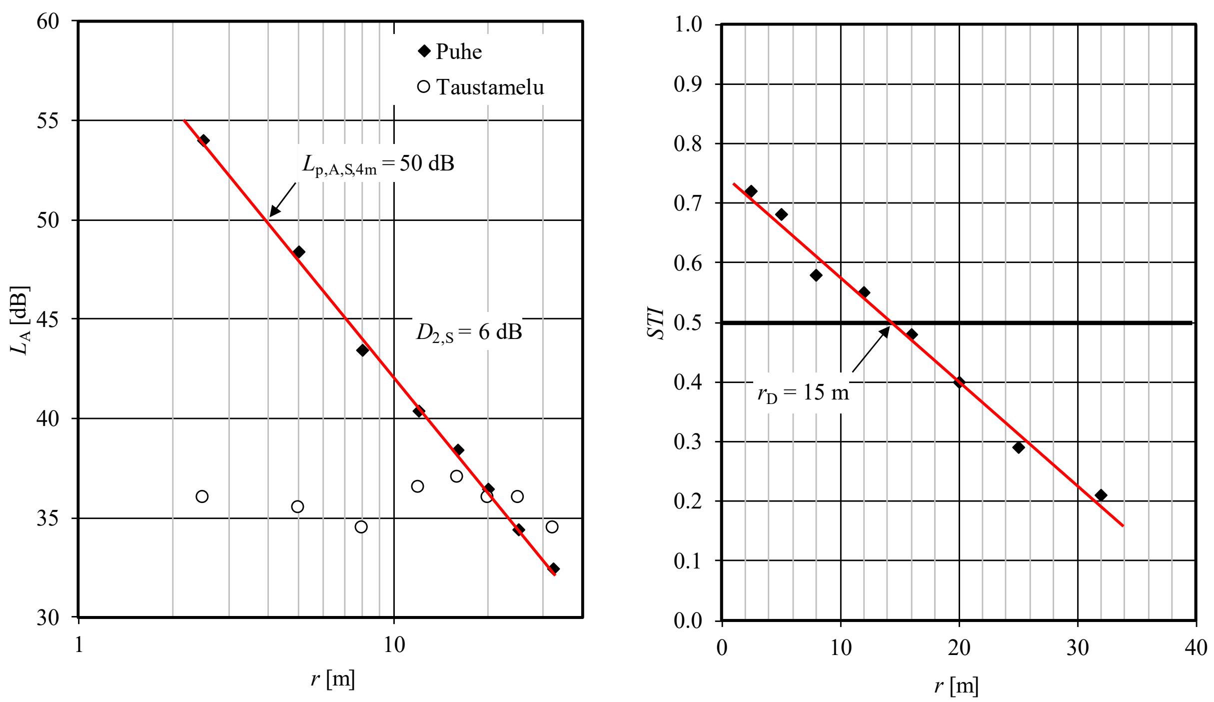
The guideline value for STI corresponds to the maximum disturbing distance of 8 m in unfurnished space and 5 m in furnished space (Ministry of the Environment, 2019). A high value of $D_{2,S}$ and a low value of r_D and a value of $L_{p,A,S,4m}$ are achieved when the following are achieved simultaneously:

1. absorption is maximized both on the ceiling of the room and on the wall surfaces;
2. between workstations there are screens as high as possible and sound absorbing;
3. The mode has an appropriate level of speech overlay.

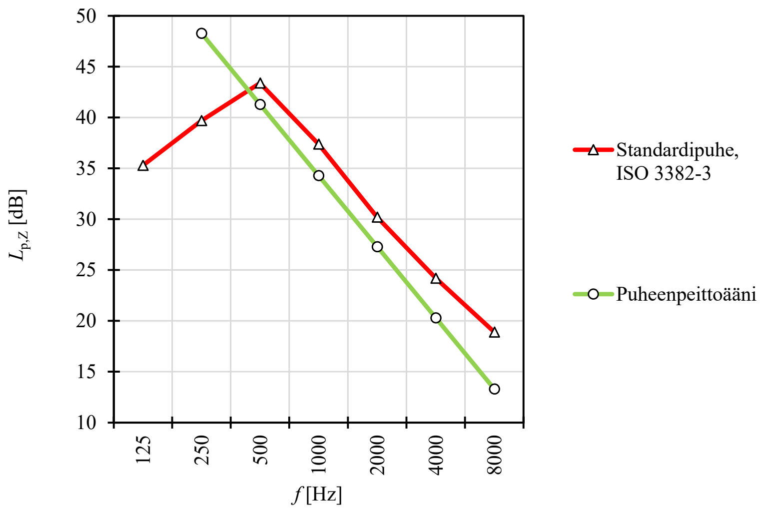
Broadband noise as shown in Fig. 4.12.3 is recommended for speech overlay. This spectrum has been arrived at in studies that have investigated different spectra from the perspective of irritation and speech masking. The volume is usually adjusted so that the sound level at workstations is 40-45 dB L_{Aeq}. Sound is usually produced from ceiling-mounted speakers, which are needed at the rate of about 1 piece per 10 m^2. The sound level should not vary by more than 3 dB between different workstations.

The publication of the Ministry of the Environment 2019:28 (Kylliäinen and Hongisto, 2019) and guideline RIL 243-3-2008 discuss planning solutions in detail.

Fig. **4.12.1** compares the sound level of speech acoustically in a good and bad open-plan office and in a room office where the corridor doors are open or closed. With low attenuation, speech is slowly attenuated in an open-plan office, and the background sound level of 32 dB cannot be reached even 20 meters away from the speaker. With good planning, the level of speech goes beyond the background only right near the speaker. In addition, if the background sound level is increased to 42 dB using an overlay system, speech will be lost in the background already 3–5 meters away from the speaker. If the doors of a room office are open and both the hallway and the rooms echo, the voice can stand out better in a room office than in a well-designed open-plan office.



4.12.2. (a) Speech sound level and (b) speech transmission index *STI* as a function of speaker-listener distance *r*. The spatial decay rate of speech, $D_{2,S}$, speech level at 4 m from the speaker $L_{p,A,S,4m}$ and distraction distance r_D are determined using linear fittings on logarithmic (a) or linear (b) distance axis.



4.12.3. Unweighted SPL, $L_{p,Z}$, as a function of frequency f by octave band for speech overlay and standard speech, both SPLs adjusted to 43 dB L_{Aeq}.

Computer models based on ray tracing can achieve moderately high accuracy of sound levels in an open-plan office. However, modelling is challenging due to the large amount of furniture. It is usually enough for an acoustic designer to know which parameter values of the ISO standard are achieved with a particular design solution. To quickly estimate ISO parameters, Keränen and Hongisto (2013) developed a spatial decay regression model based on measurements made in 16 acoustically different open-plan offices. The A-weighted SPL $L_{p,A,S,4m}$ [dB] of speech is given by:

(4.12.1) $$L_{p,A,S,4m} = L_{p,A,S,1m} - 3 \cdot h - 0.1 \cdot W - 4.6 \cdot \alpha_c - 0.8 \cdot \alpha_f$$

and the degree of speech attenuation from the equation

(4.12.2) $$D_{2,S} = 8 \cdot \frac{h}{H} + 0.16 \cdot \frac{L}{H} + 4 \cdot \alpha_c + 1.7 \cdot \alpha_f$$

where $L_{p,A,S,1m}$=57.4 dB (Fig. 4.10.1) is the SPL of undirected normal speech 1 m from the speaker in the free field, αc is the average absorption ratio of the ceiling, α_f *is the* apparent absorption ratio of furniture and/or vertical surfaces*,* W [m] is the room width, H [m] is the room height, L [m] is the room length and h [m] is the height of screens or cabinets. Vertical surfaces refer to wall surfaces, screens and cabinets. The average absorption ratio is determined as the average of the octave bands 250–4000 Hz. Based on the degree of attenuation of speech, the A-weighted sound level $L_{A,S}$ of speech at any distance r from the speaker can be calculated using the equation

(4.12.3) $$L_{p,A,S}(r) = L_{p,A,S,4m} - 3.3 \cdot D_{2,S}\left[\log_{10}(r) - \log_{10}(4)\right]$$

The regression model is only valid when sound is produced and measured at workstations at a height of 1.2 m from the floor. The model has been found to be fairly accurate, but the initial data does not include open-plan offices with absorbing screens and walls. These solutions are increasingly used nowadays and the model could be developed.

To calculate the distraction distance, the spread attenuation of the STI is required. To determine the STI*,* the signal-to-noise ratio of speech and reverberation time are required. Ao. The simplified method achieves sufficient accuracy in many design situations. The speech signal-to-noise ratio L_{SN} is the difference between the speech sound level $L_{A,S}$ and the background noise level $L_{A,B}$:

(4.12.4) $$L_{SN} = L_{A,S} - L_{A,B}$$

The signal-to-noise ratio of speech is determined at the same distances from the speaker as the sound level of speech. The background noise level is generally assumed to be the same at all points and is set to the assumed sound level of air conditioning or masking noise. The reverberation time of a room is calculated based on the absorption materials of room surfaces and furniture using Sabine's method. Like the background noise level, reverberation time is assumed to be the same at all points in the room. The STI is estimated from L_{SN} and T from the diagram in Fig. 4.11.1b at distances of 1, 2, 4, 8, 16 and 32 meters. The distraction distance can then be determined.

4.13 Auditorium design

Here, an auditorium refers to a space of approximately 30–200 people, where the speaker/presenter has a fairly fixed location and the audience as well. The grandstand can be located on a flat floor or it can be ascending. Room acoustics play a very important role in such spaces, as they do not want to use electrical sound amplification in principle.

In principle, the most important design factor is the control of background noise in any way. Ventilation sounds, as well as electrical aids such as video projectors and sound amplifiers, must be below 33 dB L_{Aeq} in both the speaker and audience areas. The sound insulation must be such that the levels of neighboring premises or sounds from outside do not exceed this value either.

Hearing registers sounds that arrive during 50 ms as the same sound, so reflections that reach the listener within less than 50 ms after direct sound are useful and should be amplified. Reflections that arrive after more than 50 ms of direct sound are harmful in speech acoustics because the fastest speech modulations (12.5 Hz, cycle duration 80 ms) are stalled. In the case of slow music, reflections up to 80ms after direct sound are useful. At room temperature, the sound travels within 50 ms within 17 m.

Therefore, the size of the space and surface treatment should be designed so that surfaces that produce first-degree reflections within less than 50 ms of direct sound are left hard. Other surfaces can be treated with sound-absorbing materials depending on the need for reverberation time.

A rising grandstand produces acoustically better results, as it guarantees that direct sound can be heard in the back row. Alternatively, the performer is located on the podium. Fig. **4.13.1** shows an example of how optimal speech acoustics can be achieved in a rising stand:

- A hard-surface roof curved above the speaker and onto the ceiling, where the sound reflected from the ceiling is directed equally to all parts of the auditorium (steps 1–5). However, this measure is not necessary.
- The front wall of the space is left hard-surfaced, as it is located close to the speaker and produces beneficial reflections;
- In the case of the audience, the sidewalls are left reflective at a height of up to 1.8 m, as useful reflections are generated from them;
- The necessary absorption material is placed on the back wall, in the public area, on the tops of the side walls, in that order. A reflective back wall can produce very harmful reflections on the front of the space, unless it is tilted in the direction of the ceiling. The seats are preferably sound-absorbing, because the audience area does not produce beneficial reflections and the reverberation of the space does not depend on the number of spectators.
- If additional placement of the absorption material is needed, start at the back of the roof and the periphery of the roof: an area of 1–2 meters on the periphery of the roof does not produce beneficial reflections.
- In the case of the performer, there should be no vibrating echo between the walls, so the walls should be lightly sound-absorbing or tilted in the direction of the audience.

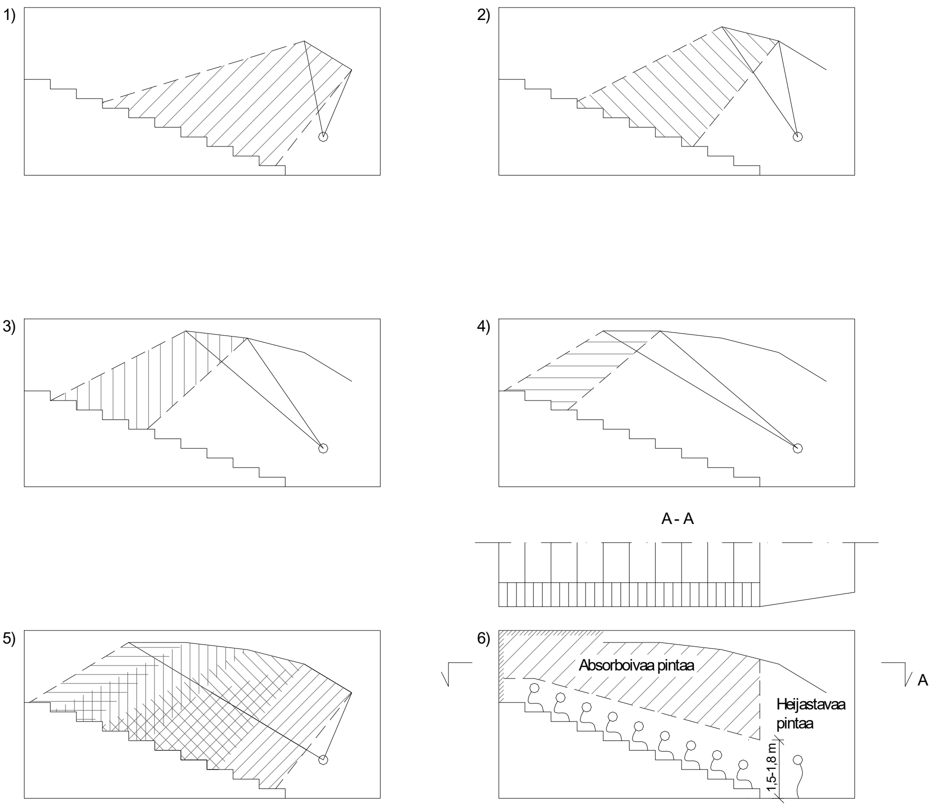


Fig. 4.13.1 Example of room acoustic design of an auditorium. (Photo: RIL 243-2-2007).

4.14. Spatial decay in industrial premises

In industrial premises, room acoustics are one of many ways to control noise exposure or other noise nuisance. The goal of the room acoustic design is to maximize the value of the spread attenuation degree DL_2. No national guideline values have been published for the spatial decay rate of industrial premises, but according to the Finnish Institute of Occupational Health, the target values should be DL_2=4–6 dB, depending on the furniture of the room. Very often, however, the DL_2 values are below 3 dB, which means that the spaces are perceived as noisy and problematic in terms of communication.

As a rule, industrial premises are more than 5 meters high. The bottom dimensions range from a few dozen to a few hundred meters. The furnishings are usually hard-surfaced, but the stored materials can absorb sound. Usually, room cushioning is handled by improving the absorptionness of ceiling and wall surfaces. The space may contain a lot of obstacles that affect the propagation of sound, such as objects and shelves. These factors cause the diffuse field equations for reverberation time and SPL propagation to not work in industrial premises. The purpose of room acoustic measures is to influence the noise level above all, and this effect is most easily determined by regression equations.

Keränen and Hongisto (2010) compared calculation models that can be used to calculate the spatial decay of SPLs in industrial premises. Models have been developed by Friberg, Embleton and Russel, Heerema, Hodgson, Kuttruff, Osipov, Sabine, Thompson, Wilson and Zetterling, among others. The models are based either on theoretical simplifications or on results derived from empirical data using statistical methods. Osipov et al. (1987) the model was able to achieve almost the same accuracy as the ray tracing method used in commercial census programs. According to the model, the distance attenuation DL [dB] at r [m] can be calculated by the equations

(4.14.1) $$DL(r) = 10\log_{10}\left\{\frac{1}{2\pi r^2} + \frac{(1-\alpha_{eff})(r+W)\mathbf{J}(\alpha_{eff},\rho)}{HW\,(r+H)}\right\}$$

(4.14.2) $$\mathbf{J}(\alpha_{eff},\rho)=\frac{0.1}{\alpha_{eff}+\rho^2 e^{0.65\rho}}$$

(4.14.3) $$\rho=-\frac{rS\ln(1-\alpha_{eff})}{4V}$$

where H [m] is the height, W [m] is the narrowest width, V [m^3] is the volume, and S [m^2] is the total area of the room surfaces. The average absorption ratio α_{eff} is the ratio of the absorption area to the total area of a room.

The spatial decay value is negative. The SPL at distance r from a sound source with a sound power level L_w is given by distance attenuation from the equation

(4.15.4) $$L_p(r)=L_w+DL(r)$$

Spread attenuation can be used to estimate the average noise level in the so-called " spatial decay". "in free space". This means that there are no obstacles between the sound source and the point being studied that strongly affect the passage of sound, such as screens.

If an industrial space is complex in shape, has a large number of sound sources to examine, and has significant obstacles affecting sound transmission, ray tracing modelling methods are recommended (Oliva, 2005; Hongisto et al., 2006). They can also be used to assess the effects of screens, absorption material placement options and layout changes.

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5 AIRBORNE SOUND INSULATION

Airborne sound insulation means how well two nearby rooms are isolated from each other. It depends on the direct sound insulation, i.e., sound reduction index, of the construction separating the rooms and flanking sound transmission via joints or sound leaks. This chapter only considers the former transmission paths.

5.1 Determination of sound reduction index

The definition of sound reduction index R [dB] is

(5.1.1) $$R = 10 \cdot \log_{10}\left(\frac{1}{\tau}\right) = 10 \cdot \log_{10}\left(\frac{W_1}{W_2}\right)$$

where is the τ transmission coefficient, and W_1 [W] is the sound power that hits the structure and W_2 is the sound power that penetrates the structure. Sound reduction index depends on the frequency.

I. PRESSURE METHOD

Sound reduction index is usually determined in the laboratory by a so-called pressure method, in which the structure is placed in a test hole between two diffuse rooms so that sound is transmitted between the rooms only through the structure. The test sound required for measurement is produced in room 1 by speakers. The sound power hitting the structure from the sound field of room 1 can be determined from the average effective sound pressure p_1 [Pa] in the transmission room according to Chapter **3** by the equation

(5.1.2) $$W_1 = I_1 S = \frac{\tilde{p}_1^2 S}{4 \rho_0 c_0}$$

where S [m2] is the surface area of the structure, I_1 [W/m2] is the intensity hitting the surface, ρ_0 [kg/m^3] is the density of air, and c_0 [m/s] is the speed of sound in air.

The penetrating sound power is determined by the sound pressure in room 2. In equilibrium situations, the sound power W2 radiated from the wall to the reception room and the sound power absorbed by the room surfaces are the same

(5.1.3) $$W_2 = \frac{\tilde{p}_2^2 A_2}{4 \rho_0 c_0}$$

where p_2 is the effective sound pressure in the receiving room and A_2 [m^2] is the absorption area of the reception room. Finally, we obtain

(5.1.4) $$R = 10\log_{10}\frac{W_1}{W_2} = 10\log_{10}\frac{\tilde{p}_1^2 S}{\tilde{p}_2^2 A_2}$$

For laboratory measurements (ISO 10140-5) and practical calculations, a form derived from it is used

(5.1.5) $$R = L_{p,1} - L_{p,2} + 10\log_{10}\frac{S}{A_2}$$

For a small building element (element: e.g. ventilation hole, mail hatch, inspection hatch), a quantity independent of the area, the so-called normalized unit sound level difference $D_{n,e}$, is determined from the equation

(5.1.6) $$D_{n,e} = L_{p,1} - L_{p,2} + 10\log_{10}\frac{A_0}{A_2}$$

where the normalized area is A_0=10 m^2. The equation gives the same result as equation **5.1.5** if we assume that the area of a small building element is 10 m^2, although of course this is not the case. A normalized unit sound level difference is used in order to directly compare the performance of a small building element with the value of the sound reduction index of the structure in which the small building element will be placed.

II. INTENSITY METHOD

The pressure method is based on the diffuse field assumption. Therefore, the method overestimates results at low frequencies, usually below 200 Hz. In addition, there are large differences between laboratories at low frequencies. In the intensity method, the sound emitted by a structure is measured using sound intensity, giving sound reduction index by the equation

$$R = 10\log_{10}\left(W_1\right) - 10\log_{10}\left(W_2\right) = 10\log_{10}\left(\frac{\tilde{p}_1^{\,2}}{4\rho_0 c_0 S}\right) - 10\log_{10}\left(I_2 S\right) \tag{5.1.7}$$

For laboratory measurements (ISO 15186-1), the format is used

$$R = L_{p1} - 6 - \left[L_{I2} + 10\cdot\log_{10}\left(\frac{S_{m2}}{S}\right)\right] \tag{5.1.8}$$

where L_{p1} [dB re 20 Pa] is the SPL in the transmitting µ room and L_{I2} [dB re 1 pW] is the intensity level radiated by the structure to the receiving room, S_{m2} [m^2] is the area of the measuring surface and S [m^2] is the area of the structure. The logarithm term is usually zero. At low frequencies of 50–200 Hz, the differences between laboratories are smaller if the SPL in room 1 is measured less than 20 mm from the surface of the structure, L_{p1S}, and the back wall of room 2 opposite the structure to be measured is sound-absorbing. For laboratory measurements (ISO 15186-3), the equation is used

$$R = L_{p1S} - 9 - \left[L_{I2} + 10\cdot\log_{10}\left(\frac{S_{m,2}}{S}\right)\right] \tag{5.1.9}$$

where L_{p1S} [dB] is the SPL on the surface of the structure in the transmission room.

The intensity method gives results closer to theory. It can also be used to measure reliable values of sound reduction index R in field conditions. The intensity method can also be used to construct a picture of sound radiation if the intensity measurement is made pointwise (Fig. **5.1.1**).

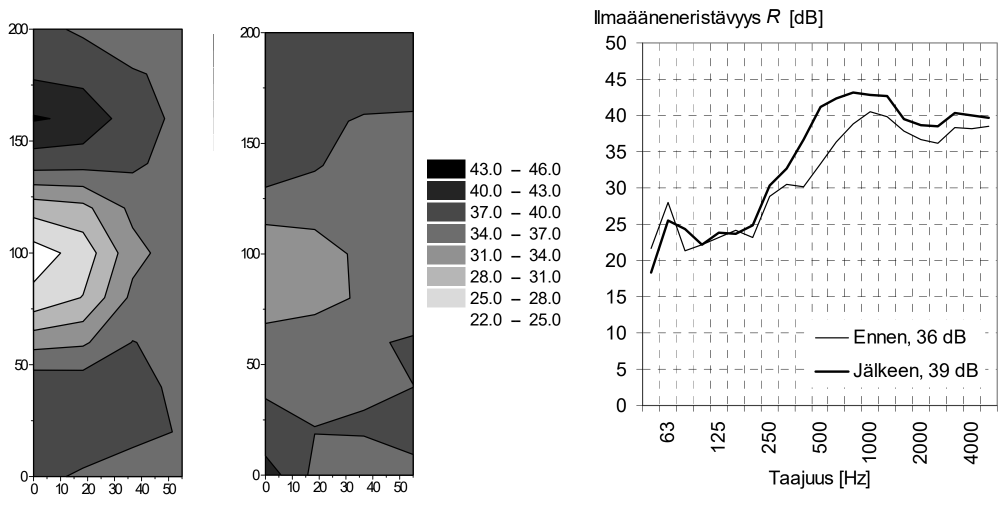


Fig. 5.1.1 Sound reduction index of a door structure before and after door lock sealing at 500 Hz. The colours roughly indicate the local sound reduction index [dB]. On the right, airborne sound insulation is shown. Before: unsealed lock. After: sealed lock. The difference in *rw* values was 3 dB.

5.2. Single-number quantities

Airborne sound insulation is usually determined at frequencies 100–3150 Hz or 50–5000 Hz, less often outside the latter frequency band. In business and regulations, airborne sound insulation is described by single-digit values. Definitions of single-number quantities are given in ISO 717-1. The most important of them is the weighted sound reduction index R_w. It is used to describe the airborne sound insulation of structures inside the building. For small elements, the normalized weighted sound reduction index $D_{n,e,w}$ is determined. The standard also presents various spectral weighting terms, of which there are 8 pieces: $C_{100\text{-}3150}$, $C_{100\text{-}5000}$, $C_{50\text{-}3150}$, $C_{50\text{-}5000}$, $C_{tr,100\text{-}3150}$, $C_{tr,100\text{-}5000}$, $C_{tr,50\text{-}3150}$, and $C_{tr,500\text{-}5000}$. The most commonly used term in Finland is *Ctr*, which is a spectral correction term for road traffic noise. In Finland, the single-digit value R_w+C_{tr} describes the airborne sound insulation of façade structures against road traffic noise.

Rw is determined using the so-called reference curve method. The method of determination is explained in Fig. **5.2.1.** The value of the spectral weighting term Cj is calculated from the equation

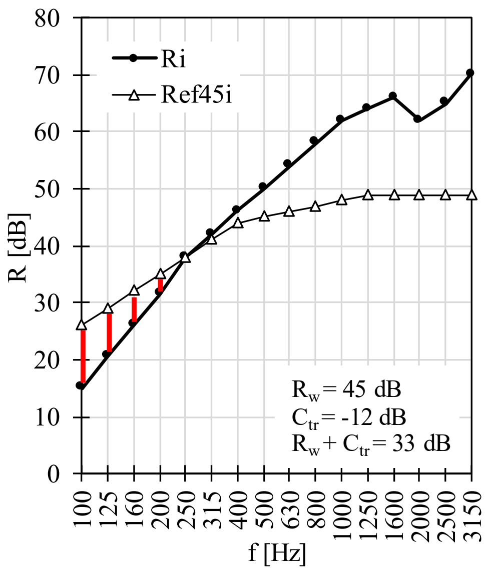
$$C_j = X_{Aj} - X_W \tag{5.2.1}$$

where j is the reference spectrum for the relevant one (either 1:A-weighted pink noise or 2:A-weighted urban traffic, tr), X_w is the single-number value determined above by the reference curve method (R_w or $D_{n,e,w}$)

$$X_{Aji} = -10\log_{10}\sum 10^{(L_{ij}-X_i)/10} \tag{5.2.2}$$

where i is the frequency band index, L_{ij} is the value of the reference spectrum j frequency band i, and Xi is the measured sound reduction index *(R* or $D_{n,e}$).

f_i	R_i [dB]	Ref_i [dB]	$Ref45_i$ [dB]	Dev_i [dB]	L_{i2} [dB]	$L_{i2} - R_i$ [dB]	$10^{(Li2-Ri)/10}$
100	15.0	*R* w-19	26	**11.0**	-20	-35.0	0.0003162
125	20.5	*R* w-16	29	**8.5**	-20	-40.5	0.0000891
160	26.0	*R* w-13	32	**6.0**	-18	-44.0	0.0000398
200	31.5	*R* w-10	35	**3.5**	-16	-47.5	0.0000178
250	38.0	*R* w-7	38	0.0	-15	-53.0	0.0000050
315	42.0	*R* w-4	41	0.0	-14	-56.0	0.0000025
400	46.0	*R* w-1	44	0.0	-13	-59.0	0.0000013
500	50.0	***R* w**	**45**	0.0	-12	-62.0	0.0000006
630	54.0	*R* w+1	46	0.0	-11	-65.0	0.0000003
800	58.0	*R* w+2	47	0.0	-9	-67.0	0.0000002
1000	62.0	*R* w+3	48	0.0	-8	-70.0	0.0000001
1250	64.0	*R* w+4	49	0.0	-9	-73.0	0.0000001
1600	66.0	*R* w+4	49	0.0	-10	-76.0	0.0000000
2000	62.0	*R* w+4	49	0.0	-11	-73.0	0.0000001
2500	65.0	*R* w+4	49	0.0	-13	-78.0	0.0000000
3150	70.0	*R* w+4	49	0.0	-15	-85.0	0.0000000



R: Ilmaääneneristävyyden mittaustulos
Ref: Vertailukäyrän muoto
Ref45: Vertailukäyrä asennossa 45 dB
Dev: Ei-toivottu poikkeama: =Max(0; $Ref45_i$ - R_i)

$Sum = \Sigma[10^{(Li2-Ri)/10}]$ = 0.000473
$X_{A2} = -10 \cdot \log_{10}(Sum)$ = 33.3
$C_{tr} = X_{A2}$ - R_w = -11.7
C_{tr} = **-12**

L_{i2} on C_{tr}:n laskennassa käytettävä referenssispektri.

Ei-toivottujen poikkeamien Dev_i summa:
29.0 dB Korkein sallittu 32.0 dB.

Fig. 5.2.1 The weighted sound reduction index R_w is determined from the measured *R* values using the reference curve procedure. The shape Ref of the reference curve is always the same, but its position in the y-direction depends on what output value is given for the anchor frequency, which is 500 Hz. The anchor frequency is given the highest possible value so that the sum of unwanted deviations does not exceed 32.0 dB. An undesirable deviation occurs when the value of the reference curve is above the measurement result. With a guess of 45 dB, the reference curve falls on position Ref45. Unwanted deviations occur at frequencies 100–200 Hz. Since their sum is less than 32.0 dB, R_w is equal to the value of the reference curve at the anchor frequency.

5.3 Types of vibrations and sound radiation in the panel

There are four types of waves propagating in panel structures: bending wave, shear wave, quasilongitudinal wave, and Rayleigh wave (Fig. **5.3.1**). The phase speeds of sound for bending, shear and longitudinal waveforms in a panel can be calculated from the following equations

(5.3.1) $$c_B = \sqrt[4]{\frac{\omega^2 B}{m'}} = \sqrt[4]{\frac{\omega^2 h^2 E}{\rho_m 12\left(1-\mu^2\right)}}$$

(5.3.2) $$c_s = \sqrt{\frac{Gh}{m'}} = \sqrt{\frac{E}{\rho_m 2\left(1+\mu\right)}}$$

(5.3.3) $$c_L = \sqrt{\frac{E}{\rho_m \left(1-\mu^2\right)}}$$

where ω [Hz] is the angular frequency, E [Pa] is the modulus of elasticity, G [Pa] is the shear modulus, h [m] is the thickness of the panel, B [Nm] is the bending stiffness per unit width, m' [kg/m^2] is the surface mass, μ [] is the Poisson ratio, and ρ [kg/m^3] is the density of the panel. On the panel, the bending rigidity is of the form

(5.5.4) $$B = \frac{Eh^3}{12\left(1-\mu^2\right)}$$

The Poisson ratio is $\mu \approx 0.30$ for metals and $\mu \approx 0.20$ for all other materials (Craik 1996).

With conventional building materials, shear waves do not have a significant effect on sound radiation or sound reduction index at frequencies below 5 kHz. The speed of the Rayleigh wave is about the same as that of the shear wave. This waveform is not considered because it only occurs on very thick panels on its surface (free surface waves). Significant waveforms for the sound radiation of a panel are bending wave and shear wave, since significant transverse displacement occurs in them and thus interaction with the surrounding air. The longitudinal wave motion of the panel is irrelevant for sound radiation. There are no restrictions on the surfaces of the panel, in which case longitudinal movement also causes a slight transverse movement due to Poisson shrinkage. In this case, we are talking about pseudolongitudinal waves. When looking at frequencies below 5000 Hz, only bending waves matter on thin panels. With thicker sheets, cutting waves also play a role in terms of sound reduction index.

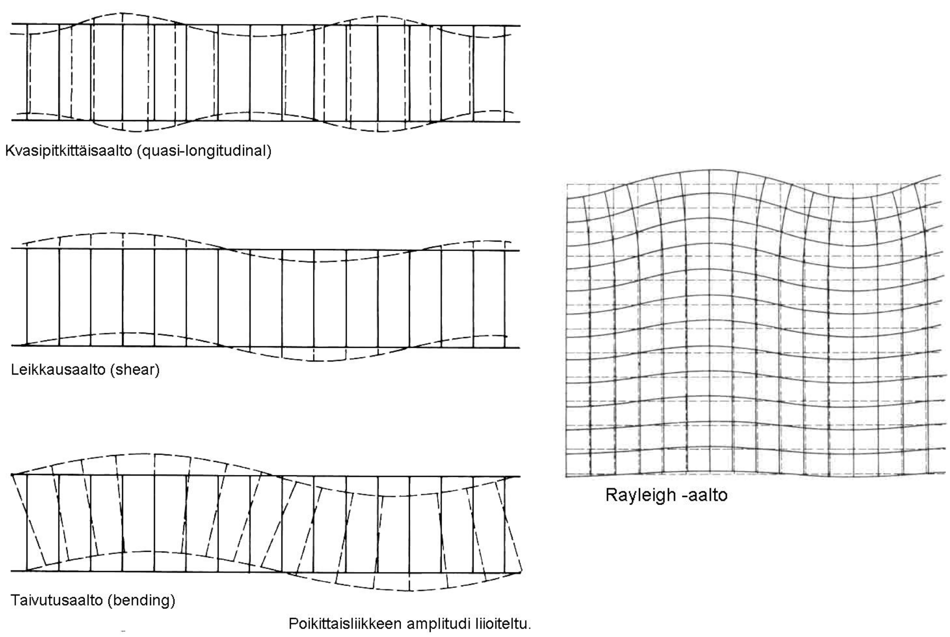


Fig. 5.3.1 The left shows the significant waveforms propagating in the panel (Source: Fahy 1985). On the right is an image of Rayleigh surface waves with the material surface at the top. (Source: Cremer and Heckl, 1988).

5.4 Coincidence

The speed of the bending wave increases with increasing frequency. With other waveforms, there is no frequency dependence. The speed of sound in the air is also constant and frequency independent. Since the bending wave is the most significant waveform in terms of sound radiation but dispersive, sound reduction index is also a frequency-dependent and therefore complex phenomenon.

When the speed of the bending wave reaches the speed of sound in air, a coincidence occurs at an angle of incidence of sound 90°. This frequency is called the limit frequency of coincidence (or critical frequency of cooccurrence) and is given by the equation

(5.4.1) $$\boxed{f_c = \frac{c_0^2}{2\pi}\sqrt{\frac{12(1-\mu^2)m'}{Eh^3}}}$$

The phenomenon of residence is illustrated in Fig. **5.4.1.** In cooccurrence, the sound wave arriving at a certain angle of incidence is perfectly coupled with the panel performing the bending wave and passes through the panel almost without attenuation, and a deterioration in the airborne soundproofing curve is observed at this frequency. However, the phenomenon is extremely narrowband. In practice, there are losses on the panel, due to which the throughput is not quite perfect.

The lowest coin frequency occurs at the angle of incidence $\theta=90°$ (*grazing incidence*). As the angle of incidence decreases, the coincidence frequency occurs by the frequency, which is the limit frequency divided by the sine of the angle of incidence. Thus, with a perpendicular angle of incidence of sound, the coin frequency is infinite.

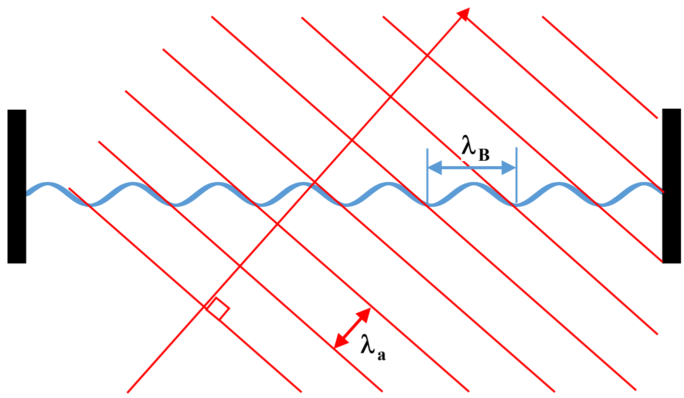


Fig. 5.4.1 Object effect illustrated by sound incidence θangle =45 °.

5.5 Specific frequencies

A finite panel rigidly supported at the edges has characteristic frequencies, i.e. modes. Specific frequencies are determined by the equation

(5.5.1) $$f_{mn} = \frac{c_0^2}{4f_c}\left[\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2\right] \qquad m,n = 0,1,2,3,...$$

where c_0 [m/s] is the speed of sound in air, f_c [Hz] is the limit frequency of cooccurrence, L_x [m] is the panel width, and L_y is the panel height [m]. The lowest axial modes f_{01} and f_{10} occur when the second measure of the panel is equal to half the wavelength λ_b of the bending wave propagating in the panel . The lowest specific frequency in direction x (f_{10}) is shown in Fig. **5.5.1.**

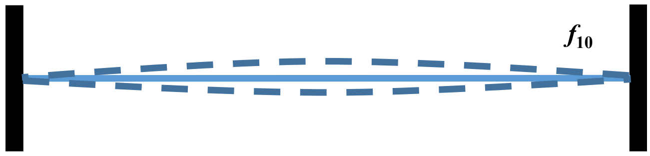


Fig. 5.5.1 The lowest specific frequency *f10* at which the panel oscillates as a whole with respect to its anchorages in the longest direction of the panel.

5.6. Sound emission from a bending wave

Above the limit frequency of coexistence, the bending wavelength λ_m is significantly longer than the wavelength λ_a of airborne sound, allowing the panel to radiate the energy of the bending waves efficiently over its entire surface area. The bending oscillation field of the panel does not emit air sound over its entire surface area below the limit frequency of cooccurrence. An acoustic short circuit occurs in the central parts **of the panel (Fig. 5.6.1-2**) because the wavelength λ_a of the airborne sound wave is longer than the wavelength λ_m of the bending wave of the panel ($c_a > c_m$). The positive and negative sound pressure emitted by the bending wave is cancelled out by a short circuit in the air.

The bending wave can effectively radiate sound below the coin frequency only at the characteristic frequencies of the panel. In this case, only the edges or corners emit sound, depending on the type of mode. Sound is emitted from an area only about a quarter of a wavelength wide in the peripheral areas, and an acoustic short circuit occurs further in.

For example, in statistical energy analysis (SEA), radiation efficiency σ (radiation coefficient, radiation ratio, radiation efficiency*)* is used to describe the radiation characteristic of the panel described above:

(5.6.1)
$$\sigma = \frac{W}{\langle v^2 \rangle \rho_0 c_0 S}$$

where W [W] is the sound power emitted by the panel, v [m/s] is the average vibration rate of the panel, and S [m2] is the surface area of the panel. Radiation efficiency can obtain values between 0.00 and 1.00. Maximum radiation efficiency is achieved at and above the limit frequency of coexistence. At lower frequencies, depending on the mode and area, the radiation efficiency is between 0 and 1 and very difficult to calculate.

Above the limit frequency of coexistence, the sound power emitted by the panel can be reliably estimated by measuring the vibration rate of the panel. For example, in concrete structures, the limit frequency of coincidence is approximately 100 Hz, which is why the sound power level emitted by the structure can be determined in the frequency range 100–3150 Hz by vibration measurements. The sound power emitted below the coincidence should be determined using the intensity method. It will have to be used in the case of light partitions in the entire frequency range of 100–3150 Hz.

Although the radiation efficiency of the bending wave is poor when $f < 1/2f_c$, the sound reduction index of the panel is not infinite. Sound radiation consists of resonant and non-resonating vibrations. The latter is also called forced vibration. Non-resonant vibration determines sound reduction index almost exclusively when $f < 1/2f_c$. When $f > 1/2f_c$, non-resonating and resonant vibrations together determine sound reduction index.

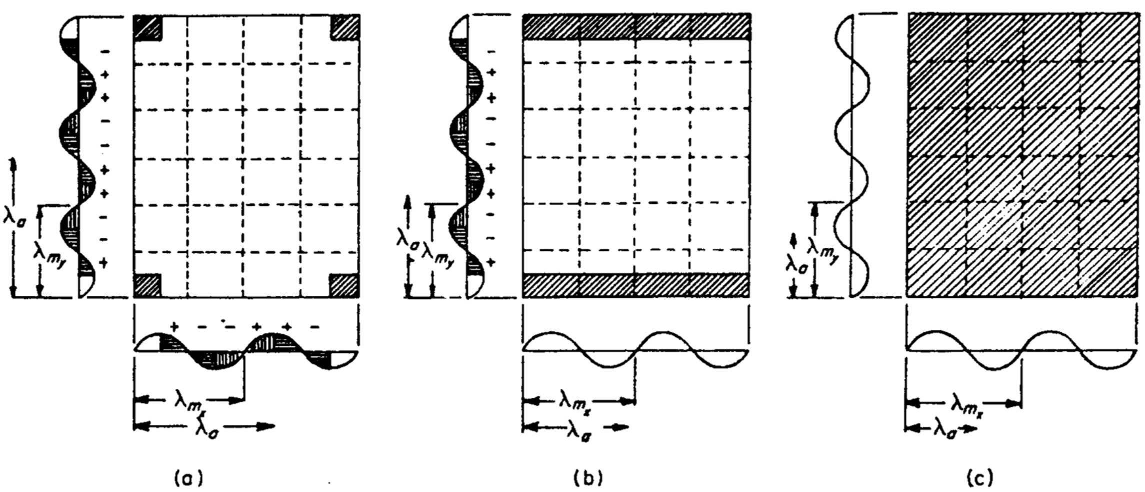


Fig. 5.6.1 Patterns of occurrence of resonant oscillation from the sides in a rigidly supported panel in different frequency ranges. A gray area indicates sound radiation from a resonating field. The dashed line on the panel denotes half the bending wavelength in the panel. (a) Corner mode (f<<). A short circuit occurs anywhere except in corners. (b) Edge mode ($f << fc$). A short circuit occurs not only in the direction of the mode. (c) Surface mode ($f \geq f_c$). There is no short circuit, but the entire panel radiates efficiently. The wavelength in air is λ_a, and the bending wave length in the panel is λ_m. (Source: Fahy 1985)

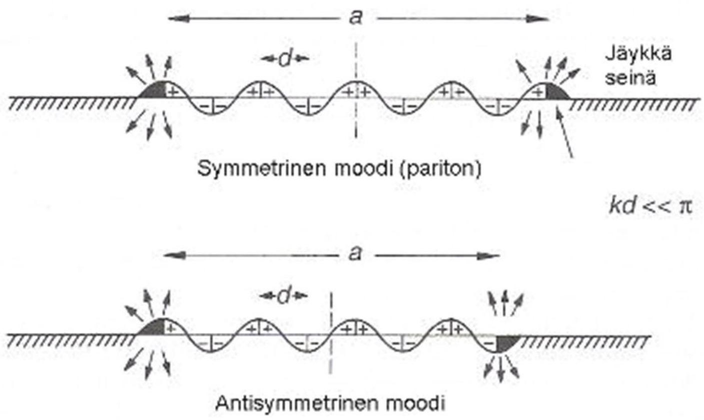


Fig. 5.6.2 Acoustic short circuit in the case of edge mode at $f<1/2f_c$. Fig. a shows the measurement of the panel in the direction of the axis under consideration. Due to a short circuit, there is turbulence in the near field of the panel and therefore intensity measurements, for example, are not recommended to be carried out less than 10 cm from the surface (Source: Fahy 1985).

5.7. Loss factor

The structure loss factor η indicates how quickly the resonant oscillation, i.e. the mode field of bending waves, is attenuated. The loss factor indicates how much energy is lost in one cycle. The oscillatory energy E is attenuated in the structure by time t as a function of the equation

(5.7.1) $$E(t) = E_0 e^{-\eta_{tot}\omega t}$$

where E_0 is the maximum energy generated by the excitation into the structure. The total loss factor is the sum of the internal loss factor, coupling loss factor and radiation loss factor (Fig. **5.7.1**)

(5.7.2) $$\eta_{tot} = \eta_{sis} + \eta_{sät} + \eta_{kyt} = \eta_{sis} + \frac{Z_0\sigma}{\pi f m'} + \frac{c_0 \sum_k l_k \alpha_k}{\pi^2 S \sqrt{f f_c}}$$

where σ is the radiation efficiency of the material, Z_0 [Pa·s/m] is the characteristic impedance of air, l_k [m] is the length of the joint k, α_k is the absorption coefficient of the joint k, and S [m^2] is the total area of the structure. Typically, a floor or wall structure has 4 joints (k: 1... 4).

The total loss coefficient of the structure at a particular installation site can be determined experimentally by the equation

(5.7.3) $$\eta_{kok} = \frac{2.2}{fT}$$

where T [s] is the reverberation time of the structure and f [Hz] is the frequency. The measurement result obtained for the structure does not apply to another installation site, where the joints and dimensions are different.

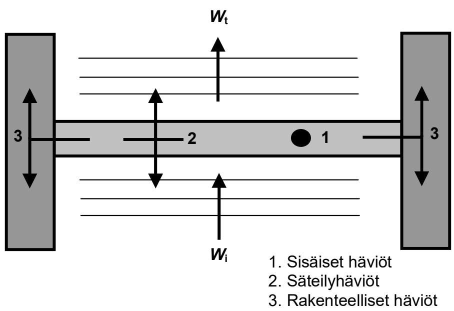


Fig. 5.7.1 Three factors of the total loss factor. Light grey is the structure to be examined (e.g. wall) and dark grey is the structure to which the structure under study is attached (e.g. tangent walls and connection to the building).

The internal loss coefficients of most materials are extremely small. It is not worth using them in calculations, since when installed in the structure, however, the internal loss factor is only a fraction of the total loss factor. Internal losses are significant, for example, in rubber or lead.

The total loss factor is strongly frequency-dependent. For most structures, the frequency dependence of the total loss factor is of the form (Troclet, 2000)

$$\eta(f) = Af^B \tag{5.7.4}$$

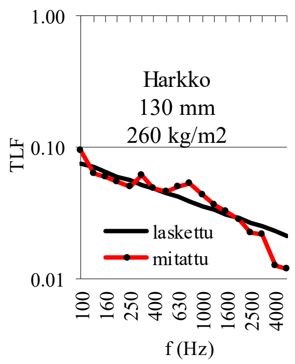
where A and B are constants. Table **5.7.1** shows the values of some structures.

Fig. **5.7.2** shows a study investigating how the flexibility of block wall sides affects the overall loss coefficient and sound reduction index. The flexible installation produced a lower R-value, since the vibrational energy cannot propagate along the joints to the building frame, but remains completely in the structure, increasing the vibration amplitude of the structure and thus the radiation efficiency of air sound. If the sound reduction index of the structure with installation method A is R_A, and total loss factor $\eta_{tot,A}$, the airborne sound insulation R_B for installation method B and the total loss coefficient $\eta_{tot,B}$ may be calculated from the equation

$$R_B = R_A + 10\log_{10}\left(\frac{\eta_{tot,B}}{\eta_{tot,A}}\right) \tag{5.7.5}$$

Table 5.7.1 Constants **A** and *B* of the frequency-dependent equation 5.7.4 of the total loss factor TLF. The values have been determined by measuring the structural reverberation time and determining the loss factor using equation **5.7.3.** On the right, an example of fitting equation **5.7.4** to the measurement results of an ingot wall.

Materiaali	Asennustapa	Mitat [m]	A	B
Calsium-silikaatti harkko 130 mm, 260 kg/m2	Muurattu seinä	3.6 x 2.8	0.33	-0.32
Calsium-silikaatti tiili 200 mm, 360 kg/m2	Muurattu seinä	3.6 x 2.8	1.09	-0.52
Lecaharkko 150 mm	Muurattu seinä	5.0 x 4.0	0.35	-0.40
Teräsbetoni 180 mm	Muurattu seinä	3.0 x 6.0	0.50	-0.42
Teräsbetoni 160 mm	Laatta teräspalkkikehän päällä	4.1 x 2.5	0.12	-0.30
Kipsi EK 13 mm	Ruuvattu k600	1.2 x 2.2	0.05	-0.10
Kipsi N 13 mm	Ruuvattu k600	1.2 x 2.2	0.04	-0.08
Teräslevy 2 mm	Ruuvattu k1200	1.2 x 2.2	1.66	-0.72
Teräslevy 4 mm	Ruuvattu k1200	1.2 x 2.2	0.07	-0.25



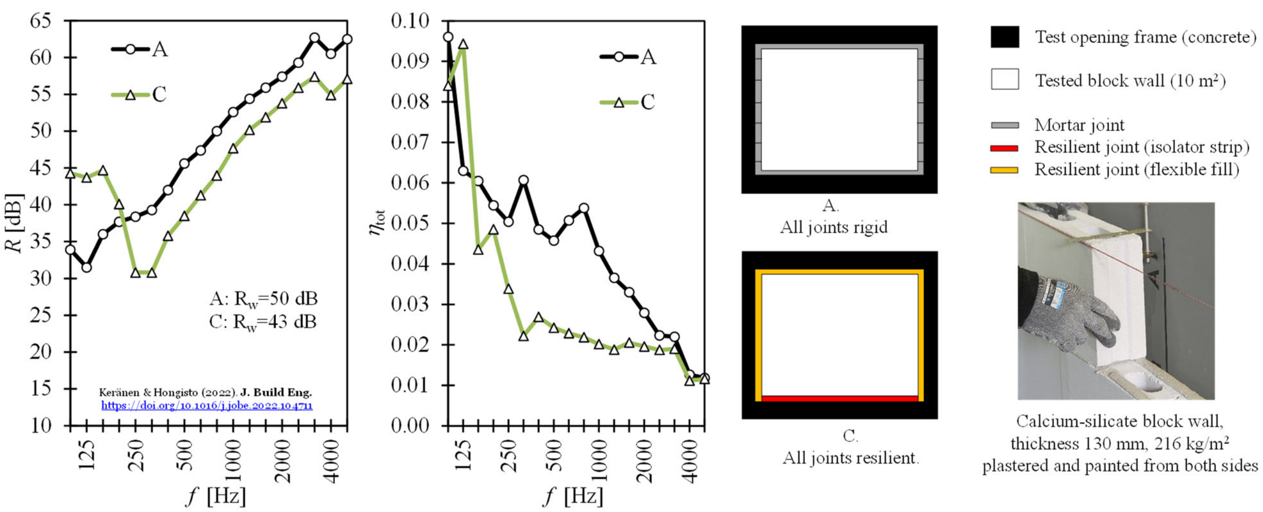


Fig. 5.7.2 Sound reduction index of the block wall, R, and total loss factor, η_{tot}, as a function of frequency, *f*, when the joints of the block wall are rigid (A) or elastic (B). The rigid installation method is the most common (Keränen &; Hongisto, 2022).

5.8 Single panel

If the sound field in the transmission room is diffuse and the surface area of the single panel is assumed to be infinite, the sound reduction index R [dB] can be estimated with good accuracy by a pair of equations (Rindel, 2018)

$$R = \begin{cases} 20\cdot\log_{10}\left(\dfrac{\pi m' f}{\rho_0 c_0}\right) + 20\cdot\log_{10}\left(1-\left(\dfrac{f}{f_c}\right)^2\right) - 5, & f < f_c \\ 20\cdot\log_{10}\left(\dfrac{\pi m' f}{\rho_0 c_0}\right) + 10\cdot\log_{10}\left(\dfrac{2\eta f}{\pi f_c}\right), & f \geq f_c \end{cases} \tag{5.8.1}$$

where m' [kg/m^2] is the surface mass, f is the frequency [Hz] and is η the total loss factor. The upper equation is the so-called. the law of mass and applies to the frequency range of forced oscillation, that is, non-resonant oscillation. According to the mass law, R increases by 6 dB with doubling of frequency or mass. The lower equation concerns the frequency range of resonant oscillation. However, even above the residency frequency, the sound reduction index cannot exceed the value according to the law of mass at any frequency. Fig. **5.8.1** shows a calculation example. Table **5.8.1** shows parameter values for different materials.

Table 5.8.1 Material values for different building materials. (Siikanen, 2001; Larm et al., 2006)

	m'	ρ_m	E	R_w	
	kg/m^2	kg/m3	GPa	dB	
Kipsikartonkilevy					
Saneeraus 6.5 mm	5.9	907.7	5.4	28	
Tuulensuoja 9 mm	7.6	844.4	4.8	28	
Normaali 13 mm	8.8	676.9	3.0	28	
Erikoiskova 13 mm	11.7	900.0	4.5	29	
Lattia 15 mm	16.8	1120.0	7.8	31	
Selluloosasementti (Minerit)					
Tuulensuoja 4 mm	7.7	1925.0	15	30	
Luja A 8 mm	9.8	1225.0	9.6	30	
Tuulensuoja 8 mm	13.2	1650.0	15	31	
Heavy duty 8 mm	14.7	1837.5	16	33	
Pastel 8 mm	14.8	1850.0	15	32	
Heavy duty 10 mm	18.4	1840.0	15	32	
Puupohjaiset					
Lastulevy 11 mm	7.0	636.4	2.9	29	
Vaneri 15 mm	10.4	693.3	11	26	
Lastulevy 22 mm	13.9	632.4	3.4	29	
Vaneri 21 mm	15.0	714.3	11	28	
Puukuitu (Leijona)					
Tuulensuojalevy 12 mm	3.1	258.3	0.3	23	
Tuulensuojalevy 25 mm	8.0	320.0	0.2	30	
Sementti (Aquapanel)					
Sisä ID 13 mm	14.0	1076.9	7.9	30	
Ulko OD 13 mm	15.6	1200.0	3.9	33	
Teräs					
2 mm	15.6	7800.0	213	36	
4 mm	31.2	7800.0	213	40	
Muut					
alumiini		2700	67		
lyijy		11000	alhainen		
bitumi		1000	alhainen		
raskas betoni		2500	26		
huokoinen betoni		600	2		
savitiili*		625 - 2225	2,2 - 24,7		* reikätiili 1400 kg/m^2; täystiili 1800 kg/m^2.
kalkkihiekkatiili*		625 - 2225	7,5 - 10,0		* reikätiili 1400 kg/m^2; täystiili 1800 kg/m^2.
tavallinen rakennuslasi		2500	70		
kevytsoraharkko		650 tai 950			
karkaistu kevytbetoniharkko (mm. siporex)		400-500	1,0 - 1,4		

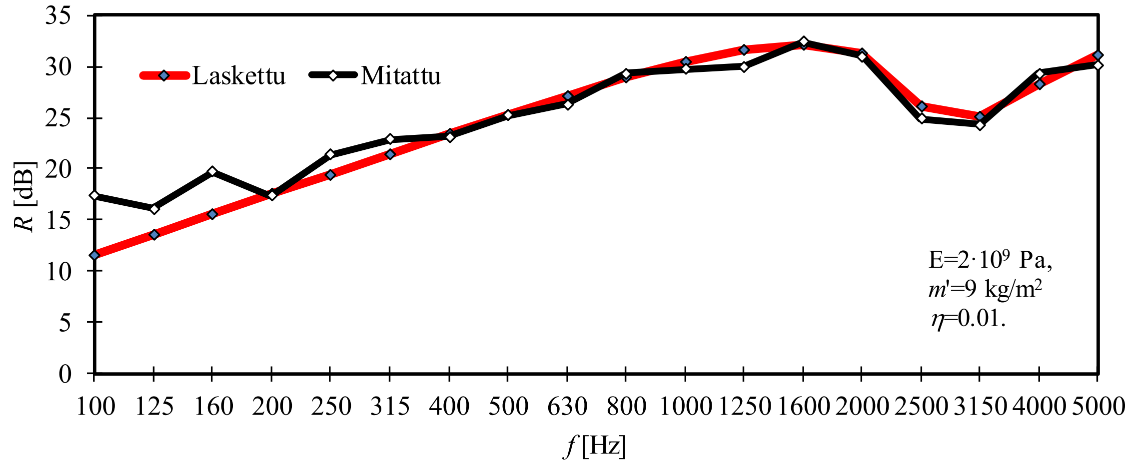


Fig. 5.8.1 Measured and calculated sound reduction index of 13 mm thick drywall.

5.9. Two overlapping thin sheets

If *N* pieces of thin panels are placed on top of each other without any cavity between them, the sound reduction index of the combination can be estimated from the equation

(5.9.1) $$R_N = 20 \cdot \log_{10}\left(\sum_{i=1}^{N} 10^{R_i/20}\right)$$

where R_i [dB] is the sound reduction index of panel i. The equation is suitable for a situation where the panels oscillate independently of each other and the bending rigidities of the panels do not change. In this case, the coincidence limit frequencies of the individual panels remain unchanged and the sound reduction index can be summed up.

Usually, the equation overestimates the result by at least 1 dB, because the panels have to be attached to the same exoskeletons and the panels do not actually oscillate independently of each other.

If the sheets are attached to each other, for example by gluing, the equation overestimates the final result. Two sheets glued together form a single panel on which the surface mass and bending rigidity can be determined. The sound reduction index of a uniform panel is then determined by the equations of a simple panel. However, due to gluing, the loss coefficient is higher than that of a single non-glued board.

5.10 Solid sheets

Here, solid boards refer to panels with a large surface mass, where the f_c is 50–500 Hz, while for thin sheets it is over 1000 Hz. The surface mass is usually 200–700 kg/m2 for solid sheets made of aggregate and 50–250 kg/m^2 for solid boards made of wood. The air sound radiation of thin panel structures into the surrounding air depends mainly on the properties of bending waves below 10000 Hz. When the thickness of the panel is greater than one sixth of the bending wave, shear waves begin to control the sound radiation of the structure and thus also the sound reduction index. The authority of the cutting waves begins with the so-called. at the transition frequency f_s (cross-over frequency), the value of which is in the region of about 2000 Hz for typical rock aggregate structures. The frequency range above this should be modeled using shear waves, since the bending wave model gives too high sound reduction index values.

At high frequencies, the bending wave propagates faster than at low frequencies. The dispersion diagrams in Fig. **5.10.1** show the attitude of the bending wave to the velocities of other waveforms as well as the speed of sound. Notable points are the coincidence limit frequency *fc* and the transition frequency. At the transition frequency *fs*, the bending and shear wave speeds are the same (*cs*=*cB*):

(5.10.1) $$f_s = \frac{c_s^2}{2\pi}\sqrt{\frac{m'}{B}}$$

Significantly below this frequency, transverse waves significant for radiation propagate at the speed of the bending wave c_B, and above it at the speed of the shear wave c_s. The transition from the bending wave to the shear wave occurs gradually.

Since only one species of transverse waves is considered at a time, i.e., the one that emits sound most efficiently, the effective velocity $c_{B,eff}$ is used to describe the effective velocity of bending waves as follows:

$$c_{B,eff} = \frac{c_B^2}{c_s}\sqrt{\frac{1}{2}\sqrt{1+4\left(\frac{c_s}{c_B}\right)^4}-\frac{1}{2}} \tag{5.10.2}$$

Here, the shear and bending wave has been considered to behave like series-connected systems. This equation is illustrated in Fig. **5.10.1** with a dashed line.

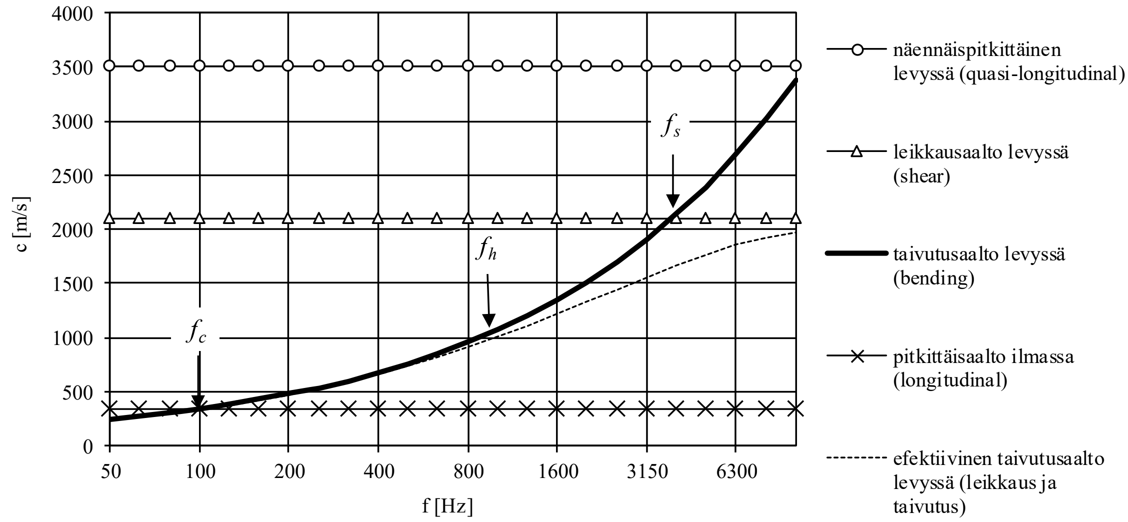


Fig. 5.10.1 Dispersion curves and different limit frequencies for waveforms of a 180 mm thick concrete slab.

A panel is considered thin from an acoustic point of view when the thickness of the panel is significantly less than the wavelength of the panel. The boundary frequency f_h, above which the panel no longer meets this condition, is

$$f_h = \frac{1}{f_c}\left(\frac{c_0}{6h}\right)^2 \tag{5.10.3}$$

that is, when the thickness of the panel is greater than one-sixth of the wavelength of the panel. Since the bending wave speed increases with increasing frequency, this means that below fh the panel can be considered thin and above it thick.

The sound reduction index of a solid board can be calculated above the critical frequency by equation (Cremer and Heckl, 1988)

$$R = 20\log_{10}\left(\frac{\omega m'}{2Z_0}\right)+10\log_{10}\left(\frac{2\eta_{tot}}{\pi}\right)+10\log_{10}\left(\frac{f}{f_c}\right) \tag{5.10.4}$$

The total loss coefficient η_{tot} and critical frequency f_c are calculated as on a thin panel. Fig. **5.10.2** shows laboratory test results for some solid walls with aggregates.

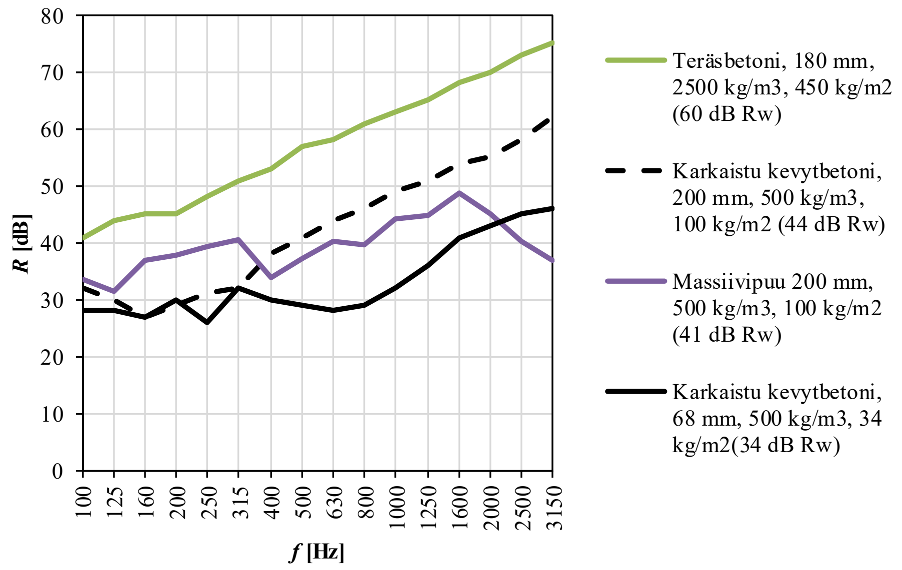


Fig. 5.10.2 Sound reduction index of heavy structures as a function of frequency. In parentheses *Rw*.

5.11. Double panel structure

In a double panel structure, two panel layers 1 and 2 are laid sequentially so that there is a cavity between the panel layers. The cavitation may include materials that are clearly lighter than the surface mass of sheet structures, the surface mass of which has no effect on sound reduction index. These can be sound absorption materials, heat insulators, fire insulators, mechanical connections between panels, air layers and combinations of these. A half of a panel structure can consist of several panels, but they are thought of as a single panel.

If the material in the cavitation is glued to the panels, then it is a question of sandwich board. This will be discussed separately.

An uncoupled double panel design means that there is no mechanical coupling between panel layers 1 and 2. With such structures, optimal performance in relation to mass and weight is achieved. Sound reduction index is improved when

i. total mass increases
ii. the thickness of the cavity increases, and
iii. the absorption ratio and quantity of cavity absorption material increase,
iv. The difference in the limit frequencies of the cooccurrence of panel layers 1 and 2 increases.

The coupled double panel design means that panel layers 1 and 2 are connected by mechanical coupling. These include, for example, frames and rails in partitions, suspension springs in suspended ceilings, frames in windows and doors, and necks/beams in floors. Connections easily dominate sound transmission. Em. Factors i–iii are still significantly less relevant. After mass, the characteristics of the couplings affect the sound reduction index the most. Sound reduction index is improved when

v. The number of connections (spindles, rails or point suspensions) is reduced
vi. The flexibility of the connections is improved, i.e. the dynamic stiffness is reduced
vii. the fastening of the panel to the spindles decreases (the number of screws decreases or the tightness of the screwing decreases)

The effect of factors ii–iii is almost negligible if the panels are connected by dense and rigid spines. If the couplings are flexible, factors ii–iii continue to play an important role.

Figures **5.11.1–4** show the effects of double panel structure parameters based on laboratory measurements.

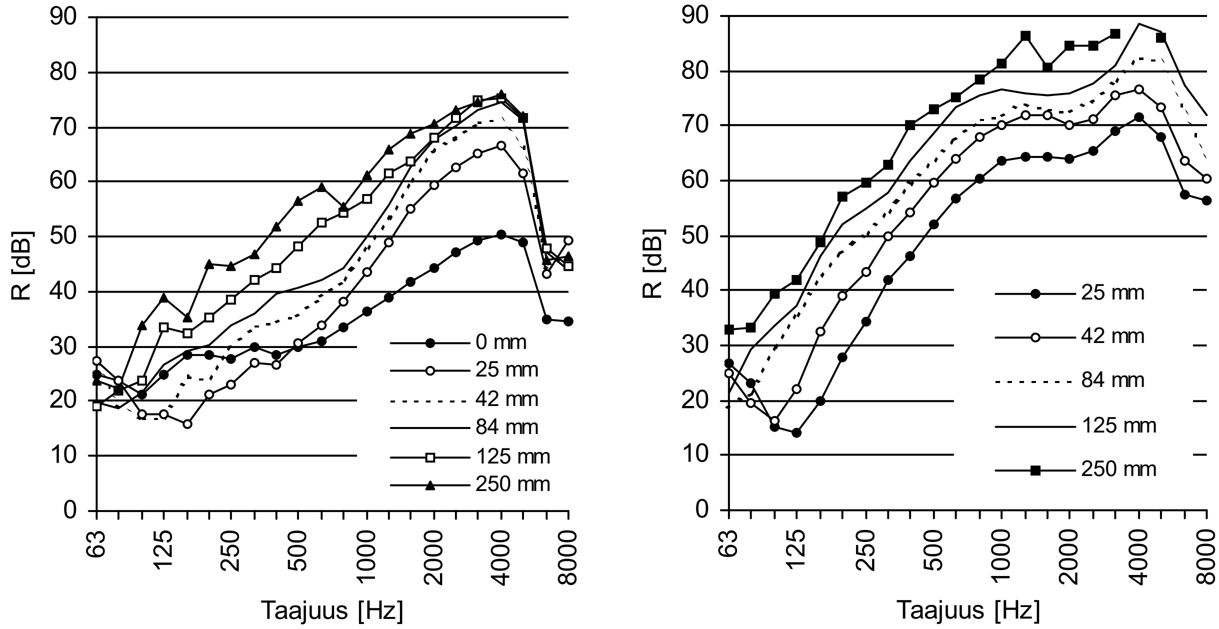


Fig. 5.11.1 Effect of cavity thickness *d* [mm] on sound reduction index at different frequencies when (a) cavity empty and (b) cavity cavity filled with mineral wool (filling rate *FR* between 60–88 %). In both figures, the surface panels are 2 mm steel panels with no connections between them (Hongisto et al., 2002, Hongisto et al., 2002).

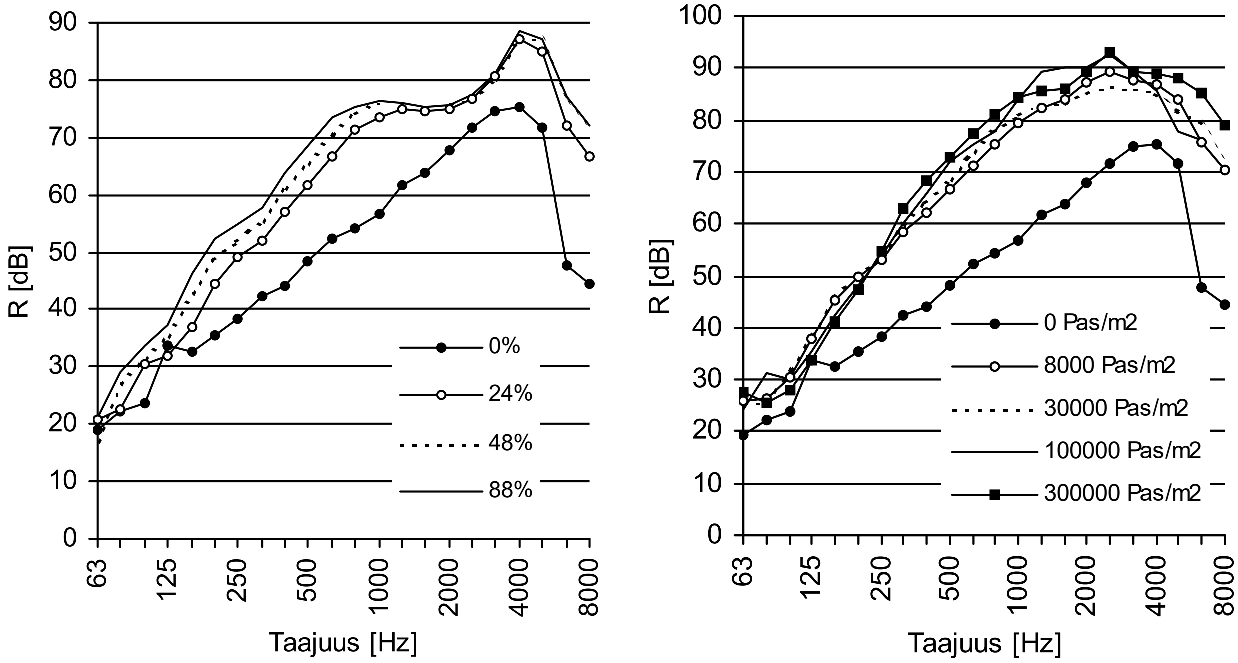


Fig. 5.11.2 (a) Effect of the filling level FR of mineral wool in cavitation on sound reduction index at different frequencies. (b) Effect of specific flow resistivity r of cavity mineral wool at filling level *FR* > 70 %. The reference point is empty cavity (0 Pa·s/m2). In both images, the surface panels are 2 mm steel sheets with a cavity of 125 mm between them without mechanical connections (Hongisto et al., 2002, Hongisto et al., 2002).

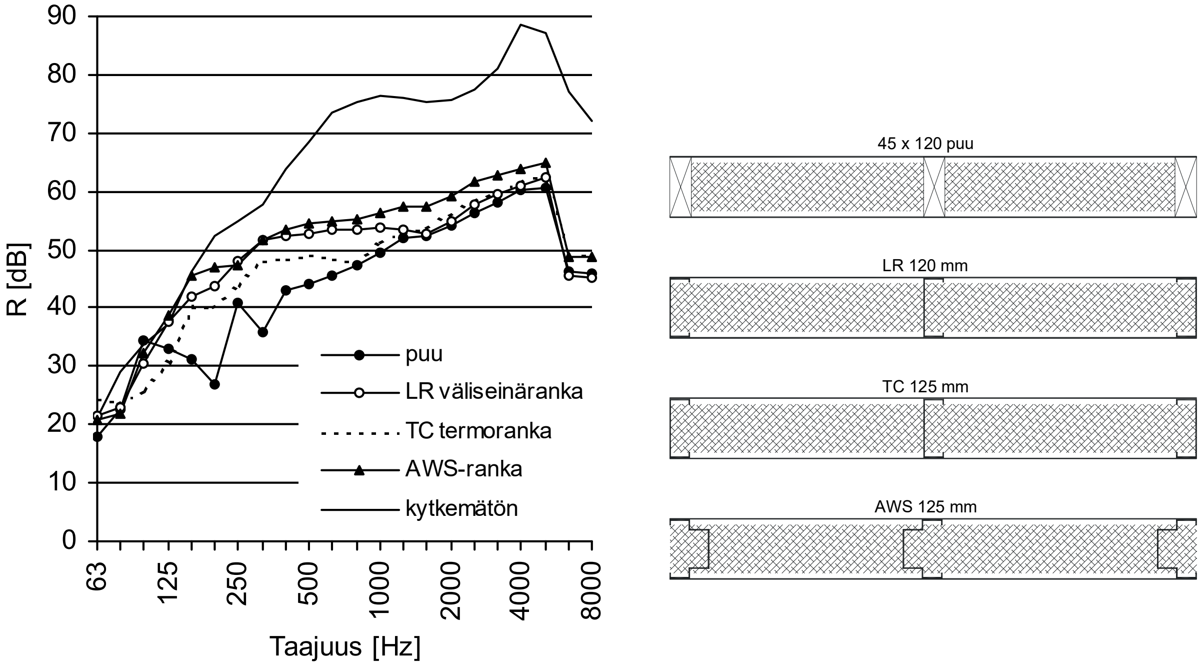
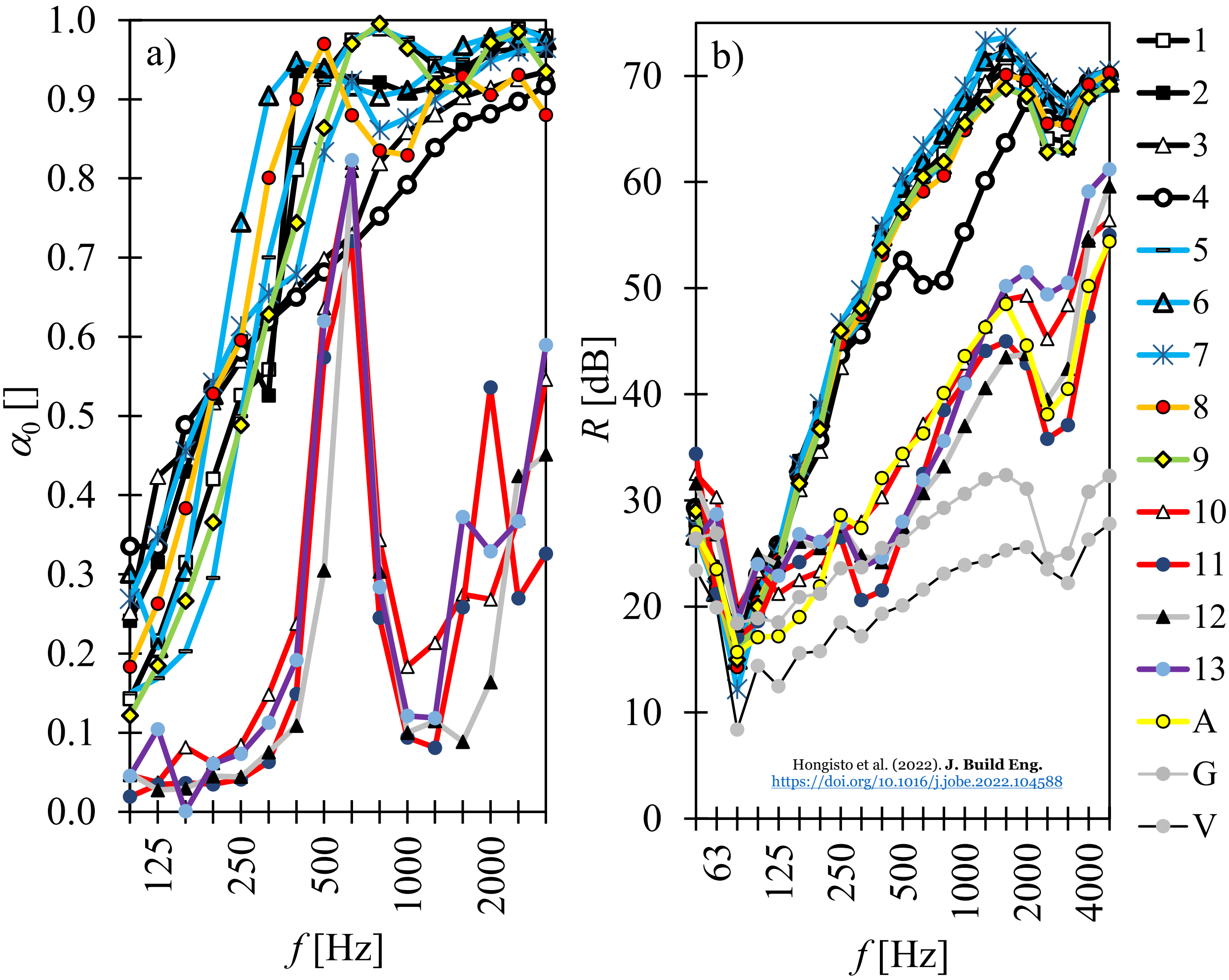


Fig. 5.11.3 Effect of dynamic rigidity of cavitation frames on sound reduction index at different frequencies with a frame pitch of b = 600 mm. The surface panels are 2 mm steel sheets with a cavity of 120 or 125 mm between them, filled with mineral wool (FR>75%). The dynamic stiffness of the spindles per unit length, K' are: Wood: not measured, estimate >100 MN/m^2, LR: 3.3 MN/m^2, TC: 2.8 MN/m^2, AWS: 0.2 MN/m^2. The point of comparison is an uncoupled structure without spindles (no figure, b=∞)(Hongisto et al., 2002, Hongisto et al., 2002).



	Insulator material	ρ [kg/m^3]	r' [kPa×s/m^2]
1	Ultra-low density stone wool slab	25	6.6
2	Low density stone wool slab	25	11.6
3	Medium density stone wool slab	75	30.7
4	High density stone wool slab	100	33
5	Ultra-low density glass wool roll	11	3
6	Low density glass wool slab	16	10
7	Medium density glass wool slab	70	15.8
8	Cellulose slab	37	11.1
9	Wood fiber slab	50	5.9
10	Expanded polystyrene board	18	530
11	Polyisocyanurate board	30	2600
12	Phenolic foam board	30	2700
13	Cellular glass board	100	68.3

Fig. 5.11.4. a) The absorption ratio of the normal angle of incidence of various heat insulators 100 mm thick, α0, as a function of the frequency f when the insulation is placed against a rigid background. (b) Sound reduction index of the double panel structure, R, as a function of frequency for 14 double sheet structures of 13 mm drywall (G), 100 mm cavity (filled with insulation 1–13 or air A), and 9 mm plywood. The R values of plywood and gypsum alone are also presented as a comparison. Insulation types 1-9 are through-porous and the rest are closed-porous. The values of the former are clearly higher than those of the latter. NOTE: The absorption peak of insulation types 10-13 at 630 Hz is not real, but is caused by an air gap between the insulation and the backpanel in the impedance tube test (panel resonator effect). This is confirmed by Fig. b, where no similar phenomenon is seen (Hongisto et al., 2022).

5.12. Double panel

I. INTRODUCTION

In the sound reduction index of a double panel structure, the sound reduction index along cavity (Rc, *cavity*) and spindles (R_c, bridge) should be taken into account separately. The combined sound insulation of the two routes is

(5.12.1) $$R = -10 \cdot \log_{10}\left(10^{-R_c/10} + 10^{-R_b/10}\right)$$

If the cavity is completely absorbing, then the formula R_{cI} is used. If the cavity is partially absorbing, the formula R_{cII} is used. If the structure has a rigid coupling, then the formula R_{bI} is used. If the structure has flexible couplings, then the formula R_{cII} is used. If there are no spines, $R_b=\infty$ and R_c is dominant.

II. PERFECTLY SOUND-ABSORBING CAVITY

Consider an ideal double panel design, where there is no mechanical coupling between the halves and the cavity is completely absorbing, preventing echo in the cavity. Sound reduction index can be calculated using equations (Sharp, 1978; Rindel, 2018):

(5.12.2) $$R_{cI} = \begin{cases} 20 \cdot \log_{10}\left(10^{R_1/20} + 10^{R_2/20}\right) + R_{mam}, & f < f_{mam} \\ R_1 + R_2 + 20 \cdot \log_{10}(fd) - 29, & f_{mam} < f < f_d \\ R_1 + R_2 + 6, & f > f_d \end{cases}$$

where R_1 and R_2 [dB] are the sound reduction index properties of surface panels 1 and 2, d [m] is the thickness of the cavity, and f [Hz] is the frequency. At the mass-air-mass resonant frequency, f_{mam} [Hz], R reaches its lowest value because halves 1 and 2 and the air spring between them resonate. It is calculated from the equation

(5.12.3) $$f_{mam} = \frac{1}{2\pi}\sqrt{\frac{1.8\rho_0 c_0^2}{d}\frac{(m'_1 + m'_2)}{m'_1 m'_2}} = 80\sqrt{\frac{(m'_1 + m'_2)}{dm'_1 m'_2}}$$

where ρ_0 is the density of air [1.2041 kg/m3 ; 20 ºC], c0 *[m/s]* is the speed of sound in air (343 m/s) and m'1 and m'_2 [kg/m2] are the surface masses of halves 1 and 2. Sound reduction index at low frequencies (*f<fmam*) is the same as if the panels of the structure were together and a simple panel was considered.

The limit frequency f_d [Hz] is calculated from the equation:

(5.12.4) $$f_d = \frac{c_0}{2\pi d}$$

The limit frequency corresponds to the condition k_d=1 between the thickness and frequency of the cavity, i.e., a situation where d is about 16% of the wavelength. At frequencies higher than this, a transverse sound field begins to form in relation to the panels, and the coupling between the panels slightly intensifies, and the increase in sound reduction index drops from 18 dB/octave to 12 dB/octave.

At the mass-air-mass resonant frequency, a bump appears in the sound reduction index, which is especially striking if the cavity is sound-absorbing. If it is to be taken into account in the calculations, the equation can be used (Rindel, 2018)

(5.12.5) $$R_{mam} = 20 \cdot \log_{10}\left[1 - \left(\frac{f}{f_{mam}}\right)^2\right]$$

The mass-air-mass resonant frequency and the exact depth and shape of the pit are difficult to predict because the phenomenon is narrowband, measurements are made in 1/3 octave and measurement uncertainties are high at low frequencies. The depth of the pit also depends on the type and amount of absorption material, as well as the tightness of the cavity.

R_1 and R_2 are obtained either computationally or from measurements. The measurement result obtained by the pressure method overestimates the actual sound reduction index at low frequencies, especially if the sample size is small. The computational result therefore easily leads to overestimation at low frequencies if measurement results are used, since the possible error is doubled.

Above the frequency f_{mam}, sound reduction index increases very quickly, according to the model 18 dB/octave. Above the limit frequency f_l, the growth rate slows down to 12 dB/octave. When the limit frequency of the coincidence of the panels is reached, the sound reduction index begins to deteriorate, as with a simple panel structure. Above the limit frequency, sound reduction index starts to increase sharply again. If the panels have different frequencies of residence, the loss of sound reduction index at this frequency will not be as pronounced.

III. INCOMPLETELY SOUND-ABSORBING CAVITY

If there is no absorption material in the cavity, the sound reduction index decreases sharply due to the echo forming in the cavity. The modes can be divided into lower (k_d<1) and upper (k_d>1). The lower modes occur in the direction of the panels. The upper modes occur perpendicular to the panels (higher modes). The upper modes play much less role in sound reduction index than the lower modes. At high frequencies (k_d>1), reverberation is also perpendicular to the panels.

The cavity does not echo at frequencies lower than the lowest mode, so the absorption material also does not affect the sound reduction index. The lowest resonance f_{c1} of the cavity is calculated from the equation

(5.12.6) $$f_{c1} = \frac{c_0}{2 \cdot \max\left[L_{x,c}; L_{y,c}; L_{z,c} \right]}$$

where $L_{x,c}$, $L_{y,c}$, and $L_{z,c} = d$ [m] are the width, height and thickness of the cavity. Fig. **5.12.1** shows a diagram showing the resonant frequencies at different cavity dimensions.

According to experimental studies, the sound reduction index deteriorates logarithmically with respect to the filling rate of the absorption material. The sound reduction index in the case of an incompletely absorbing cavity is calculated from the equation

(5.12.7) $$R_{cII} = R_{cI} + \Delta R_{abs}$$

where the deterioration term ΔR_{abs} [dB], determined by the cavity absorption ratio, is given by

(5.12.8) $$\Delta R_{abs} = \begin{cases} 0, & f < f_{c1} \\ 10 \cdot \log_{10}\left(\alpha_{eff}\right), & f \geq f_{c1} \end{cases}$$

If the cavity is only partially filled with absorption material, the effective absorption ratio is given by the formula

(5.12.9) $$\alpha_{eff} = \alpha_c \cdot FR$$

where α_c is the absorption ratio of the cavity material and *FR* is the cavity filling ratio, which takes values 0–1. If the cavity is filled with absorption material, *FR*=1. The absorption ratio of the cavity material may be values measured or calculated according to the thickness direction d of the material. However, the absorption ratio may be higher than this, since in the lower modes below the limit frequency fd, the sound field occurs only in the direction of the surface of the panels, and not perpendicular to the surface of the cavity material.

If there is no separate absorption material in the cavity, two weak absorption mechanisms remain, which means that the absorption ratio is never zero in the cavity:

- absorption of hard inner surfaces of the cavity (perpendicular angle of incidence)
- viscous friction losses of large panel surfaces of the cavity when sound travels in the direction of the surfaces.

Especially at low cavity thicknesses, viscous friction losses are significant, if not large. The exact value of the absorption ratio of cavity is difficult to determine. Brekke (1980) has proposed the following approximation for the absorption ratio of empty cavity:

(5.12.10) $$\alpha_{eff} = \begin{cases} 0.5, & d \leq 0.02\,\text{m} \\ 0.01/d, & d > 0.02\,\text{m} \end{cases}$$

There are no table values for the effective absorption ratio because the same material produces different absorption in different cavitations. The recommended upper limit for forecast calculations is α_{eff}=0.95. Based on measurements, the differences between empty and fully absorbing cavity rarely exceed 15 dB at any frequency. The recommended lower limit is 0.04, with ΔR_{abs} = -14 dB.

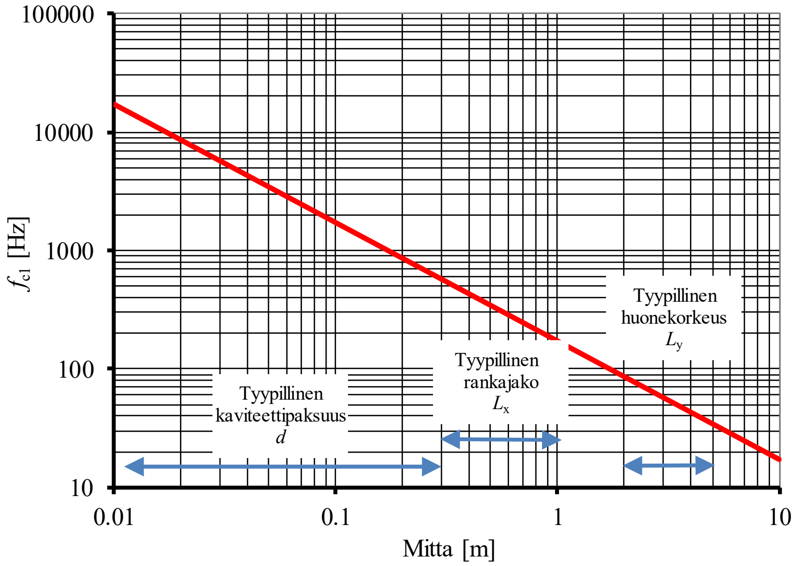


Fig. 5.12.1 Dependence of the lowest mode formed in the cavity on the measure of cavity. The modes in the direction of the board are usually at frequencies of 40–600 Hz. The modes perpendicular to the direction of the panel are usually at frequencies above 600 Hz.

IV. Rigid coupling between panels

If panels 1 and 2 are connected by a rigid coupling, such as a wooden frame or rigid point connections, then the sound reduction index along the connection path is

(5.12.11) $$R_{bI} = 20 \cdot \log_{10}\left(10^{R_1/20} + 10^{R_2/20}\right) + \Delta R_b$$

where must be $\Delta R_b > 0$ dB.

The frequency-independent improvement ΔR_b [dB] with respect to the sum of the sound reduction index of two panels (logarithm term) achieved by rigid line connections such as timber frames or fully rigid steel frames is given by:

(5.12.12) $$\Delta R_b = 10 \cdot \log_{10}\left(bf_{cL}\right) + 20 \cdot \log_{10}\left(\frac{m'_1}{m'_1 + m'_2}\right) - 18$$

where b is the distance [m] between the line connections, i.e. the column division, and fcL [Hz] is

(5.12.13) $$f_{cL} = \left[\frac{m'_1\sqrt{f_{c2}} + m'_2\sqrt{f_{c1}}}{m'_1 + m'_2}\right]^2$$

where f_{ci} [Hz] is the limit frequency of the coincidence of panel i and m'_{i} [kg/m2] is the surface mass of panel i.

<u>In the case of rigid point connections</u>

(5.12.14) $$\Delta R_b = 10 \cdot \log_{10}\left(\frac{S\pi^3 f_{cP}^2}{N8c_0^2}\right) = 10 \cdot \log_{10}\left(\frac{Sf_{cP}^2}{N}\right) - 45$$

where S [m^2] is the area of the structure, N_{p} [] is the number of point connections in the structure, and f_{cP} [Hz] is

(5.12.15) $$f_{cP} = \frac{m'_1 f_{c2} + m'_2 f_{c1}}{m'_1 + m'_2}$$

A rigid point connection is represented, for example, by a suspended ceiling implemented with metal hanging wires. A situation similar to rigid point coupling arises, for example, in the case of cross-collision, where N is the number of hybrids.

V. FLEXIBLE CONNECTION BETWEEN PANELS

With a flexible connection, sound reduction index is better compared to a rigid connection. Sound reduction index along a rigid coupling is then

(5.12.16) $$R_{bII} = R_{bI} + \Delta R_{fb}$$

The additional insulation ΔR_{fb} produced by flexible coupling resembles the force insulation capacity of a spring (Bradley&Birta, 2001):

(5.12.17) $$\Delta R_{fb} = -5\log_{10}\left(\frac{1+4\zeta^2\left(\frac{f}{f_r}\right)^2}{\left[1-\left(\frac{f}{f_r}\right)^2\right]^2+4\zeta^2\left(\frac{f}{f_r}\right)^2}\right)$$

where f_r [Hz] is the resonant frequency formed by panel 1, flexible coupling and panel 2 and is the ζ attenuation ratio (constant), the value of which, based on measurements, is usually about 0.30. Fig. **5.12.2** shows the frequency dependence of the equation with different damping ratio values.

The frequency *fr* must be determined separately based on the dynamic stiffness of the springs, the number of springs and the masses of the panels. It is worth sizing it so that it is not the same as f_{mam}. In the case of flexible spines, f_r is given by:

(5.12.18) $$f_r = \frac{1}{2\pi}\sqrt{\frac{K'}{b}\frac{(m'_1+m'_2)}{m'_1 m'_2}}$$

where m'_1 and m'_2 are the surface masses of the panel halves [kg/m^2], K' [N/m^2] is the dynamic stiffness of the spine per unit length, and b [m] is the rank division in the double panel structure under consideration.

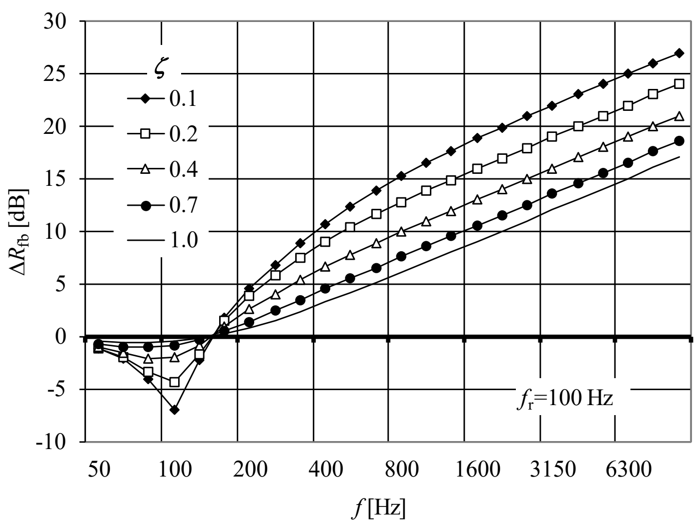


Fig. 5.12.2 Additional insulation by flexible spine ΔR_{fb} for sound reduction index through the coupling of a double panel structure at different values of the damping ratio ξ when the resonant frequency f_r is constant.

Laboratory studies have shown that a panel structure with flexible spines does not achieve the theoretical value

ΔR_{fb} but is limited to 10–20 dB, depending on the case (Virjonen, 2009). The reason is thought to be that flexible frames use mounting rails at the top and bottom, which are stiffer than the flexible frames in the middle of the structure. In addition, the rails are usually rigidly attached to the floor and ceiling. The flexible spines on the sides are also attached to the vertical structures, which stiffens them. The stiffness at the bottom is increased separately if the panels touch the floor and are not elastically resting on rails and frames. Due to the rigid peripheral connection thus formed, sound is emitted from a width of a few tens of centimetres on its sides, similar to if the sides were surrounded by a rigid spine (Fig. **5.12.3**). Flexible rail structures would solve the problem.

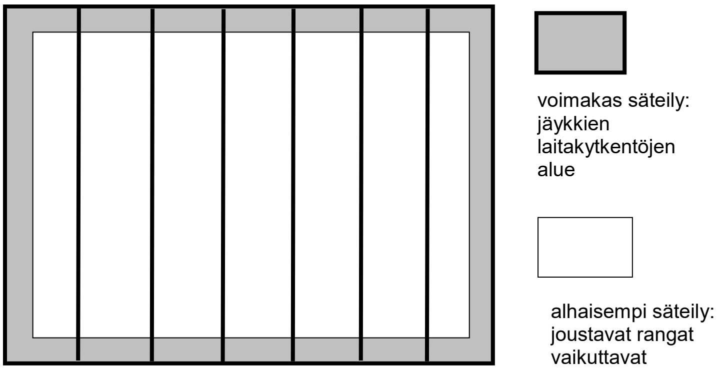


Fig. 5.12.3 The sound reduction index of a double sheet structure wall with flexible frames is reduced if the sides are rigid connections.

5.13 Sandwich construction

Sandwich construction (Fig. **5.13.1**) refers to a double panel structure in which the surface panels are connected to each other by a flexible and light nuclear material over their entire surface area. In terms of sound reduction index, sandwich structures are significantly inferior to a sheet structure wall of similar weight with rigid frames. At some solutions or frequencies, the sound reduction index of a sandwich structure is even weaker than that of a simple panel of the same mass. Sandwich structures are very rigid in relation to their mass and are increasingly encountered in construction, especially when mineral wool insulation is replaced with hard insulation with higher rigidity than wool. The sandwich structure works like a mass-spring-mass system that resonates strongly at a certain frequency. Sandwich structures include, for example:

- wall and roof elements in which mineral wool lamellas are glued between two sheet metal sheets
- EPS-filled thermal insulation doors or
- fire doors in which sheet metal sheets on the surface are glued to a layer of stone wool that is a nuclear material.
- "honeycomb-filled" doors
- double structure, in which wool encased in cavitation mechanically connects the panel halves to each other,
- Floating floors

Examples of the sound reduction index behaviour of lightweight sandwich structures are given in Fig. **5.13.2.** In both examples, sound reduction index does not improve much when the structure is thickened by thickening the core material, unlike in the case of a double panel structure. The dilation resonance of the sandwich structure is calculated from the equation

(5.13.1) $$f_d = \frac{1}{2\pi}\sqrt{s'\frac{m'_1+m'_2}{m'_1\,m'_2}}$$

where s' [N/m^3] is the dynamic rigidity of the nuclear material per unit area and m'_1 and m'_2 [kg/m^2] are the surface masses of the panel halves. Underfloor wools are often in the order of 10–30 MN/m^3. With harder insulation, the values can exceed 1000 MN/m^3. The resonant frequency increases as the thickness of the nuclear material decreases due to the stiffening of the spring.

Depending on the dynamic rigidity of the nuclear material, the sound reduction index can vary very much. In **the examples in Fig. 5.13.1**, the surface masses of the panels do not differ much, but the results are substantially better at high frequencies if the nuclear material is looser. The reason for this is that sound reduction index begins to increase only above the resonant frequency.

The direction of the fibers of the wool material affects the sound reduction index of the sandwich structure. This is due to the fact that the dynamic rigidity of wool is significantly higher in the direction of the fibers (the so-called cross wool) than perpendicular to them (the so-called clapboard). In a more rigid case, the resonant frequency is at a higher frequency. Fig. **5.13.3** shows the comparative result when the same structure and manufacturing method has been used, but the direction of the fibres of the wool used as nuclear material has been changed.

The sound reduction index of a sandwich structure cannot be improved without a significant increase in mass or replacement of nuclear material. Sound reduction index is often improved by panel structure cladding added on top of the structure.

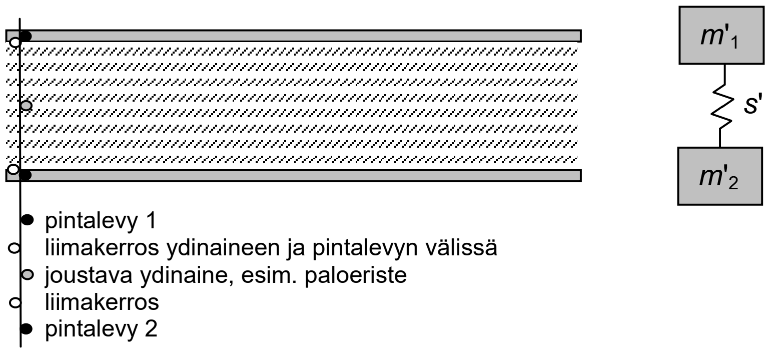


Fig. 5.13.1 In sandwich construction, the surface panels are strongly attached to the nuclear material, so that the entire sandwich structure oscillates like a solid panel.

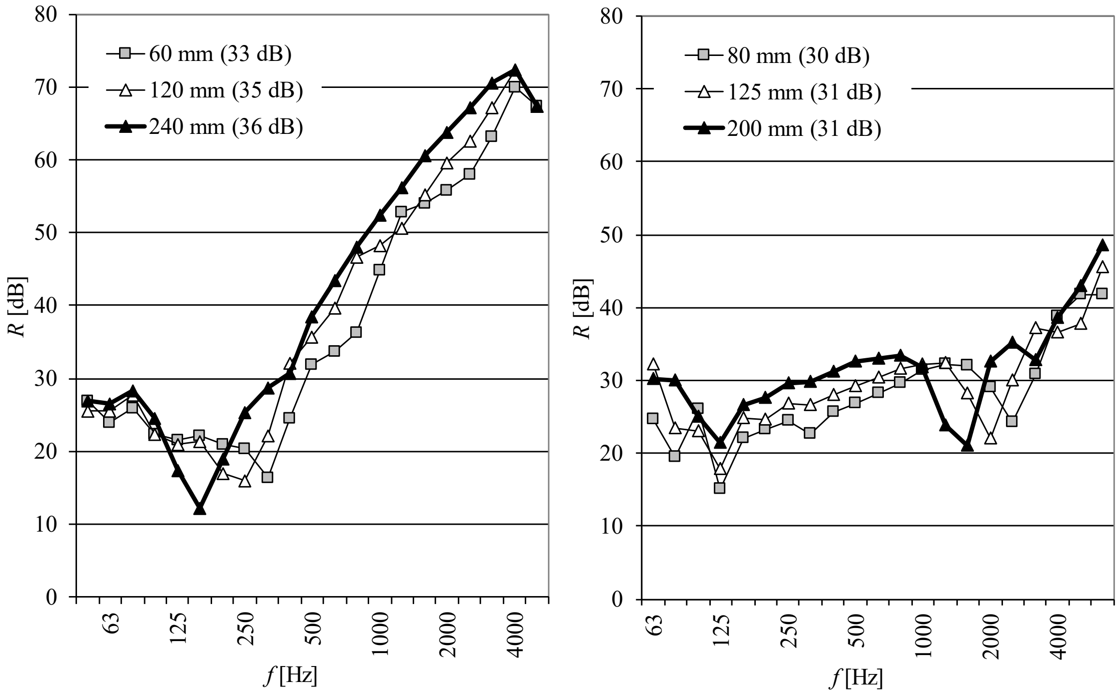


Fig. 5.13.2 The dependence of the sound reduction index of a sandwich structure on the thickness of the core material. On the left, the surface panel was 1 mm steel and the core material was mineral wool (100 kg/m^3). The parentheses have the R_w value. The surface panel on the right is 0.6 mm steel and the core material is mineral wool (125 kg/m^3). On the left, the dynamic rigidity of the wool was less than on the right.

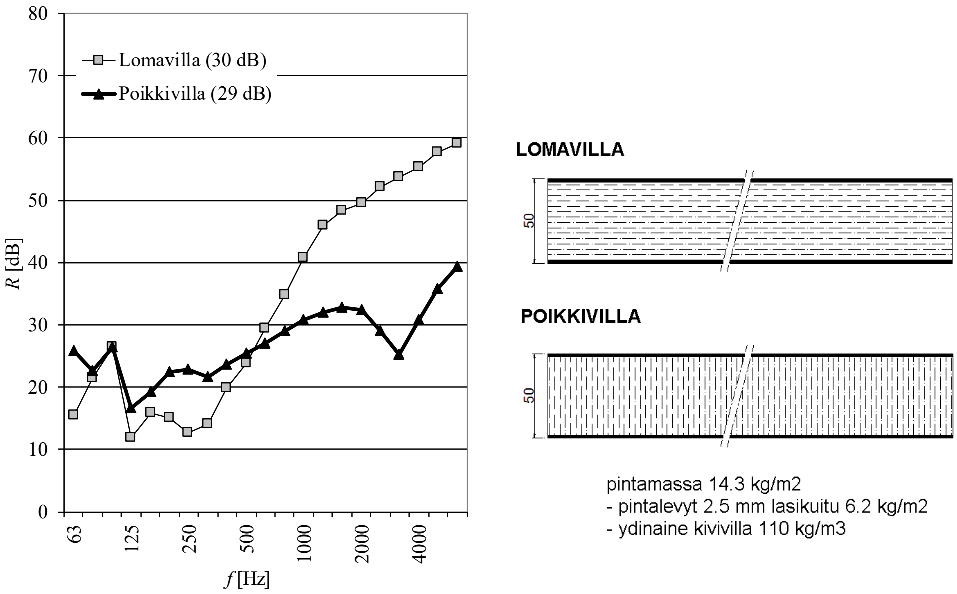


Fig. 5.13.3 The effect of changing the direction of the fiber of mineral wool on sound reduction index in a sandwich structure. When the wool fibers were perpendicular to the panel (poikkivilla), the dynamic rigidity of the wool was 90 times that of lamellar wool (lomavilla).

5.14. Porous board

Fig. **5.14.1a** shows the dependence of the sound reduction index of a porous panel on thickness and flow resistance. The sound reduction index of mineral wool improves with an increase in flow resistance and density. Density affects the sound reduction index of low frequencies and flow resistance of high frequencies. The sound reduction index of the porous board increases at high frequencies almost linearly with increasing thickness. When the thickness of the porous board doubles, the sound reduction index almost doubles. With dense building boards, according to the law of mass, sound reduction index increases by no more than 6 dB with a doubling of the mass (thickness). The sound reduction index of a porous board at high frequencies is based on friction in the pores, and not on the bending vibration of the entire panel. Therefore, the sound insulation of the absorption air depends on the distance traveled by the sound inside the porous panel and thus on the thickness of the porous panel.

Fig. **5.14.1b** shows the effect of absorbent coating on the sound reduction index of the wall structure. The principle of operation of the absorbing panel is different in the cavitity of the panel structure than on the visible surface of the structure. When located on the surface, the absorption panel improves sound reduction index in the truest sense of the word. In contrast, in cavitation, a porous panel eliminates echoing. Thus, the porous layer prevents an unfavorable increase in sound level in the cavity. In other words, the board does not "improve sound reduction index" but prevents the sound reduction index from deteriorating due to echoing.

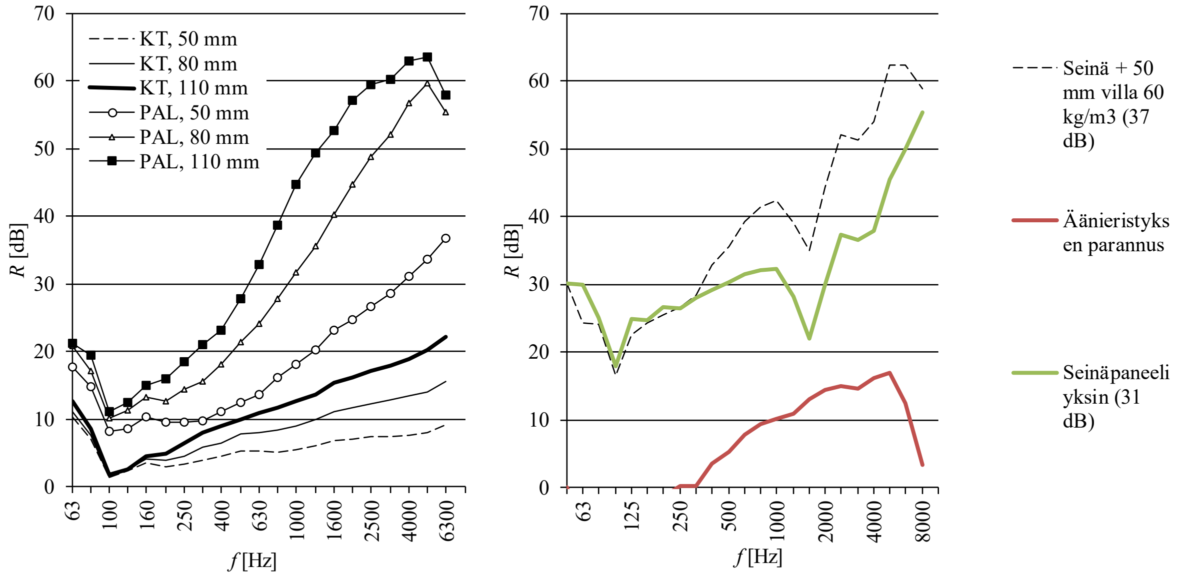


Fig. 5.14.1 (a) Sound reduction index of glass wool as a function of thickness and density. PAL: ρ=117 kg/m3, r=90 000 Pa·s/m^2. KT: ρ=19 kg/m^3 and r=8 000 Pa·s/m^2. (Hongisto et al., 2002) (b) the effect of 50 mm thick mineral wool slabs on the SRI of the wall panel. The Rw values are shown in parentheses. Absorption coat-ing improved sound reduction index mainly above 200 Hz. A similar improvement has also been observed with absorption panel coating installed on top of a massive wall. (RIL 243-1-2007)

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6 IMPACT SOUND INSULATION

6.1. Impact SPL

The purpose of impact sound insulation is to reduce the level of structure-borne noise caused by impacts to the floor from above in other rooms of the same building. This chapter focuses on the room below, where the sound is usually loudest. In addition, this chapter remains in laboratory conditions and does not consider secondary road transitions.

Shock sounds in housing include walking on the floor, running, playing or other exercise, objects falling to the floor, moving furniture, vacuuming and other similar sounds. With impact, the structure begins to vibrate. Part of the vibration is converted into air sound through radiation.

The sound heard in a room with a shock sound source is called a drum sound. Drumming noise is most common on parquets and laminates (Johansson &; Nilsson, 2005). That is not covered in this chapter.

Impact sound insulation is determined using a calibrated sound source, a tapping machine (Fig. **6.1.1**). The output power of the excitation is constant and does not need to be measured separately. The SPL produced by the tapping machine in the neighbouring room is of interest. The lower the SPLs caused by the impact sound device, the better the impact sound insulation performance of the structure.

A tapping machine according to ISO 10140-5 consists of five hammers weighing 0.5 kg that fall from a height of 40 mm to the floor at a speed of 0.886 m/s twice per second at equal intervals, denoted by Ti = 0.1 s. The floor is subjected to 10 impacts per second. This causes a force spectrum with a maximum frequency *of fi* = 1/ Ti = 10 Hz and its integer multiples. The impact tone instrument must not weigh more than 25 kg in order to achieve the same load effects on different machines.

In fact, a impact tone device produces a different excitation on hard surfaces than on flexible surfaces. In addition, the excitation also depends on the load on the impact sound device.

Fig. 6.1.1 Examples of commercial impact sound sources: impact sound device, rubber ball and bang machine. The images are not on the same scale. Bang machine is not available in European countries.

The normalized impact SPL Ln (L_n) caused by a tapping machine is determined under laboratory conditions (ISO 10140-3) using the equation

(6.1.1) $$L_n = L_2 + 10 \cdot \log_{10}\left(\frac{A_2}{A_0}\right)$$

where L_2 [dB re 20 μPa] is the SPL in the reception room, A_2 [m^2] is the absorption area in the reception room and A_0 = 10 m^2 is the reference absorption area. The measurement is made in third octaves at frequencies 50–5000 Hz or 100–3150 Hz. Examples of measurement results are presented in several figures in this chapter. The normalized impact SPL measured in the field is L'_n.

The impact SPL is a quantity based on the absolute level, while airborne sound insulation is a quantity based on power damping. The lower the L_n, the better the impact sound insulation.

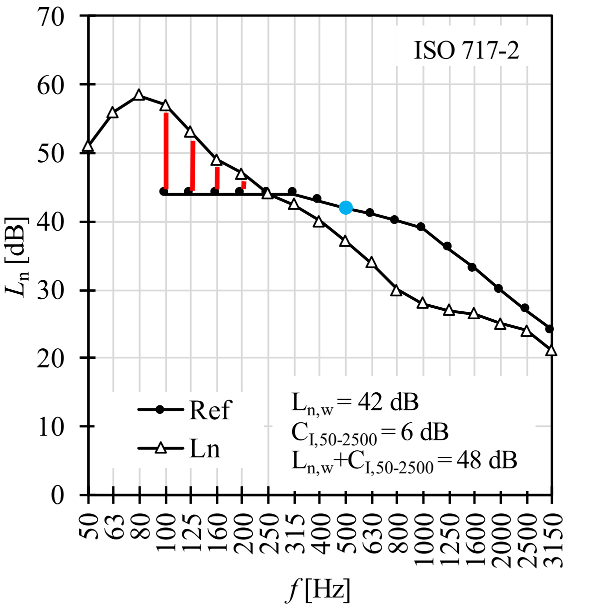
In the laboratory environment, side road transitions are minimized, so that the sound from the tapping machine does not travel significantly along other structural routes to the reception room. In the building, the impact sound insulation of the intermediate floor will be determined from 1.1.2018 using the apparent standardised impact SPL $L'_{n,T}$:

(6.1.2) $$L'_{n,T} = L_2 - 10\log_{10}\left(\frac{T_2}{T_0}\right)$$

where T_2 [s] is the reverberation time in the reception room and T_0=0.5 s is the so-called reverberation time. reference reverberation time. Half a second is the typical reverberation time of a living room, regardless of volume (Kylliäinen et al., 2016). The absorption area increases with increasing room volume and the correction term in equation **6.1.1** therefore seemingly depends on the room volume. L'_{nT} therefore corresponds better to the impact SPL observed in the building's room than L'_n, which was used until 31.12.2017 in building regulations for field measurements of the impact SPL.

6.2. Single-number quantities

f	Ln [dB]	Ref [dB]	Ref [dB]	Dev [dB]	$10^{Ln/10}$
50	51.0				125893
63	56.0				398107
80	58.5				707946
100	57.0	*L* n,w+2	44	**13.0**	501187
125	53.0	*L* n,w+2	44	**9.0**	199526
160	49.0	*L* n,w+2	44	**5.0**	79433
200	47.0	*L* n,w+2	44	**3.0**	50119
250	44.0	*L* n,w+2	44	0.0	25119
315	42.5	*L* n,w+2	44	0.0	17783
400	40.0	*L* n,w+1	43	0.0	10000
500	37.0	***L* n,w**	42	0.0	5012
630	34.0	*L* n,w-1	41	0.0	2512
800	30.0	*L* n,w-2	40	0.0	1000
1000	28.0	*L* n,w-3	39	0.0	631
1250	27.0	*L* n,w-6	36	0.0	501
1600	26.5	*L* n,w-9	33	0.0	447
2000	25.0	*L* n,w-12	30	0.0	316
2500	24.0	*L* n,w-15	27	0.0	251
3150	21.0	*L* n,w-18	24	0.0	



Ei-toivottujen poikkeamien summa:
30.0 dB (Korkein sallittu 32.0 dB).

Ln: Normalisoitu askeläänenpainetaso (mittaustulos)
Ref: Vertailukäyrä asennossa 42 dB (500 Hz on ankkuritaajuus)
Dev: Ei-toivottu poikkeama: Max(0; Ln - Ref)

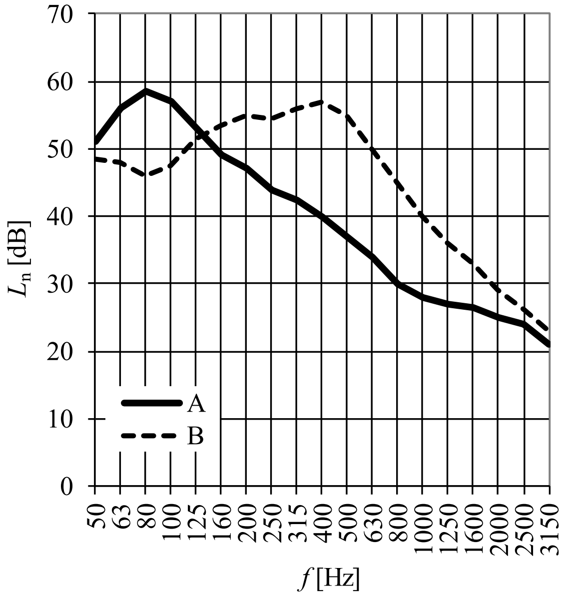
$S=\Sigma[10^{Ln,i/10}]=$ 2125782
$L_{n,sum}=10*\log_{10}(S)$ 63.3
$\mathbf{C_{I,50\text{-}2500}=L_{n,sum} - 15 - L_{n,w}=}$ **6.3**

Fig. 6.2.1 The weighted impact SPL, $L_{n,w}$, is determined from the measured L_n values by the reference curve procedure. The shape Ref of the reference curve is always the same, but its position in the y-direction depends on what value is given for the anchor frequency, which is 500 Hz (trial of $L_{n,w}$). The value at this anchor frquency is given the lowest possible value so that the sum of unwanted deviations does not exceed 32.0 dB. An unwanted deviation occurs when the value of the reference curve is below the measurement result. With a guess of 42 dB, the reference curve is positioned at Ref42. The unwanted deviations in this example occur at frequencies of 100–200 Hz. Since their sum is less than 32.0 dB, $L_{n,w}$ is equal to the value of the reference curve at the anchor frequency.

In business and regulations, the impact SPL is described by single-number values. Their definitions are set out in ISO 717-2. The most important of them is the normalized impact sound level number $L_{n,w}$. It is determined by the so-called. by the reference curve method **as shown** in Fig. 6.2.1.The standard also presents two spectral weighting terms: C_I and $C_{I,50\text{-}2500}$. Although spectral weighting terms are optional according to ISO 717-2, the single digit value $L_{n,w}+C_{I,50\text{-}2500}$ is used bindingly in many countries because impact sounds often contain audible low-frequency components at frequencies below 100 Hz. The spectral weighting term $C_{I,50\text{-}2500}$ is calculated from the equation

(6.2.1)
$$C_{I,50\text{-}2500} = 10 \cdot \log_{10}\left(\sum_{i=50}^{2500} 10^{L_{n,i}/10}\right) - 15 - L_{n,w}$$

Fig. **6.2.2** demonstrates the use of the spectral weighting term for two different types of constructions. If the floor resonance (the peak of the impact SPL) is below the frequency 100 Hz and the L_n values at the resonant frequency are greater than the other frequencies, the value of the spectral weighting term increases because $L_{n,w}$ does not take this frequency band into account. If, on the other hand, the resonance is at or above 125 Hz and the L_n values at the resonant frequency are higher than at other frequencies, undesirable deviations at this frequency occur and are also reflected in the value of $L_{n,w}$. In this case, the value of the spectral weighting term remains close to zero, and there is no "double sanction" for resonance.



	A	B
$L_{n,w}$	42 dB	49 dB
$C_{I,50\text{-}2500}$	6 dB	0 dB
$L_{n,w}+C_{I,50\text{-}2500}$	48 dB	49 dB
f_0	80 Hz	400 Hz

A - Kelluva lattia
teräsbetoni
askeläänieriste 30 mm
kelluva laatta
lattianalusmatto
parketti

B - Tavanomainen lattia
teräsbetoni
lattianalusmatto
parketti

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Fig. 6.2.2 Behaviour of spectral weighting term $C_{I,50\text{-}2500}$ with two different types of midsoles.

6.3 Impact soundproofing of a massive slab

A massive slab refers to a simple panel structure with a large surface mass and a low coincidence limit frequency. The normalised impact SPL can be described by sound radiation caused by point excitation directed at the panel. If a standardized tapping machine is used, the normalized impact SPL per third octave band can be estimated from the equation (Rindel, 2018)

(6.3.1)
$$L_n \cong 82 - 10 \cdot \log_{10}\left(m'^2 \frac{\eta}{f_c}\right), \qquad f_c < f < \tfrac{1}{2} f_s$$

where η [kg/m^2] is the total loss coefficient, m' is the surface mass, and f_c [Hz] is the coincidence limit frequency and fs [Hz] is the bending and shear wave limit frequency. These are defined in Ch. **5**. Since the loss factor is frequency-dependent, it also depends on the frequency *of Ln*. Above the critical frequency, doubling the thickness of the panel means a 9 dB decrease in impact SPL.

Above the frequency of $1/2 f_s$, oscillation is mainly caused by shear waves. The approximation equation for the impact SPL of a third octave band is

(6.3.2)
$$L_n \cong 82 - 10 \cdot \log_{10}\left(m'^2 \frac{\eta}{f_c}\right) + 10 \cdot \log_{10}\left(\frac{2f}{f_s}\right), \qquad f > \tfrac{1}{2} f_s$$

In the frequency range of the cutting waves ($f > f_s$), doubling the thickness of the panel means a decrease in the impact SPL of 12 dB.

Fig. **6.3.1** shows the impact SPL of the uncoated reference floor.

f [Hz]	$L_{n,r,0}$ [dB]
100	67.0
125	67.5
160	68.0
200	68.5
250	69.0
315	69.5
400	70.0
500	70.5
630	71.0
800	71.5
1000	72.0
1250	72.0
1600	72.0
2000	72.0
2500	72.0
3150	72.0

Teräsbetoni
140 ± 20 mm

$L_{n,r,0w}$	$C_{I,r,0}$
[dB]	[dB]
78	-11

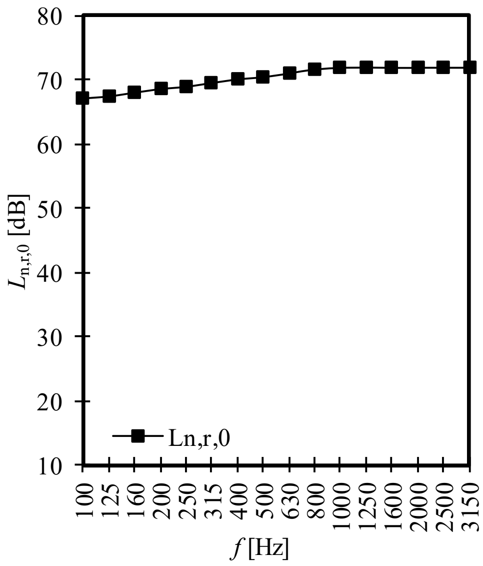


Fig. 6.3.1 Impact SPL expected from a standard reference floor according to ISO 10140-5 (2010).

6.4. Improvement by soft floor coverings

The impact SPL of the massive raw intermediate floor is high because the radiation coefficient is above the limit frequency of coincidence at one (see below). **Ch. 5**). Increasing the thickness is not a cost-effective way to reduce the L_n value. A more effective method is a flexible coating installed on top of the floor to reduce the force exerted on the floor by the hammer of the tapping machine. Soft floor coverings include vinyl, cork and textile carpets.

The mass of the hammer, the elasticity of the elastic coating and the floor below form a mass-spring-mass system, the resonant frequency f_0 [Hz] of which is given by the equation

(6.4.1) $$f_0 = \frac{1}{2\pi}\sqrt{\frac{S_h E_t}{m_h h_t}}$$

where m_h [kg] is the mass of the hammer, S_h [m^2] is the area of the hammer, E_t [Pa, N/m^2] is the modulus of elasticity of the coating, and h_t [m] is the thickness of the coating. For a standardized impact sound device, m_h = 0.500 kg and S_h=700 mm2. The impact SPL on the paved floor begins to decrease above the resonant frequency.

The impact noise level improvement ΔL due to the floor covering *(* Fig. **6.4.1**) is

(6.4.2) $$\Delta L = L_{n,0} - L_n$$

where $L_{n,0}$ [dB] and L_n [dB] are the impact SPLs without and with the coating. In standardized measurements, the so-called standard measurements are used. standard intermediate floor made of 140±20 mm thick reinforced concrete (size 10–12 m^2). The impact SPL improvement number ΔL_w [dB] produced by the floor covering is determined by the difference between the impact sound level figures according to the above measurement results

(6.4.3) $$\Delta L_w = L_{n,0,w} - L_{n,w}$$

Manufacturers of floor coverings indicate the impact sound insulation improvement figures ΔL_w as product values. In this case, the value can be subtracted from the value *of* $L_{n,0,w}$ of a raw reinforced concrete intermediate floor of any thickness, if a reliable value is known. ΔL or ΔLw values measured with a standard concrete intermediate floor should not be applied to wooden intermediate floors.

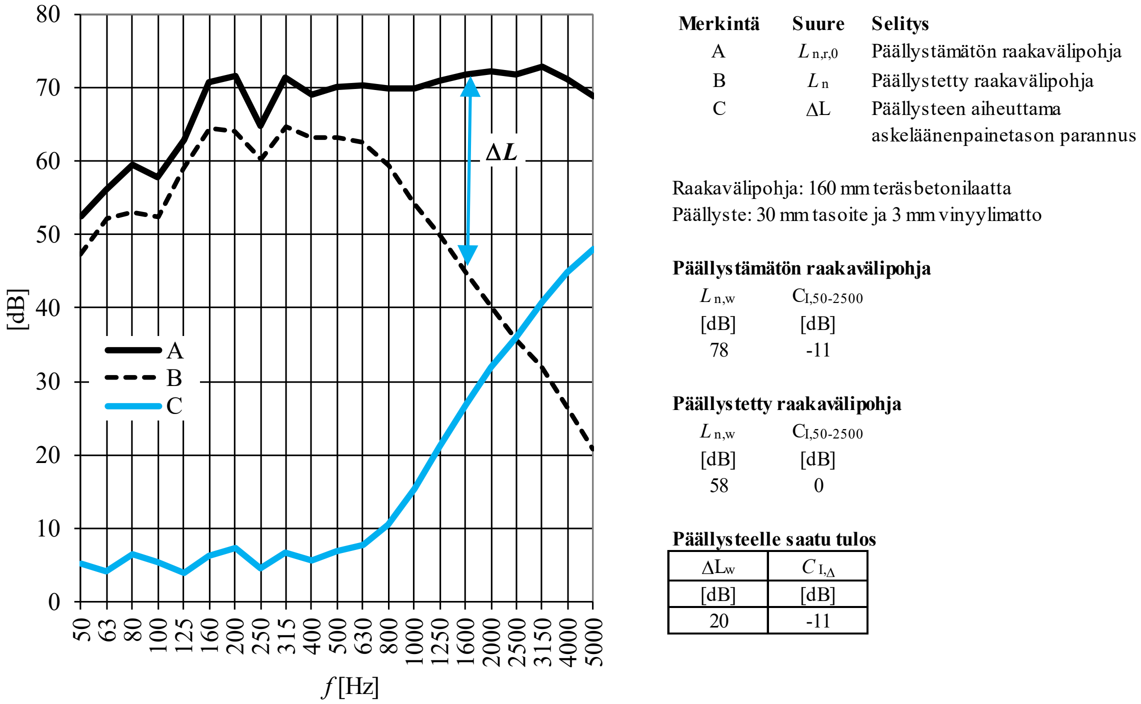


Fig. 6.4.1 Test result of a impact sound improvement for a floor covering.

The improvement produced by the flexible coating per third octave band can be calculated from the equation

(6.4.4) $$\Delta L \cong 40 \cdot \log_{10}\left(\frac{f}{f_0}\right) \qquad f > f_0$$

Below the resonant frequency, there is no improvement:

(6.4.5) $$\Delta L = 0$$

Fig. **6.4.2** shows examples of improvements in impact sound levels for some floor coverings.

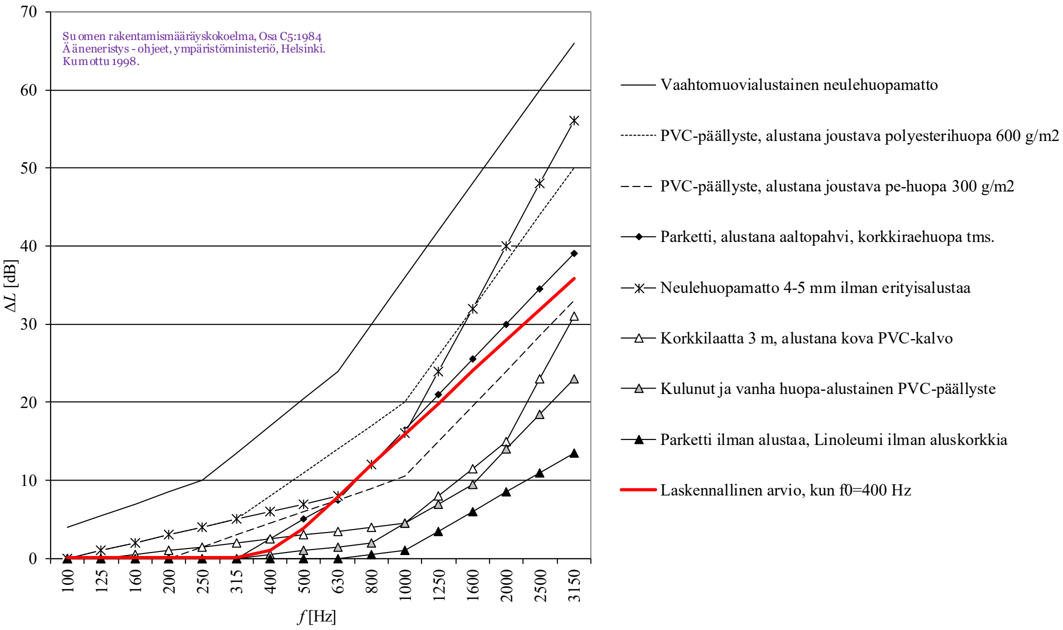


Fig. 6.4.2 Impact SPL improvements produced by some flexible floor coverings, ΔL, as a function of frequency, *f*.

6.5 Floating floor structure

The floating floor structure consists of a load-bearing slab, a flexible layer and a floating panel (Fig. **6.5.1**). An elastic layer is an elastic but panel-like layer. Typical examples are mineral wool impact sound insulation, urethane insulation with water pipe grooves for underfloor heating, or parquet underlay felt. The installation floor can also be flexible, but this will not be discussed here. The floating panel may be a layer of aggregate to be cast in place or a building panel(s). If necessary, a floor covering such as vinyl or parquet underlay felt or parquet is installed on top of the floating board. Parquet underlay felt and parquet (or laminate) mounted on top of the load-bearing slab are also considered floating structures, but have a high resonant frequency of 400–600 Hz, while for more massive floating floor structures it is less than 100–200 Hz.

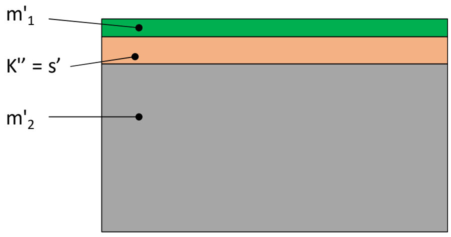


Fig. 6.5.1 A floating floor consists of a load-bearing slab *(m'2*), a flexible layer (s') and a floating panel (*m*'*1*).

The elastic layer is made of elastic material. It is characterized, among other things, by the modulus *K*' [N/m^2] of elasticity, which indicates the stiffness per unit area and unit thickness:

(6.5.1) $$K' = \rho_m c_m^2$$

where *m* [kg/m^3] is the density of the flexible layer and ρ_{cm} [m/s] is the speed of sound in the flexible layer. The dynamic stiffness of the flexible layer per unit area *K*'' [N/m^3] is given by:

(6.5.2) $$K'' = \frac{K'}{d}$$

where *d* [m] is the thickness of the elastic layer. Table **6.5.1** gives some examples of modulus values for materials. It can also be used to calculate the speed of the longitudinal wave (compression wave) of sound in various substances. According to measurements, mineral wool is usually c_m=20–50 m/s.

Table 6.5.1 Values of modulus of elasticity, *K'*. The thickness of the material is not taken into account.

Materiaali	MN/m^2
Ilma (ρc^2) huoneenlämmössä	0.14
Mineraalivilla 50-160 kg/m^3	0.02 - 0.40
Polystyreenivaahto 10-20 kg/m^3	0.3 - 3
Korkki	10-30

If the material of the flexible layer has air pores, *K*'' includes the elasticity provided by both the material, K''_m, and the air spring, K'_0, contained in the flexible layer:

(6.5.3) $$K'' = K''_m + K''_0 = \frac{\rho_m c_m^2}{d} + \frac{\rho_0 c_0^2}{qd}$$

where ρ_0 [kg/m^3] is the density of air and c_0 [m/s] is the speed of sound in air. Porosity *q* is defined **in Chapter 3**.

The most important characteristic for the acoustic operation of a floating floor is its specific frequency f_0 [Hz]:

(6.5.4) $$f_0 = \frac{1}{2\pi}\sqrt{K''\frac{m'_1 + m'_2}{m'_1 m'_2}}$$

where m'_1 and m'_2 [kg/m^2] are the surface masses of the floating panel and load-bearing slab (Fig. **6.5.1**). In most cases, $m'_1 << m'_2$ apply, giving

(6.5.5) $$f_0 = \frac{1}{2\pi}\sqrt{\frac{K''}{m'_1}}$$

From this can be derived the most well-known form of calculating the resonant frequency

(6.5.6) $$f_0 = 159\sqrt{\frac{s'}{m'_1}}$$

where s' [MN/m^3] is the dynamic rigidity of the flexible layer per unit area. This is used because product values are usually indicated by the order of square brackets. Table 6.5.2 gives examples of s' values for some product types.

Table 6.5.2 Dynamic stiffness per unit area determined in accordance with ISO 9052-1 (1989) for some underfloor products and floor coverings.

Tuotetyyppi	Paksuus [mm]	Tiheys [kg/m^3]	s' [MN/m^3]
Kevyt lasivilla 50 mm	50	45	3
Lattianalusvilla 50 mm	50	92	12
EPS levy 50 mm	50	18	12
Lattianalusvilla 13 mm	13	115	16
Parketinalushuopa 2.1 mm	2.1	77	68
Pehmeä tekstiilimatto (asuntokäyttöön)	8.2	470	80
Kova tekstiilimatto (julkisiin tiloihin)	3.8	550	230
Pehmeä vinyylimatto (asuntokäyttöön)	2.4	703	2900
Kova vinyylimatto (julkisiin tiloihin)	2.2	1420	3400

Floating floors can be divided into two types. In a globally resonant situation, the floating panel radiates efficiently throughout its field and thus interacts with the elastic material throughout its field. The floating panel is resonant when $f > f_c$. Rocky massive floating panels usually have an f_c < 300 Hz and are therefore resonant for most of the frequency range. In a locally reacting, i.e. non-resonating, situation, the floating panel interacts with the elastic material only locally, i.e. in the vicinity of the impulse in an area half the size of the bending wave, outside which an acoustic short circuit blocks sound radiation. The tile is locally responsive when $f < f_c$. Floating panels implemented on construction boards usually have f_c > 1000... 2000 Hz and are locally responsive in the case of the most significant frequency range in terms of impact sound insulation (below 1000 Hz).

In the case of a resonant floating panel, the impact sound level improvement ΔL [dB] is (Schiavi, 2018)

(6.5.7) $$\Delta L = -15 \cdot \log_{10}\left(\sqrt{\frac{1+\eta_m\left(\frac{f}{f_0}\right)^2}{\eta_m\left(\frac{f}{f_0}\right)^2 + \left[1-\left(\frac{f}{f_0}\right)^2\right]^2}}\right)$$

and non-resonant floating slab

(6.5.8) $$\Delta L = -20 \cdot \log_{10}\left(\sqrt{\frac{1+\eta_m\left(\frac{f}{f_0}\right)^2}{\eta_m\left(\frac{f}{f_0}\right)^2 + \left[1-\left(\frac{f}{f_0}\right)^2\right]^2}}\right)$$

where η_m is the loss coefficient of the flexible layer. It typically receives values between 0.05 and 0.30. It can be determined in the process of determining the dynamic rigidity of the flexible layer from the half-value width of the resonant frequency.

The floating floor design delivers a significant impact sound insulation improvement at frequencies 1.5 times higher than the resonant frequency. If the loss coefficient is low, a decrease in impact sound insulation is observed at the resonant frequency, in an area about an octave band wide. The impact sound insulation improvement ΔL_w produced by uncoated floating floors placed on top of massive load-bearing structures are usually greater than 25 dB.

Covering a floating floor with a flexible floor covering reduces the impact sound level number less than the impact sound insulation improvement ΔL_w measured for the floor covering in connection with a massive load-bearing structure. In the case of the most efficient floating floors, the effect of the floor covering on the impact sound level reduction is zero.

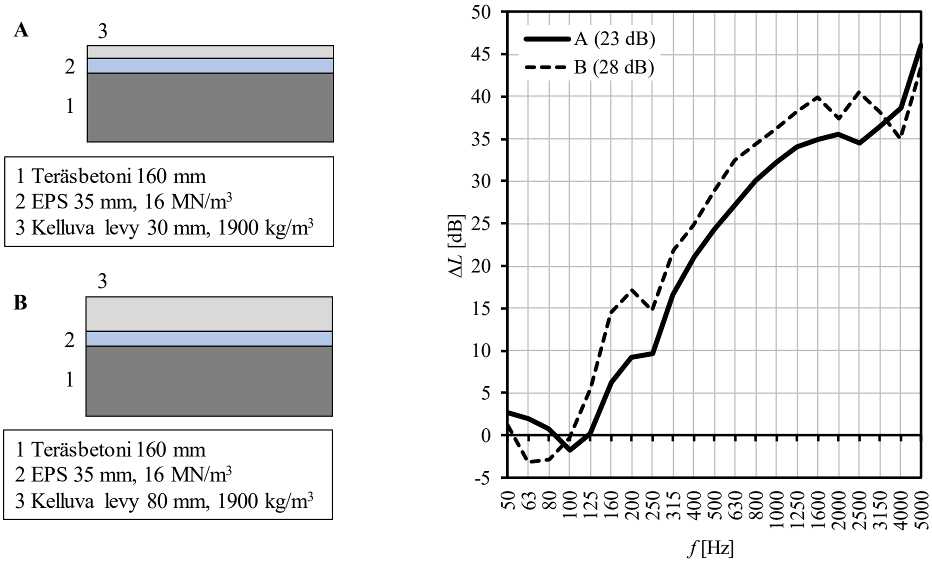


Fig. 6.5.2 Example of an improvement in impact SPL ΔL over the raw midfloor (1) by two slightly different floating floor structures. ΔL_w values are enclosed in parentheses.

6.6 Alternative impact sound stimuli

A impact sound device produces sound that does not correspond to the most common impact sounds in housing, such as normal walking, children bouncing, things hitting the floor, dragging things on the floor and playing on the floor. Fig. **6.6.1** compares the spectra of some impact sounds.

There are three standard options for a impact sound device: a modified tapping machine, a rubber ball and a bang machine. The modified impact tone instrument is one with soft-tipped hammers as shown in Fig. **6.1.1.** Although the spectrum it produces is more reminiscent of the sound of walking, the SPL it produces is not sufficient to measure heavy structures. A rubber ball and a banging machine (Fig. **6.1.1**) produce a sound similar to children's bouncing. The banger is large and difficult to use and cannot be bought in Europe. The rubber ball has become more common among researchers, and the sound it produces is used as such in psychoacoustic studies, for example, to represent children's bouncing. The rubber ball has already been taken into account in ISO standards (ISO 10140-5, ISO 10140-3, ISO 16283-2, ISO 717-2), so the method is presented below.

In laboratory measurement of the floor structure, the maximum impact SPL Li,Fmax caused by the rubber ball is determined. To do this, the rubber ball produces the desired impact sound by dropping it from a height of 1 m, 1.00 m from a stick steal, onto the floor to be examined at 4 different excitation points j and the SPL measurement is carried out in the reception room at each excitation point at four positions in the k octave bands 63−500 Hz or in the third octave bands 50−630 Hz. The values obtained at each excitation point *j* are averaged according to the energy principle by the equation

(6.6.1) $$L_{i,F\max,j} = 10 \cdot \log_{10}\left(\frac{1}{m}\sum_{k=1}^{m} 10^{L_{F\max,k}/10}\right)$$

where $L_{\mathrm{Fmax,k}}$ [dB] is the maximum noise pressure level corrected for background noise at k *(*k = *1... 4)*. The final result $L_{\mathrm{i,Fmax}}$ is obtained by averaging according to the energy principle the mean values of each excitation site using the equation

(6.6.2) $$L_{i,F\max} = 10 \cdot \log_{10}\left(\frac{1}{n}\sum_{j=1}^{n} 10^{L_{i,F\max,j}/10}\right)$$

where $L_{\mathrm{i,Fmax,j}}$ [dB] is the spatial mean of the maximum SPL at excitation position *j* (j=1... 4). From the final result $L_{\mathrm{i,Fmax}}$ determine the single digit value, the maximum value of the A-weighted impact SPL, LiA,Fmax [dB], according to ISO 717-2 (2020), by calculating the total A-weighted SPL from $L_{\mathrm{i,Fmax}}$ values between 50–630 Hz. Fig. **6.6.3** shows an example of the measurement result. In general, the result is normalized to volume V=50 m^3 and reverberation time T=0.5 s ($L_{\mathrm{i,Fmax,V,T}}$), so that results obtained for the same structure from different environments can be compared. However, the normalisation procedure is not discussed here.

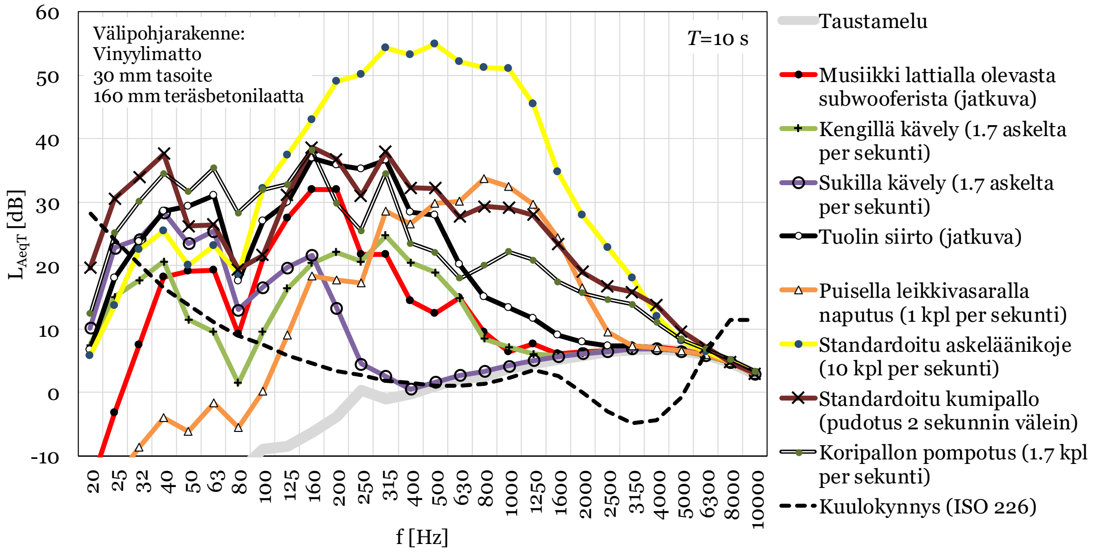


Fig. 6.6.1 A-weighted SPLs equivalent to the upper surface of the intermediate floor structure in the room below. Sounds that exceed the hearing threshold are perceptible. The sound of impulsive impact sounds should therefore be evaluated using the Fast time-weighted maximum. There are no guideline values for SPLs for actual impact sounds, so they are only of research significance.

f [Hz]	$L_{\mathrm{i,Fmax,V,T}}$ [dB]
50	67.8
63	58.6
80	53.8
100	55.7
125	52.2
160	60.2
200	61.5
250	63.7
315	56.6
400	54.6
500	52.5
630	48.9
$L_{\mathrm{iA,Fmax,V,T}}$	59.2

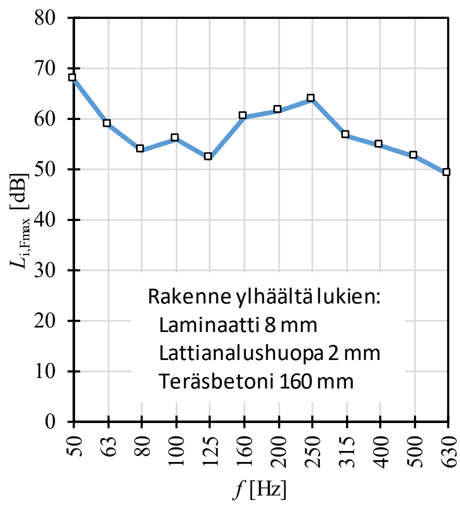


Fig. 6.6.2 Measurement result obtained with a rubber ball for a concrete intermediate floor structure.

6.7 Lightweight intermediate floor structures

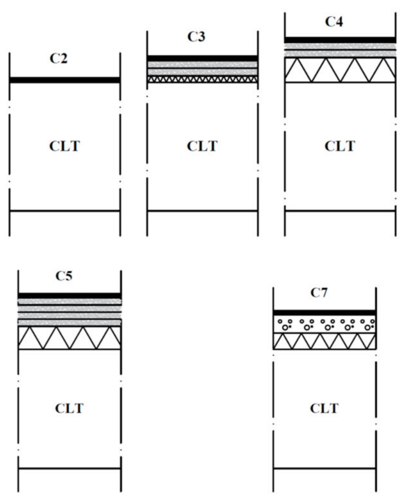
Lightweight intermediate floors are defined as structures whose load-bearing structure consists of steel profiles, wooden joists, CLT panels, LVL panels, or similar structures. The mass of such intermediate floor structures, including all structural layers, is usually less than 150 kg/m^2. Impact sound insulation of lightweight intermediate floor structures is based on the mass and cavitations of separate layers of sheets. When aiming for low impact sound levels, lightweight midsoles usually consist of several structural layers, which include:

- flexible floor coverings
- tiles or panels floating on stretch floor insulation
- installation floors (flexible or rigid connection to the load-bearing structure)
- suspended ceiling (flexible or rigid connection to load-bearing structure)

The impact sound insulation of a lightweight intermediate floor is affected by the same factors as the airborne sound insulation of a lightweight wall. Figures **6.7.1-5** show laboratory measurement results showing how different factors affect the impact sound insulation of the wooden intermediate floor. A particularly striking improvement effect is observed in the stretch suspended ceiling and suspended wire ceiling.

Walking and other impacts to the floor cause not only audible sound, but also vibrations below the hearing range at frequencies from 6 to 20 Hz. Such vibrations are perceived as vibrations. The vibration of light structures from walking is stronger than that of massive structures, since a blow from a person with its weight causes greater movement in a light structure than in a massive structure.

	$L_{n,w}$ [dB]	$L_{n,w}+C_{I,50\text{-}2500}$ [dB]	R_w [dB]
C2	65	65	42
C3	55	59	54
C4	47	53	61
C5	45	51	61
C7	60	63	57



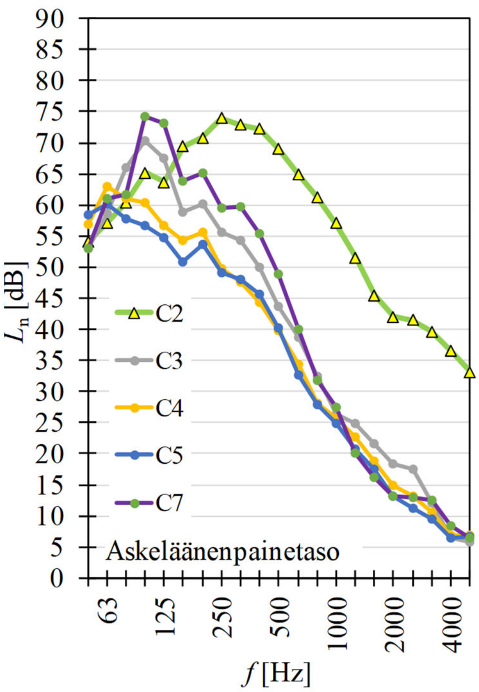
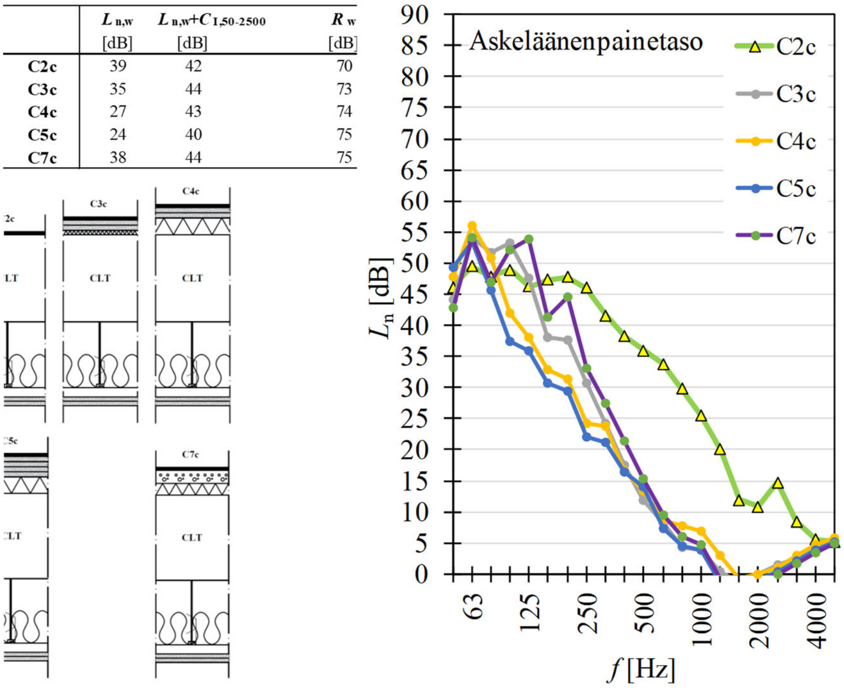
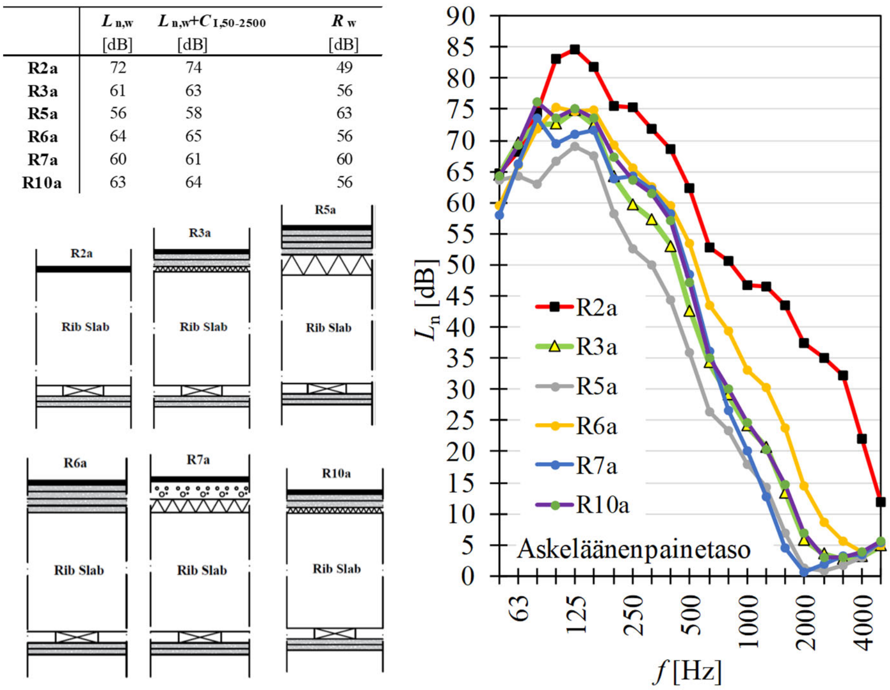


Fig. 6.7.1. Normalized impact SPL as a function of frequency when the load-bearing structure is 260 mm thick solid wood slab (CLT) and the tile coatings are subject to changes (Hongisto et al, 2023).



	$L_{n,w}$ [dB]	$L_{n,w}+C_{I,50\text{-}2500}$ [dB]	R_w [dB]
C2c	39	42	70
C3c	35	44	73
C4c	27	43	74
C5c	24	40	75
C7c	38	44	75

Fig. 6.7.2. Normalized impact SPL as a function of frequency when the load-bearing structure is a 260 mm thick solid wood slab (CLT) on a suspended wire suspended ceiling and the slab coverings are subject to changes (Hongisto et al, 2023).



	$L_{n,w}$ [dB]	$L_{n,w}+C_{I,50\text{-}2500}$ [dB]	R_w [dB]
R2a	72	74	49
R3a	61	63	56
R5a	56	58	63
R6a	64	65	56
R7a	60	61	60
R10a	63	64	56

Fig. 6.7.3. Normalized impact SPL as a function of frequency when the load-bearing structure is a 370 mm thick wooden open box slab (Rib Slab) with a rigidly coated suspended ceiling and the slab coverings are subject to changes (Hongisto et al, 2023).

	$L_{n,w}$ [dB]	$L_{n,w}+C_{I,50\text{-}2500}$ [dB]	R_w [dB]
R2b	55	55	62
R3b	46	47	70
R5b	39	41	75
R6b	44	44	71
R7b	47	47	72
R9b	47	47	68

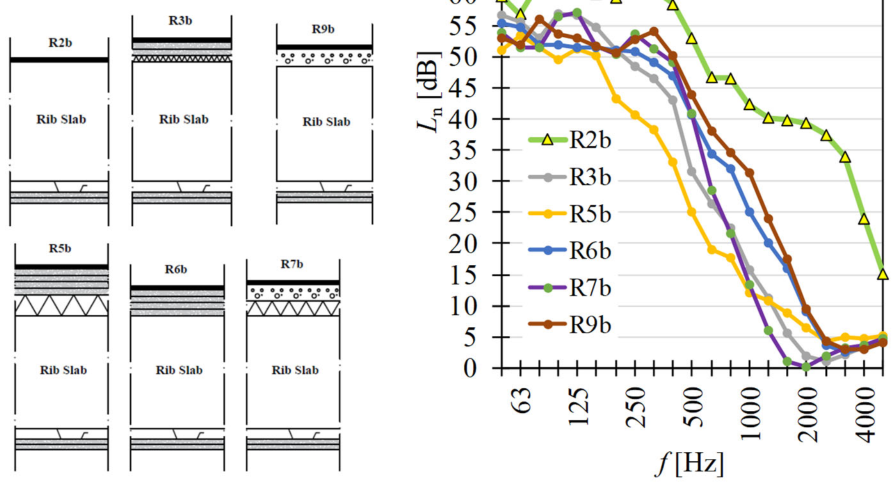


Fig. 6.7.4. Normalized impact SPL as a function of frequency when the load-bearing structure is a 370 mm thick wooden open box slab (Rib Slab) with a flexibly coated suspended ceiling and the slab coverings are subject to changes (Hongisto et al, 2023).

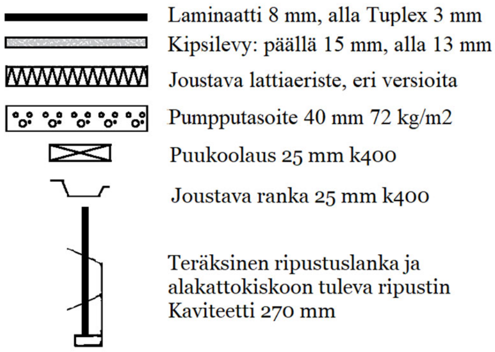


Fig. 6.5.5. Explanation of the structural layers used in Figs. 6.7.1 to 6.7.4.

6.8 Installation floors

The impact sound insulation of the installation floor (Fig. **6.8.1**) depends on the mass of the surface structure, the height of the cavity between the surface structure and the load-bearing structure, the absorption material of the cavity and the dynamic rigidity (spring constant) of any vibration insulators placed between the supporting structures supporting the surface structure and the load-bearing structure.

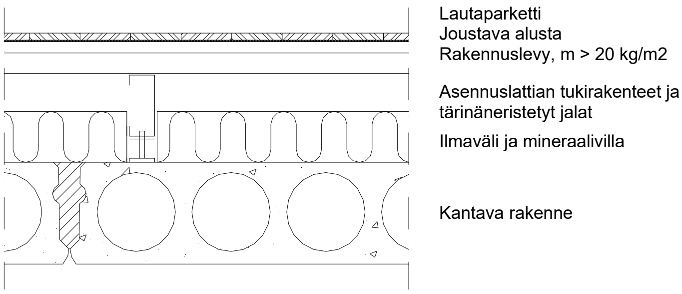


Fig. 6.8.1 The structural principle of the installation floor. (Source: RIL 243-1:2007)

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7 VENTILATION NOISE

7.1. Noise sources

Building HVAC equipment (heating, water, air conditioning, electricity, automation) includes, for example, elevators, water and sewage equipment, compressors, ventilation equipment, cooling equipment, heating equipment and various electrical appliances. Central vacuum cleaners, carpet vacuum cleaners and house laundry equipment, such as washing machines, centrifuges, drying fans and mantels, are also equated to them.

This chapter focuses on noise abatement of mechanical ventilation, since ventilation is the most common stage in construction that requires acoustic calculations. The chapter focuses on sound attenuation and the calculation of sound levels. The ventilation system causes noise, for example, as follows:

- the air sound from the ventilation fan travels through the ductwork to the room, through the casing to the engine room or through the ductwork to the outside air;
- the flow of air in ductwork, bends and branches, terminal and regulating devices generates flow noise;
- Leaks in the ductwork cause so-called leaks. leakage noise;
- The sound from the engine room can be carried into the apartment through the structures.

In addition, ventilation ductwork is a sound pathway that must be taken into account in the following respects, among others:

- Through a fresh air valve placed on the façade, environmental noise is carried into the room, which must be taken into account in the soundproofing dimensioning of the façade;
- air sound is carried between rooms through ventilation ductwork from the terminal device to the terminal device (secondary displacement of air sound);
- unsealed penetrations in ducts and pipes can impair the airborne sound insulation of the surrounding structure (sound leaks);
- a strong sound of ventilation inside the duct can carry through the duct wall into the room;

If the ventilation system has air conditioning and the refrigeration units are located in the building, there are separate sound and vibration insulation challenges associated with refrigeration units, which are not covered here.

Ventilation systems can be gravitational or mechanized. The advantage of gravity systems is the noiselessness of the air flow. Noise problems can be caused by environmental noise carried through fresh air valves.

This chapter focuses on mechanical systems, where sound control plays a key role immediately after the air distribution has been dimensioned. Mechanized systems include:

1. Central exhaust ventilation
2. Central supply and exhaust ventilation
3. Apartment-specific supply and exhaust ventilation

Centralised systems usually have a common trunk ductwork system to which all rooms or apartments are connected. Alternatively, a trunk duct is brought into each apartment directly from the engine room. The latter is more expensive and takes up more space, but allows for more individual adjustment and allows for better airborne sound insulation between apartments.

Fig. **7.1.1** shows an example of a ventilation plant, its sound sources and sound pathways.

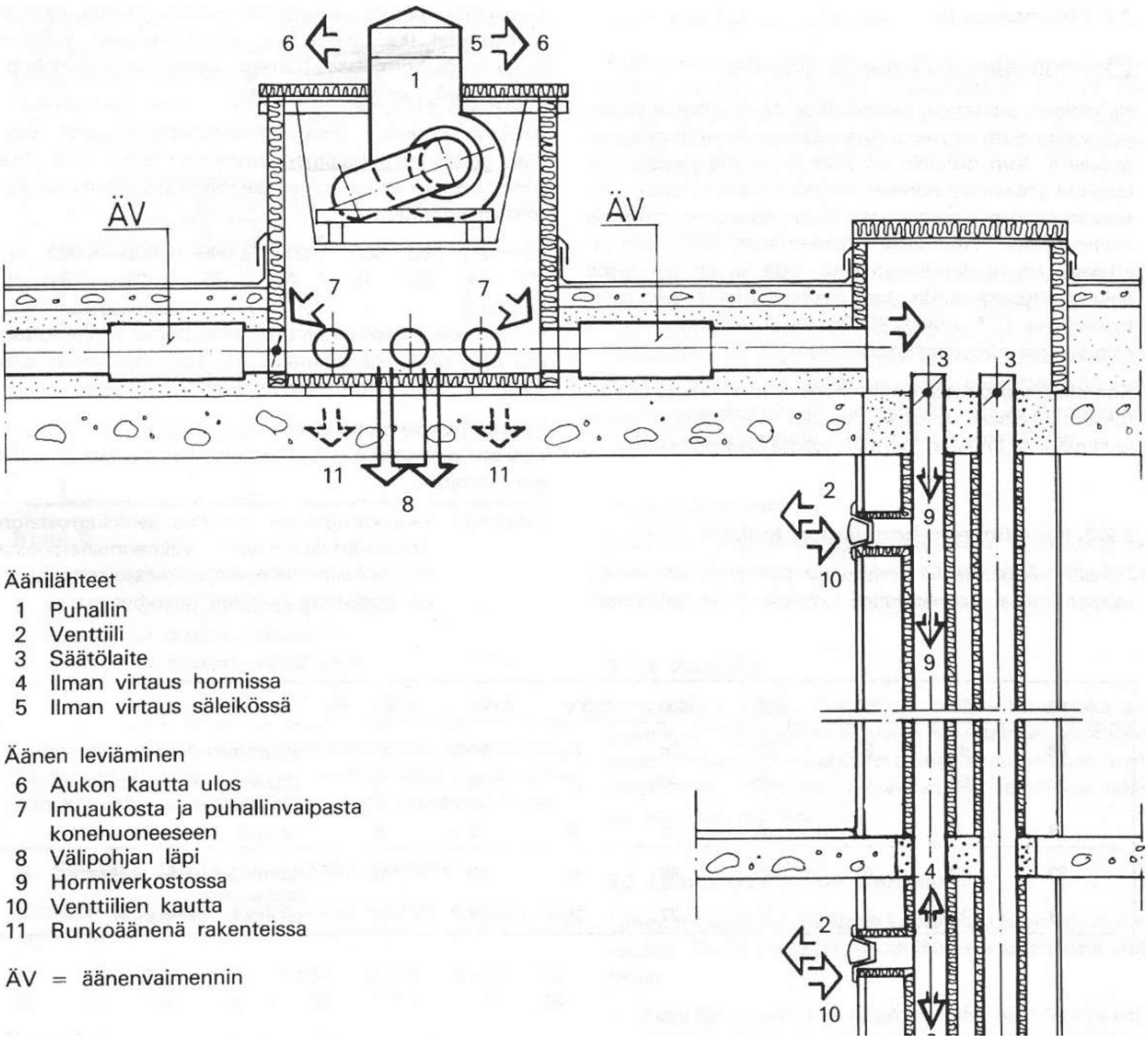


Fig. 7.1.1. Co-ducted air conditioning plant. Only exhaust ventilation is shown in the figure. (Finnish National Building Code C6:1984)

7.2 Sound level calculations

Sound calculations for ventilation determine the total A-weighted SPL in the room under consideration. The sound sources considered in the calculations are:

1. fans (inlet and exhaust fan separately);
2. adjusting devices between the fan and the room under consideration in each ductwork;
3. terminals in the room;
4. Engine room: the air sound carried through structures from the engine room.

In addition, noise is generated due to poor design or contracting, e.g. due to air flow in ductwork, duct branches, bends and air leaks. There are no output values for these declared by component manufacturers, so these sound sources are not processed.

Sound absorption factors include:

1. Distribution of noise in the duct branches D_d
2. Mufflers D_s
3. Terminal attenuation D_t
4. Room cushioning D_r

Subscripts stand for division, silencer, terminal, and room. In addition, the sound is attenuated as it moves through the ductwork (absorption or penetration of walls) and bends. These attenuation factors are weak and are usually left as a safety margin.

The sound level calculation is made for each room by first calculating separately the SPL of all sound sources 1–4 in the octave bands 63–8000 Hz. The partial results are summed up and compared with the target value.

The SPL of the fan, $L_{p,f}$ [dB] in the room, is calculated by the equation

(7.2.1) $$\boxed{L_{p,f} = L_{W,f} - D_d - D_s - D_t + D_r}$$

where $L_{W,f}$ [dB] is the sound power level of the fan. The sound power level of the fan is selected from the product brochure. To do this, you need to know the pressure drop in the fan and the total air volume.

The SPL of the damper, $L_{p,d}$ [dB] in the room, is calculated by the equation

(7.2.2) $$\boxed{L_{p,d} = L_{W,d} - D_s - D_d - D_t + D_r}$$

where $L_{W,f}$ [dB] is the sound power level of the damper. The sound power level of the control device is selected from the product brochure. To do this, you need to know the pressure drop in the fan and the total air volume.

The SPL in the room of the terminal unit, either inlet or outlet unit), is calculated using the equation

(7.2.3) $$\boxed{L_{p,t} = L_{W,t} + D_r}$$

where $L_{W,t}$ [dB] is the sound power level of the terminal. The sound power level of the terminal is selected from the product brochure. To do this, you need to know the pressure drop in the fan and the total air volume.

Product brochures for control and terminal equipment usually give the noise emission in the form Lp10A, which means the A-weighted SPL in a room with an absorption area of 10 m2. The value is converted to the A-weighted sound power level *LWA* by the equation

(7.2.4) $$L_{WA} = L_{p10A} + 4$$

7.3 Fan sound

The supply fan brings fresh air into the building and the exhaust fan removes the used air from the building. Fans emit both mechanical noise and the sound of air flow.

Fan manufacturers indicate the sound power level of the fan as a function of air volume and total pressure in the suction and pressure duct and over the casing to the environment (engine room). An example of the presentation of product values is shown in Fig. **7.3.1.** When the operating point is known, the A-weighted sound power level L_{WA} and the rotational speed range are read from the graph. The unweighted sound power level $L_{WZ,okt,i}$ in the octave band i is given by:

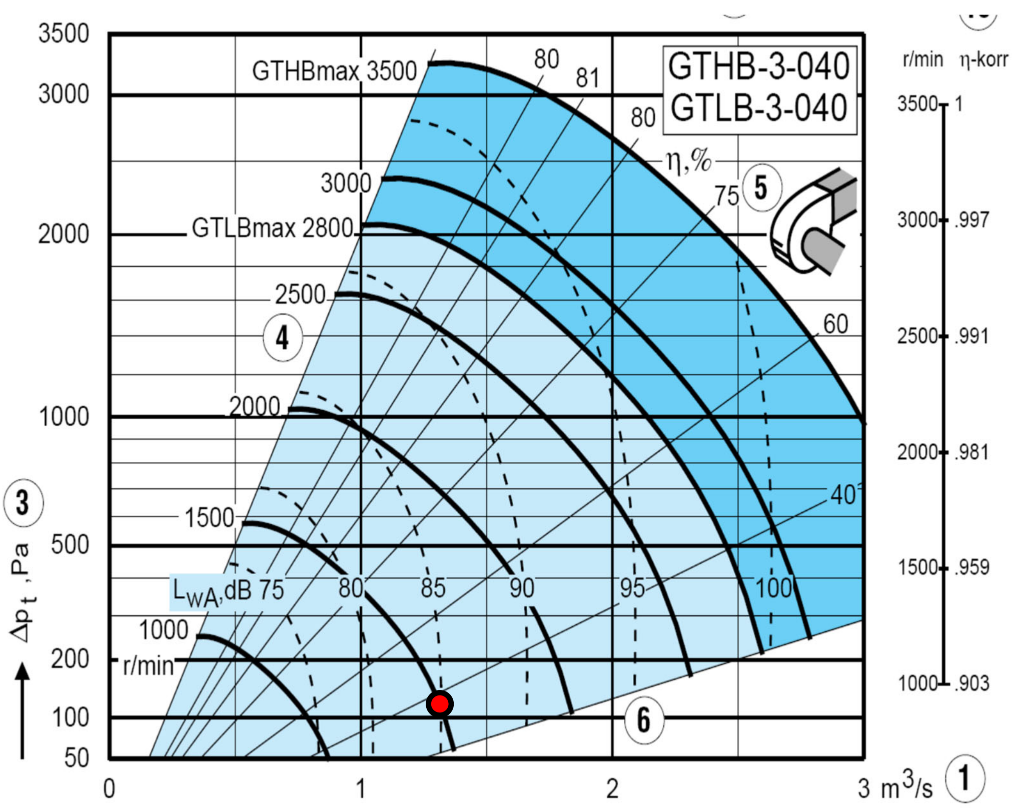
(7.3.1) $$L_{WZ,okt,i} = L_{WA} + K_{okt,i}$$

where $K_{okt,i}$ [dB] is obtained from product values, taking into account the rotational speed range, if necessary.

The integrated inlet exhaust unit includes two fans (supply and exhaust fans) as well as various components (filter, heat recovery, cooling, heating, humidification, muffler). The integrated inlet exhaust machine includes four channels:

- fresh air intake (fresh air sucked from outside);
- fresh air pressure duct (fresh air entering the room);
- waste air intake channel (waste air sucked from rooms);
- waste air pressure duct (exhausted waste air);

An apartment-specific integrated supply exhaust unit can include a fifth channel for sucking hot air from the fireplace. In integrated machines, the sound power level indicated by the product value includes the sound absorption caused by the components.



Äänitie (s)	Pyörimisnopeusalue r/min	Korjaus K_{okt}, dB Oktaavikaista, keskitaajuus, Hz 63	125	250	500	1 000	2 000	4 000	8 000	$L_{WA(s)} - L_{WA}$ dB	$L_{Wt(s)} - L_{WA(s)}$ dB
Painekanavaan (1)	0 – 964	0	5	2	-3	-6	-9	-14	-18	0	8,2
	965 – 1928	-2	-1	3	-3	-6	-9	-14	-17	0	6,4
	1929 – 3200	-3	-4	-4	-1	-6	-8	-13	-16	0	4,1
Imukanavaan (2)	0 – 964	4	3	0	-3	-4	-9	-12	-14	0,4	7,8
	965 – 1928	2	-1	0	-3	-5	-8	-10	-13	0,3	6,2
	1929 – 3200	-2	-5	-6	-2	-4	-7	-9	-14	0,8	3,3
Ympäristöön, kanavaan liitetty puhallin (3)	0 – 964	-8	-5	-6	-8	-11	-15	-22	-33	-6,1	6,1
	965 – 1928	-10	-8	-6	-10	-12	-16	-25	-36	-7,3	5,8
	1929 – 3200	-12	-14	-11	-8	-10	-16	-24	-35	-6,4	3,1
Puhaltimen paine-aukkoon, vapaasti puhaltava puhallin (4)	0 – 964	-9	0	0	-3	-6	-9	-14	-18	-0,6	5,5
	965 – 1928	-13	-6	1	-3	-6	-9	-14	-17	-0,5	4,5
	1929 – 3200	-17	-9	-6	-1	-6	-8	-13	-16	-0,1	2,3

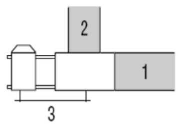
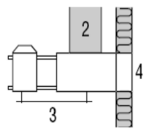
Äänitien kuvaus	Koejärjestely
1 = imukanavaan 2 = imukanavaan 3 = ympäristöön (kanavaan liitetty puhallin)	
4 = puhaltimen paineaukkoon (vapaasti puhaltava puhallin)	

Fig. 7.3.1 Example of indicating the product value of a ventilation fan (Fläktwoods Centrimaster GT-3). At the red operating point, the air volume is 1.3 m3/s, the pressure difference between the intake and pressure duct is 120 Pa, the rotational speed is 1500 rpm and the A-weighted sound power level is 85 dB.

7.4. Sound of the adjusting device

Control devices are used to reduce the pressure in the ductwork. The adjustment device can be in a standard setting or it can be automatically adjusted without need. Control devices include a damper and an iris regulator.

There is usually high pressure in the trunk duct leaving the fan. From the trunk duct, air branches into different zones (e.g. floors or apartments) along branch ducts. Usually, the branch duct has a control device first to control the total air volume entering the zone and ventilation noise. From the branch duct, air branches into the terminal organs along the terminal ducts. The end ducts may also have a control device if the air volume is to be adjusted, for example, according to the degree of use.

Adjusting devices are usually noisy and are usually followed by a muffler. An example of the display of sound data from the adjusting device is shown in Fig. **7.4.1.** When the operating point is known, the A-weighted SPL

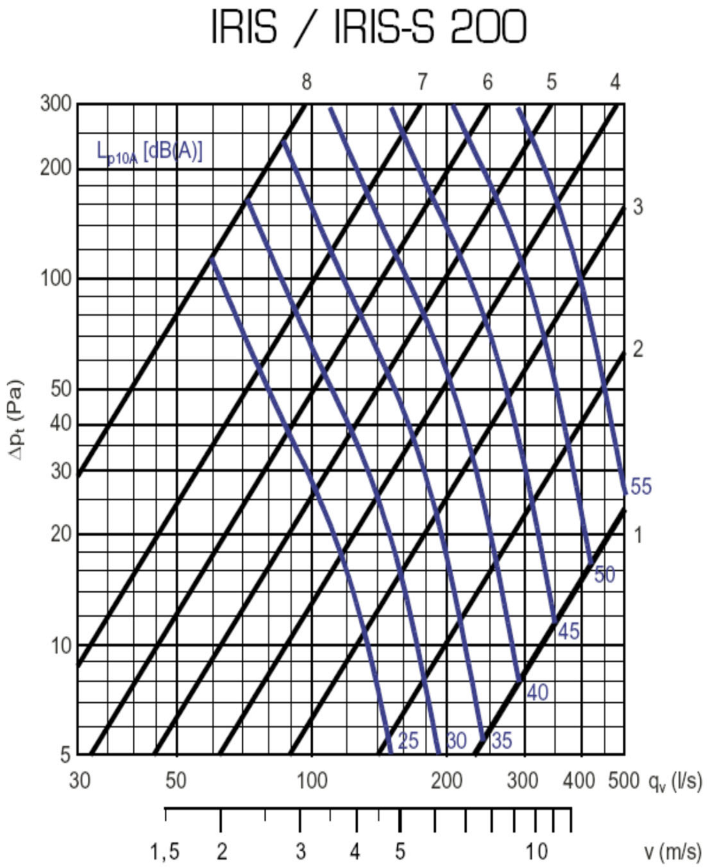
L_{p10A} [dB] shall be read from the diagram.The unweighted sound power level LWZ,okt,i in the octave band i is given by:

(7.4.1) $$L_{WZ,okt,i} = L_{p10A} + K_{okt,i}$$

where $K_{okt,i}$ [dB] is obtained from the audio data of the product.

The diagram uses the SPL to enable the ventilation designer to assess the suitability of the device and/or the need for a silencer in connection with the control device at the design stage more easily than if the diagram included a sound power level.

The regulator also makes noise through its mantle into the environment. In the sound data of the adjusting devices, this sound path is not always presented.



Äänen tehotaso L_W

IRIS	KORJAUS K_{okt} (dB)							
	Oktaavikaistan keskitaajuus (Hz)							
	63	125	250	500	1000	2000	4000	8000
80	10	16	12	9	5	-1	-6	-23
100	25	21	16	9	4	-6	-12	-25
125	17	17	13	7	1	-4	-6	-17
150	21	20	14	8	0	-6	-16	-29
160	19	18	14	6	-1	-6	-13	-25
200	20	17	12	5	-2	-5	-14	-26
250	16	12	8	3	1	-4	-17	-32
315	24	12	5	0	1	-2	-13	-27
400	15	9	6	2	-1	-4	-9	-13
500	14	7	4	1	-1	-4	-8	-11
630	15	7	3	2	-1	-5	-9	-11
800	9	5	3	3	-1	-6	-10	-13
Tol.±	6	3	2	2	2	2	2	3

Fig. 7.4.1 Example of reporting sound data of a control device (Fläktwoods IRIS-S 200).

7.5 Terminal sound

Usually, there is a functional terminal at the interface between the ventilation ductwork and the room in order to achieve the desired supply or exhaust air volume, visual appearance, and sound level in the room.

Terminal equipment includes, for example, grille, valve, diffuser, nozzle duct and active chilled beam (Fig. **7.5.1**). A separate terminal is not always needed, but the channel can also end in the room as is.

Grating-type terminals (or in a situation without terminals) do not have the possibility to adjust the airflow, in which case the adjustment is carried out by means of adjusting devices located in the duct. Usually, the terminals have an adjustment option and the terminal is adjusted during the installation phase so that it achieves the desired air volume and pressure drop.

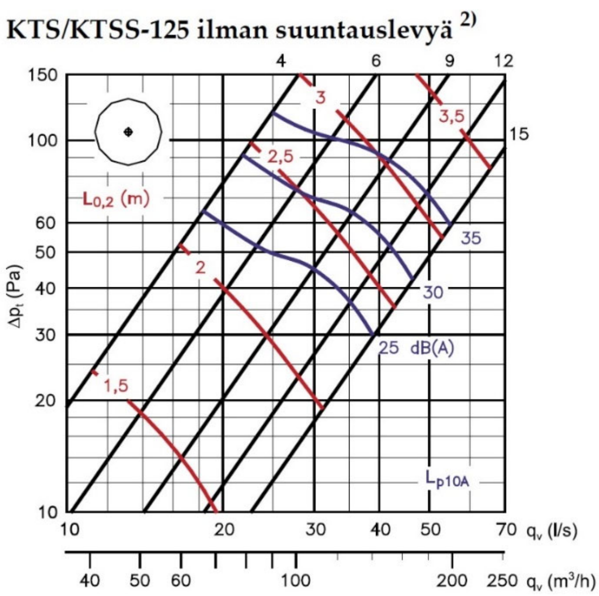
An example of the product value of a terminal equipment is shown in Fig. **7.5.2.** When the operating point is known, the A-weighted SPL L_{p10A} [dB] shall be read from the diagram. The unweighted sound power level $L_{WZ,okt,i}$ in the octave band i is given by:

(7.5.1) $$L_{WZ,okt,i} = L_{p10A} + K_{okt,i}$$

where $K_{okt,i}$ [dB] is obtained from product values.

For terminals, sound absorption is also indicated. This will be discussed later.

Fig. 7.5.1 Top exhaust air valve. The shape of exhaust air valves varies less than that of supply air valves. The most typical supply air valves look pretty much the same as the exhaust air valve in the picture. If the supply air is to be directed more efficiently, diffusers are used, for example. At the bottom there is a nozzle, cone, vortex and wall diffuser (Fläkt Woods Oy 2006).



Äänen tehotaso Lw

Ilman suuntauslevyä

KTS KTSS	KORJAUS K_{okt} (dB) Oktaavikaistan keskitaajuus (Hz)						
	125	250	500	1000	2000	4000	8000
100	-2	2	1	-1	-4	-5	-11
125	4	5	3	-1	-11	-17	-29
160*	7	6	3	-2	-11	-19	-32
Tol.+/-	3	2	2	2	2	2	3

* vain KTS

Äänen tehotasot oktaavikaistoittain saadaan lisäämällä äänen kokonaispainetasoon L_{p10A}, dB(A), taulukossa esitetyt oktaavikaistojen korjaukset K_{okt} seuraavan kaavan mukaan:

$$L_{Wokt} = L_{p10A} + K_{okt}$$

Äänenvaimennus ΔL

KTS	Äänenvaimennus ΔL (dB)							
KTSS	63	125	250	500	1000	2000	4000	8000
100	22	18	13	11	9	8	7	8
125	20	16	11	9	9	7	6	5
160*	18	14	10	9	9	7	6	6
Tol.+/-	6	3	2	2	2	2	2	3

* vain KTS

Keskimääräinen äänenvaimennus ΔL kanavasta huoneeseen sisältää liittyvän kanavan päätevaimennuksen kattoasennuksessa.

Fig. 7.5.2 Example of presenting the product value of a supply air unit (Fläktwoods KTS 125). It is a conventional disc valve in which the air volume can be adjusted by rotating the valve disc. The valve opening adjustment in millimeters is represented by straight lines. Of the openings, s = 4 mm is the smallest value (closed) and s = 15 mm is the maximum value (open). The sound attenuation value of the terminal is used for the sound calculation of the control device and fan, but not for the sound calculation of the terminal.

7.6 Airflow sound

The air flow in the ductwork generates sound because the flow is not laminar but has vortices and the flow rate is different at different points in the cross-section. In smooth-surfaced ducts, the sound of air flow is particularly low.

The airflow causes noise especially when it hits a point of discontinuity, such as a cross-sectional change, branch, bend, control device or terminal. The more pressure drop the obstacle causes, the louder the sound is generated.

In a constant flow field with constant geometry, the sound power level of the flow sound can be estimated by the formula

$$L_w = 10 \cdot \log_{10}(S) + 10 \cdot \log_{10}(v^n) + L_0 \tag{7.6.1}$$

where S [m^2] is the area of the flow cross-section, v [m/s] is the air flow rate and L_0 [dB] is the constant specific sound power level due to the geometry of the section under consideration. The value of the specific sound power level can be determined by measuring the sound power level at a single flow rate. Integer n indicates how the sound power level is proportional to the flow rate. For turbulent flow (low speed) n=*5* and for turbulent flow n=*6* (high speed).

It should be noted that there is no frequency in equation **7.6.1.** Thus, the shape of the flow sound spectrum does not depend on the flow rate.

The equation is not used to make absolute sound development calculations. It is applied to assess how changes in flow rate and surface area affect the amount of flow sound. Fig. **7.6.1** shows an example calculation of how a change in surface area or flow rate affects the sound power level.

In a properly designed air conditioning plant, the flow rates in the exposed ductwork are so low that the sound generated by the flow in the ducts is insignificant. Table **7.6.1** shows the recommended flow rates in the ductwork according to the target sound level.

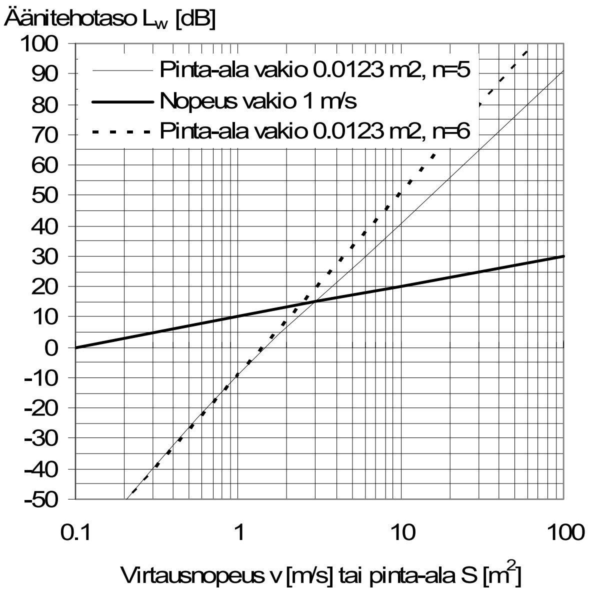


Fig. 7.6.1 Sound power level with change in flow rate or surface area one being constant. $L_0=10$ *dB.*

Table 7.6.1 Maximum recommended air velocity in duct [m/s] when aiming for a sound level below the guideline value.

	Ohjearvo L_{Aeq} [dB]		
	25	**30**	**35**
Kantikkaat kanavat			
Päätekanava (pääte-elimeen saapuva kanava)	2.5	3.0	4.0
Haarakanava (kanava josta päätekanava haaroittuu)	4.0	5.0	6.0
Pyöreät kanavat			
Päätekanava	3.5	4.0	5.0
Haarakanava	5.0	6.5	8.0

Lähde: Suomen rakentamismääräyskokoelma, osa C6:1984

7.7. Muffler silencer

The most significant part of fan noise is usually suppressed by the main mufflers integrated into the ventilation unit, i.e., primary mufflers. Primary mufflers are usually lamellar dampers, in which air passes through the gaps between the absorbent panels (baffels), which are usually 10–20 cm wide. If the ventilation unit does not have integrated mufflers, separate mufflers are usually needed to dampen the fan noise so that the frame and branch ductwork is not too noisy.

Duct silencers, i.e., secondary silencers, are installed in the ductwork. They are either cylindrical or ladder, with a height equal to the channel but lateral absorption material (Fig. **7.7.1**). An example of silencer silencing values is given in Fig. **7.7.2.**

Fig. 7.7.1. (a) Straight lamella silencer, (b) lamellar silencer with 90° angle, (c) cylindrical duct silencer. The lamellas are an absorbent material. In a direct muffler, the absorption material is on the walls. Usually, the absorption material is protected, for example, with perforated sheet metal to prevent the absorption material from tearing in the air flow or sweeping the ductwork.

Äänenvaimennus ΔL

Pituus 600 mm

KLT	ÄÄNENVAIMENNUS ΔL (dB)							
	Oktaavikaistan keskitaajuus (Hz)							
	63	125	250	500	1000	2000	4000	8000
100	9	17	19	36	47	44	40	31
125	11	15	16	35	44	45	42	32
160	15	13	12	32	40	43	37	26
200	15	6	11	23	30	39	29	22
250	12	4	8	17	24	28	15	16
315	5	3	6	15	20	18	14	14
400	5	2	5	15	18	16	15	19
Toler.±	6	3	2	2	2	2	2	3

Fig. 7.7.2 Example of silencing values of a cantilever.

7.8. Sound attenuation of the terminal equipment

The sound absorption of the terminal consists of four factors:

1. termination attenuation of the terminating channel (**chapter 7.15**): a strong change in impedance causes a reflection when the channel ends, whether there is a terminal or not;
2. reflection from the terminal: if the terminal has an air flow barrier, it will reflect sound back in the direction of the duct;
3. absorption in the terminal: if sound-absorbing materials are present in the terminal, sound attenuation occurs upon contact with them and is not propagated or reflected back into the duct;
4. Position: In the corner, the terminal causes a stronger sound power level with the bass than if it were in the middle (**number 7.15**).

If the airflow barrier has a wide range of motion, sound attenuation can be indicated separately (open/closed) in these extreme positions. Fig. **7.8.1** shows the terminal attenuation values for a terminal organ with two different duct connection sizes. They show the effect of both the reflection factor and the absorption factor.

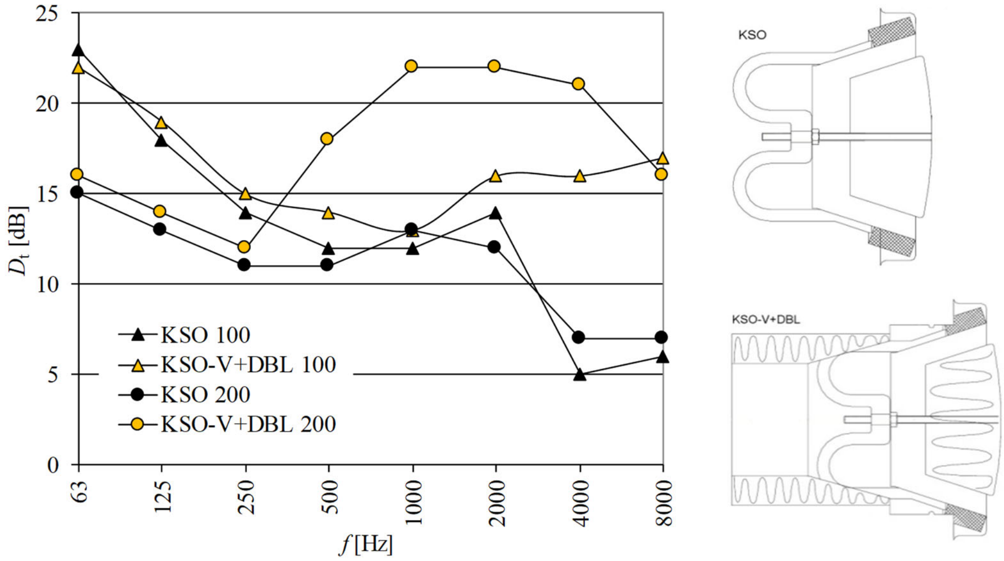


Fig. 7.8.1 Sound attenuation values of a manufacturer's terminal organ from duct to room. The product is mounted on the wall surface (Ω=2π). (Fläkt Woods Oy 2007)

7.9. Sound attenuation of ductwork branches

The sound of the fan is distributed in different directions in the branches of the ductwork, which is a significant factor affecting the final result of the sound attenuation calculation. The reflection of sound in a branch of the ductwork is a phenomenon dependent on geometries **and frequency (Chapter 7.14**), and there are no simple calculation methods for calculating sound attenuation over the entire frequency range.

The sound attenuation D_d [dB] caused by the duct branches from the engine room to the room under consideration is therefore estimated in a simplified manner, which assumes that sound is spread through the ductwork in the same proportion as the airflow. The sound attenuation of the branches of the ductwork from point A to point B is given by the equation

(7.9.1) $$D_d = 10\log_{10}\left(\frac{Q_A}{q_B}\right)$$

where q_B [m^3/s] is the air volume at point B and Q_A [m3/s] is the air volume at the point of departure. Usually, the starting point for noise calculations is either a fan or a damper.

7.10 Room attenuation

Room attenuation D_r [dB] is calculated by the equation

(7.10.1) $$D_r = 10\log_{10}\left(\frac{4}{A}\right)$$

where A [m^2] is the absorption area of the room. If the terminal equipment is located very close to a significant point of residence, such as the speaker's seat, the distance of the terminal device to the listener can also be taken into account, in which case the room attenuation is determined using the equation

(7.10.2) $$D_r = 10\log_{10}\left(\frac{1}{\Omega r^2}+\frac{4}{A}\right)$$

where r [m] is the distance of the observation point to the terminal and Ω is the solid angle to which the terminal radiates sound. For the heading factor k, a value of 1 is used, since terminal equipment manufacturers do not specify directional factor values and sound is usually emitted from terminals quite spherically.

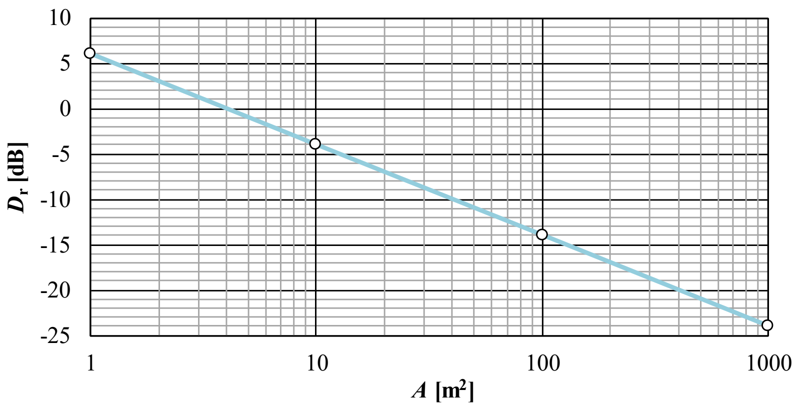


Fig. 7.10.1 Room attenuation *Dr* as a function of absorption area.

7.11. Sound absorption of the ventilation unit room

The SPL, $L_{p,f}$ [dB], of the fan or ventilation unit formed in the engine room is estimated by the diffuse field equation

$$L_{p,f} = L_{W,f} + 10\log_{10}\left[\frac{4}{A}\right] \tag{7.11.1}$$

where $L_{W,f}$ [dB] is the sound power level of the fan or ventilation unit through the envelope to the environment and A [m^2] is the absorption area of the engine room. The sound power level is obtained by selecting the A-weighted sound power level L_{WA} according to the operating point of the fan and applying the octave band values to the "environment" (e.g. Fig. **7.3.1**).

The SPL in the adjacent room of the engine room is given by the equation

$$L_{p,2} = L_{p,1} - R + 10\log_{10}\frac{S}{A_2} \tag{7.11.2}$$

where $L_{p,1}$ [dB] is the SPL in the engine room and R [dB] is the sound reduction index between the engine room and its adjacent room.

7.12. Limit frequency

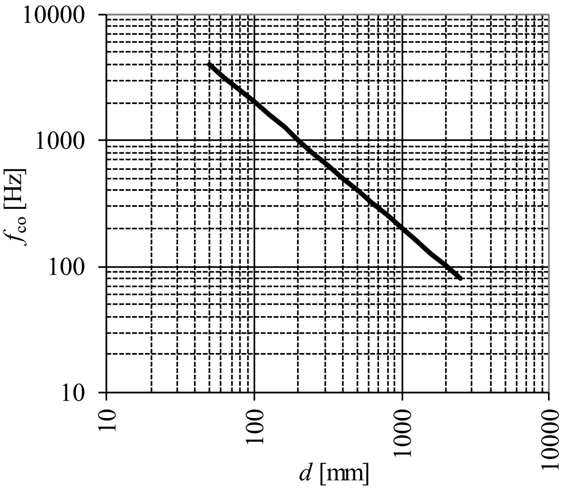
The sound propagation in the channel is from 1 dimension to the cut-off frequency, *f*co (cut-off frequency). Above this limit, the sound field in the channel is 3-dimensional. The boundary frequency occurs when the largest dimension *d* [m] of the channel cross-section is equal to the half of the wavelength

$$f_{co} = \frac{c_0}{2d} \tag{7.12.1}$$

where c_0 is the speed of sound [m/s]. The limit frequencies for channel diameters of round channels are shown in Fig. **7.12.1** at room temperature. The limit frequency increases as the channel size decreases.

Above the limit frequency

- above the limit frequency, sound propagates from channel to room almost undamped,
- The Piening equation does not work, since it does not take into account the transverse sound field,
- bends begin to muffle sound,
- Direct mufflers are not effective because sound can propagate unmuffled through them.



d	***f*_co**	***S***
[mm]	[Hz]	[m²]
100	1715	0,008
125	1372	0,012
160	1072	0,020
200	858	0,031
250	686	0,049
315	544	0,078
400	429	0,126
500	343	0,196
630	272	0,312
800	214	0,503
1000	172	0,785

Fig. 7.12.1 Circular channel boundary frequency *f*co as a function of channel diameter *d*. On the right is the tabulated duct area *S* according to the channel diameter *d* [mm].

7.13. Damping in duct cross-sectional change

This chapter provides a physical explanation of how a change in cross-section attenuates sound. The effect of the cross-sectional change does not need to be taken into account in the sound calculations for ventilation.

When the cross-sectional area of the ductwork changes from area *S1* to area *S2*, reflection occurs at the point of change and the sound power is attenuated behind the point of change. Sound absorption occurs in different ways above and below the limit frequency.

Case (i): *f>fco*

When the wavelength is small relative to the dimensions of the channel cross-section, the sound wave propagating in the central part of the channel in the direction of the channel does not perceive the edges of the channel, but the wave propagates as if it were in a free field. If the channel area increases ($S_1<S_2$), the shape of the wavefront does not change much at the point of cross-sectional change, but the front proceeds from area 1 to area 2 without reflection and the sound absorption is zero:

(7.13.1) $$D_{ac} = 0$$

If, on the other hand, the area decreases ($S_1>S_2$), part S_1-S_2 of the sound S_1 hit the wall and part S_2 advances past the point of change. Sound attenuation is obtained by:

(7.13.2) $$D_{\mathrm{ac}} = 10 \cdot \log_{10}\left(\frac{S_1}{S_2}\right)$$

Case (ii): *f<fco*

Below the limit frequency, the wavelength is large relative to the dimensions of the channel cross-section. The shape of the wavefront changes completely when it hits the discontinuity point of the canal area. Therefore, continuities must be changed to keep the flowrate the same on different sides of the point of discontinuity.

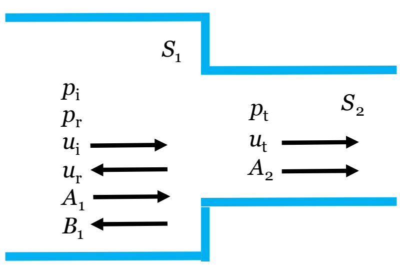


Fig. 7.13.1 Boundary conditions for cross-sectional change boundary area.

According to Fig. **7.13.1**, the boundary conditions are given for sound pressure

(7.13.3) $$p_i + p_r = p_t$$

and volumetric flow rate

(7.13.4) $$S_1\mathbf{u}_i + S_1\mathbf{u}_r = S_2\mathbf{u}_t$$

Sound pressures are of the form

(7.13.5) $$p_i = A_1\cos(\omega t - k_1 x)$$

(7.13.6) $$p_r = B_1\cos(\omega t + k_1 x)$$

(7.13.7) $$p_t = A_2\cos(\omega t - k_2 x)$$

and particle velocities form

(7.13.8) $$\mathbf{u}_i = \left(\frac{A_1}{\rho_0 c_0}\right)\cos(\omega t - k_1 x)$$

(7.13.9) $$\mathbf{u}_r = \left(-\frac{B_1}{\rho_0 c_0}\right)\cos(\omega t + k_1 x)$$

(7.13.10) $$\mathbf{u}_t = \left(\frac{A_2}{\rho_0 c_0}\right)\cos(\omega t - k_2 x)$$

Because for a plane wave, air in a free reflection-free sound field applies

(7.13.11) $$\frac{p}{u} = \rho_0 c_0$$

obtained from:

(7.13.12) $$\left(\frac{S_1}{\rho_0 c_0}\right)p_i - \left(\frac{S_1}{\rho_0 c_0}\right)p_r = \left(\frac{S_2}{\rho_0 c_0}\right)p_t$$

Because the particle velocity vector of the reflected sound points in the opposite direction to the incoming sound. Now let's set $x=0$ and t=0 to place the cosine terms first. As a result, the boundary conditions for sound pressure and particle velocity are reshaped, which depend only on amplitudes and surface areas

(7.13.13) $$A_1 + B_1 = A_2$$

(7.13.14) $$\frac{A_1}{S_1} - \frac{B_1}{S_1} = \frac{A_2}{S_2}$$

Since sound power is obtained in a free field from sound pressure by the equation

(7.13.15) $$W = IS = \frac{p^2}{\rho_0 c_0}S$$

can be derived as the transmission coefficient by the equation

(7.13.16) $$\tau = \frac{4S_1S_2}{(S_1 + S_2)^2}$$

and sound absorption

(7.13.17) $$D_{ac} = 10 \cdot \log_{10}\left[\frac{(S_1 + S_2)^2}{4S_1S_2}\right]$$

The same equation applies both when the ductwork area increases and decreases.

Case (iii): ***f≈fc***

In the range of the limit frequency, sound attenuation is estimated by interpolation.

Fig. **7.13.2** shows sound attenuation calculations when ductwork decreases or increases. At high frequencies, sound attenuation occurs when moving from a larger to a smaller channel, since some of the sound hits the wall and is reflected back.

In general, the cross-sectional areas are changed sliding (transition element, funnel) rather than suddenly in order to avoid strong pressure drops and flow noise. The equations in this number do not apply if the point of change is sliding.

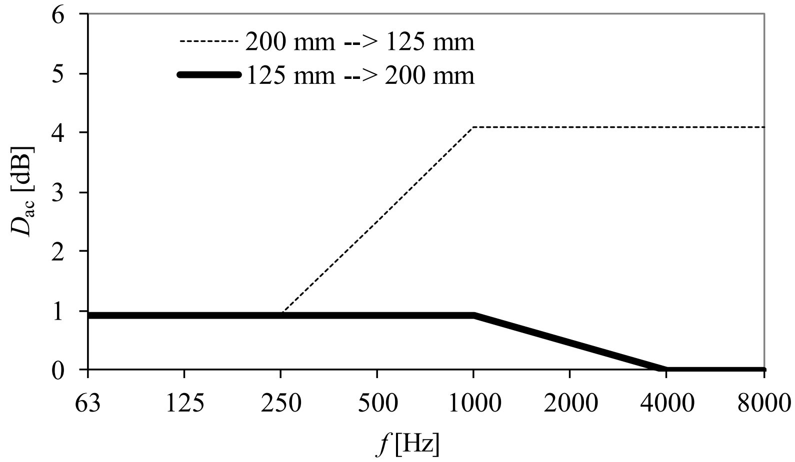


Fig. 7.13.2 Sound absorption at the point of sudden change of the circular ventilation duct when the duct decreases (thin line) or increases (thick line). Dimensions are channel diameters.

7.14. Damping in a single branch of the duct

The analysis of sound attenuation produced by a single branch is based on the situation shown in Fig. **7.14.1.** It assumes that the area of the outgoing branch and the total area of the branches are approximately the same, that is, $S_1 \approx S_2 + S_3$. Otherwise, the area change formula presented above must be taken into account in addition to those presented here.

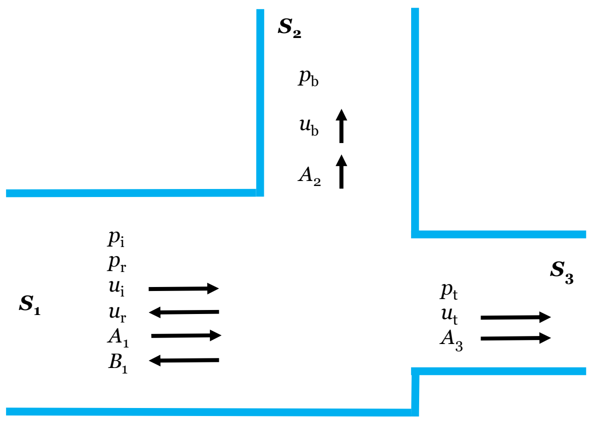


Fig. 7.14.1 Consider the transition of sound from branch 1 to branch 2.

Again, let's perform a review above and below the limit frequency fco.

Case (i): ***f>fco***

At high frequencies, all sound energy continues its journey to branching channels 2 and 3, and sound attenuation to branch 2 is obtained by:

(7.14.1) $$D_{\mathrm{d},2} = 10 \cdot \log_{10}\left(\frac{\sum S - S_1}{S_2}\right) = 10 \cdot \log_{10}\left(\frac{S_2 + S_3}{S_2}\right)$$

Case (ii): ***f<fco***

At low frequencies, sound attenuation to branch 2 is either the same as above or slightly higher

(7.14.2) $$D_{\mathrm{d},2} = 10 \cdot \log_{10}\left[\frac{\left(\sum S\right)^2}{4S_1S_2}\right]$$

Fig. **7.14.2** shows branch attenuation. If the area of branches 2 and 3 is less or greater than the area of branch 1, greater attenuation will be seen at low frequencies than at high frequencies because the impedance changes.

If the total area does not change substantially after the branch (S1, ≈S2+S3), equation 7.14.2 may be used over the entire frequency range.

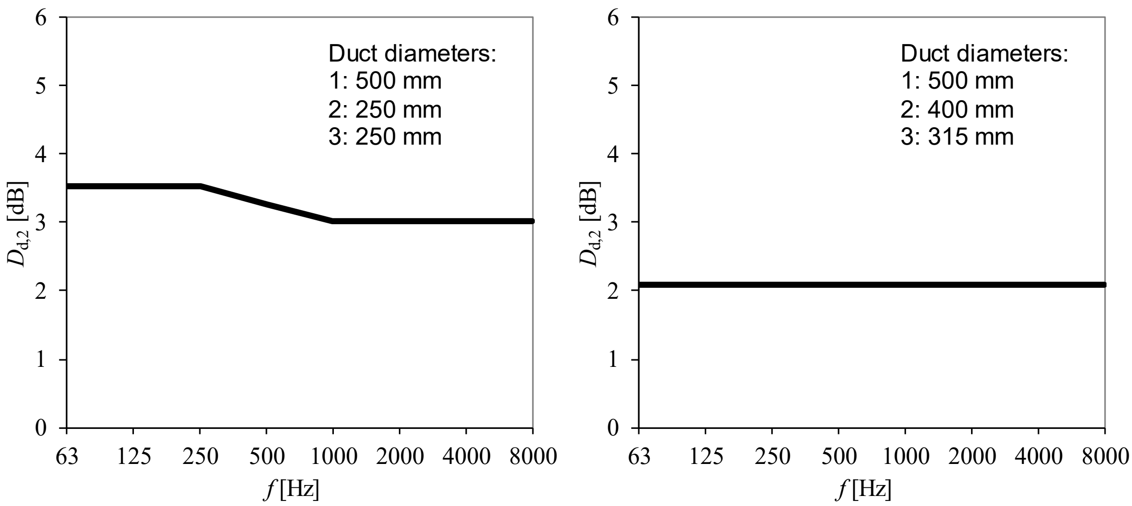


Fig. 7.14.2 Sound attenuation in the crotch in two basic situations. On the left, the total area is halved after the branch, and on the right it does not change at all.

7.15. Termination attenuation of terminating duct

When the channel ends in the room, the change in the cross-sectional area is very large. If there is no terminal organ at the end point of the channel, the terminal attenuation $D_{t,0}$ of the terminating channel is calculated from the equation

(7.15.1) $$D_{t,0} = 10 \cdot \log_{10}\left[1 + \left(\frac{c_0}{4\pi f}\right)^2 \frac{\Omega}{S}\right]$$

where c_0 [m/s] is the speed of sound, f [Hz] is the frequency, Ω is the solid angle of sound radiation, and S [m^2] is the area of the terminating channel.

The dependence of open channel terminal attenuation on frequency and duct cross-sectional area is shown in Fig. **7.15.1.**

At frequencies clearly lower than the boundary frequency ($f < f_{co}$), the sound pressure is constant over the entire channel cross-sectional area, and the change in cross-section affects the plane wave propagating in the channel.

At the end of the duct there is a strong change in impedance, and this causes a reflection, which is visible in the room as sound absorption.

Diffraction also occurs at the end of the channel, the intensity of which depends on the frequency. At the endpoint, diffraction affects that part of the plane wave propagating in the channel that is less than half a wavelength from the edge of the channel. At low frequencies, the diameter of the channel is small in relation to the wavelength, in which case diffraction affects the entire cross-sectional area of the channel and diffraction is very strong: before the point of change, a plane wave propagates in the channel, and after the point of change, a sphere wave propagates in the channel. In contrast, at high frequencies, diffraction occurs at the end of the channel only at the periphery of the channel, and the central part of the plane wave continues into the room without any waveform change at the channel end.

If the end of the duct is in the corner (Ω=π/2), $D_{t,0}$ decreases because the change in cross-sectional area is not as steep as when the end of the duct is in the center of the room (Ω=4π). At high frequencies, the location of the channel head does not affect anything.

The product value of terminal sound attenuation includes the end attenuation $D_{t,0}$ of the open channel head described here, although it is not due to the terminal itself, but to the end of the duct.

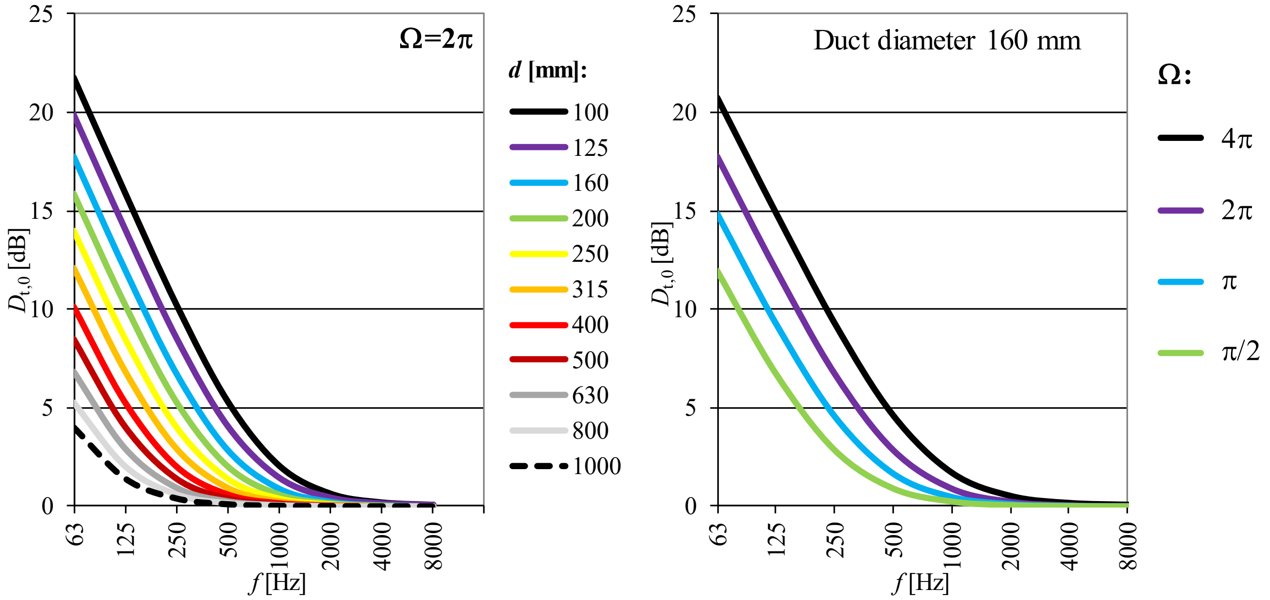


Fig. 7.15.1 (a) Open channel end attenuation *Dt,0* with wall mounting (Ω=2π). (b) Effect of terminal position, i.e. solid angle, on open channel terminal attenuation.

7.16. Duct wall sound absorption

The walls of the ventilation duct are, for example, made of galvanized steel or polypropylene. In the walls, both sound absorption and sound penetration through the channel occur.

Sound absorption is typically as follows (Halme and Seppänen, 2003):

- in circular channels, 0,30 dB/m at 63 Hz and reduced to 0,05 dB/m at 8 kHz,
- in rectangular channels from 0.60 dB/m at 63 Hz to 0.05 dB/m at 8 kHz,
- in a rectangular channel insulated with a layer of wool and external sheet metal at 1,20 dB/m at 63 Hz and reduced to 0,05 dB/m at 8 kHz,

The sound absorption of the duct wall is usually ignored in the calculations.

7.17. Damping of the bend

Fig. **7.17.1** shows the sound attenuation values for the non-attenuated bend. There is no simple calculation formula for the sound absorption Dm produced by the duct bend . Below the limit frequency ($f<f_{co}$), due to diffraction, the sound bends easily past the bend and the sound attenuation of the bend is small. At high

frequencies ($f > f_{co}$), sound collides with the channel wall and the main part of the sound is reflected back and only a small part bends or is reflected around the bend.

In air conditioning noise reduction calculations, the effect of bends on the reduction of sound power is often ignored. The underlying assumption is that sound attenuation is not a problem at high frequencies, as there is already plenty of attenuation in the ductwork. Neglecting bend damping in sound level calculations may cause an apparent need for additional damping elsewhere.

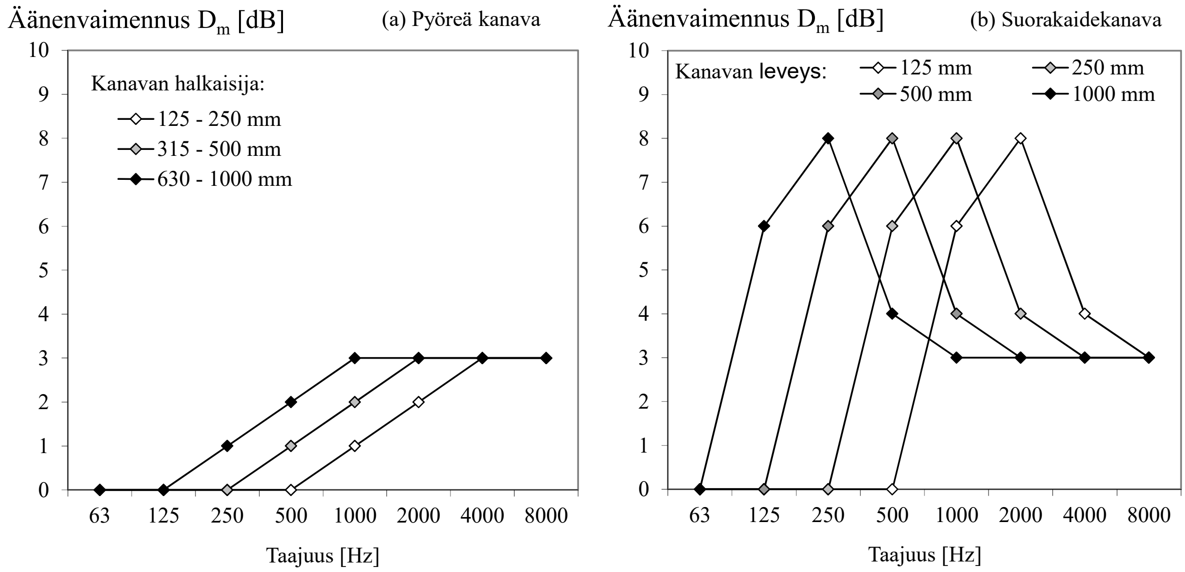


Fig. 7.17.1 (a) Sound absorption of a 90° curve in a circular channel. The rounding radius should be less than 2 channel diameters so that the flow sound is not too strong. (b) Sound absorption of a 90° bend made of rectangular duct (Source: VDI 2081:1983).

7.18. Operation of the passive silencer

A passive muffler is an element with a porous sound-absorbing material either on the inner walls or in its central parts. Typical structures are shown in Fig. **7.18.1.** Figures 7.18.1 **and** 7.18.2 show the effects of the basic parameters of silencers on silencing. Sound absorption is improved when

- the absorption ratio of the absorption material improves,
- the muffler becomes longer,
- the proportion of free cross-section decreases, and
- The muffler is curved (angle absorber).

An increase in the thickness of the absorption material is observed especially as an improvement in sound absorption of low frequencies. Reducing the cross-sectional section increases the pressure drop.

The sound absorption of a lamellar muffler does not significantly depend on the width, that is, on the number of lamellas, since the geometry of the path of sound passage and the absorption conditions of the channel remain the same.

An angle-shaped lamella muffler absorbs 0–8 dB more sound than a straight muffler of the same size. The least angle affects at low frequencies.

Fig. 7.18.1 Standard passive silencer types: rectangular lamella, angle-mounted lamella, straight channel, curve channel, low channel, low angle duct.

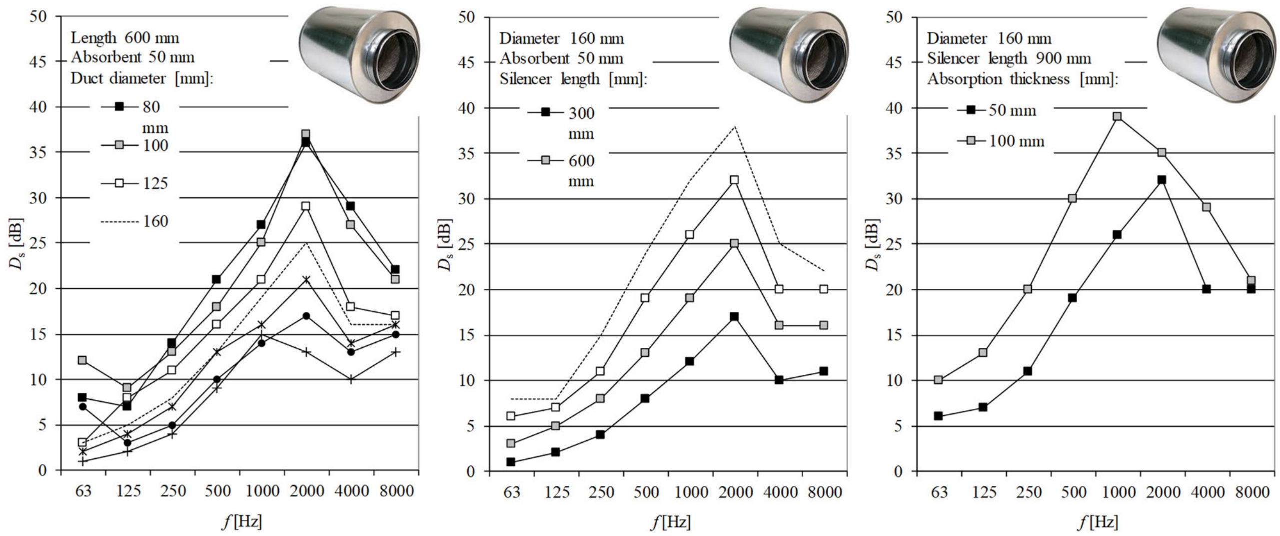


Fig. 7.18.2 Dependence of direct duct silencer sound absorption on various factors. (a) Effect of duct diameter with silencer length L=600 mm and wall mineral wool of 50 mm. b) The effect of the length of the muffler with 50 mm mineral wool in the wall and a duct diameter of 160 mm. (c) The effect of the thickness of the absorption material with a damper length of 900 mm and a duct diameter of 160 mm (Fläktwoods Oy, 2019).

Below the limit frequency, sound propagates like a stream of air in the direction of the duct and there is no transverse wave motion relative to the direction of the duct (Fig. **7.18.3a**). Since the wavelength of sound is significantly larger than the diameter of the duct, the same sound pressure prevails over the entire cross-sectional area of the duct.

Above the limit frequency, both a plane wave (channel longitudinal) and a transverse wave (Fig. **7.18.3b) propagate in the channel**. In this case, the sound attenuation mechanism must be taken into account separately in the longitudinal and transverse directions. In practice, the transverse sound wave is attenuated very quickly in conventional silencers, leaving only a longitudinal sound field in the silencer. The transverse sound field develops again when sound enters the hard-walled channel.

The sound attenuation of the longitudinal sound field plays the greatest role in the silencer's sound attenuation value. Sound attenuation D can be predicted up to about 2000 Hz with moderate accuracy using the Piening equation

(7.18.1) $$D_{\text{pit}} = 1.5 \cdot \frac{P}{S} \cdot \alpha \cdot L$$

where P [m] is the measure of the duct circuit, S [m^2] is the cross-sectional area of the channel, α is the frequency-dependent absorption ratio of the material on the wall, and L [m] is the length of the muffler. Above the frequency of 2000 Hz, sound absorption decreases sharply with increasing frequency. The shorter the wavelength, the greater the proportion of the plane wave in the central part of the silencer can propagate through the silencer without hitting the absorbent walls (**7.18.3c**).

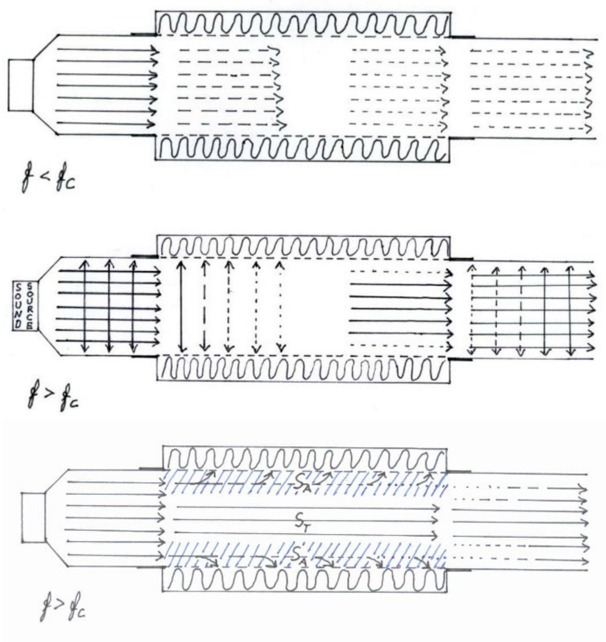


Fig. 7.18.3 On the left, the audio source feeds sound to the channel. In the middle of the muffler and on both sides of it a hard-walled channel. (a) The sound field below the limit frequency *fc*. (b) The sound field above the limit frequency. (c) Absorption of the plane wave above the limit frequency. In zone *ST*, the plane wave propagates through the damper, and in zone *S*A, the plane wave is absorbed by the walls.

7.19. Operation of the reactive silencer

Types of reactive dampers are ventricle, Helmholz and quarter-wave resonator. The sound absorption of a truly reactive silencer is based on geometry-induced reactive impedance. Damping reaches a maximum at the frequency (or frequencies) at which the inner dimensions of the chamber cause resonance. In antiresonance, attenuation does not occur. Reactive dampers work most effectively below the limit frequency, that is, when a plane wave propagates in the main channel.

Reactive silencers are rarely used in conventional air conditioning systems, as the noise produced by the systems can be suppressed by passive silencers. In particular, reactive attenuators are used to suppress periodic and very low frequency sounds (below 50 Hz), since passive attenuators do not provide sufficient sound attenuation at reasonable material strengths and pressure drops. Reactive dampers are usually used in exhaust ducts to absorb narrowband excitations from the machine. Unlike passive dampers, reactive dampers are highly resistant to high flow rates, chemicals and high temperatures.

Reactive silencers can be fitted with porous materials, which makes sound attenuation more broadband, but on the other hand, at resonant frequency, sound attenuation weakens.

The ventricular resonator (Fig**. 7.19.1**) consists of an incoming channel with area *S*1, an extension with length L and cross-sectional area S2, and an outgoing channel, the area of which is assumed here to be the same as that of the incoming channel. Sound attenuation can be estimated at low frequencies (*f<fco*) by the equation

(7.19.1) $$D = 10 \cdot \log_{10}\left[1 + \left(\frac{S_1}{2S_2} - \frac{S_2}{2S_1}\right)^2 \sin^2(kL)\right]$$

Above the limit frequency, attenuation is minimal and is caused by a change in the cross-sectional area of the channel.

Fig. **7.19.1** shows the calculated sound absorption of a chamber with geometry. Sound absorption strongly depends on frequency. The first attenuation maximum is observed at a resonant frequency, where the length of the chamber corresponds to a quarter of the wavelength. The resonant frequency occurs at odd multiples of this

(7.19.2) $$f_{res} = \frac{nc_0}{4L};\ n = 1,3,5,...$$

The maximum value of sound absorption at the resonant frequency depends on the ratio of areas

(7.19.3) $$D_{\max} \approx 20\log_{10}\frac{S_2}{2S_1}$$

The value increases by 6 dB with a doubling of the ratio S2/S1.

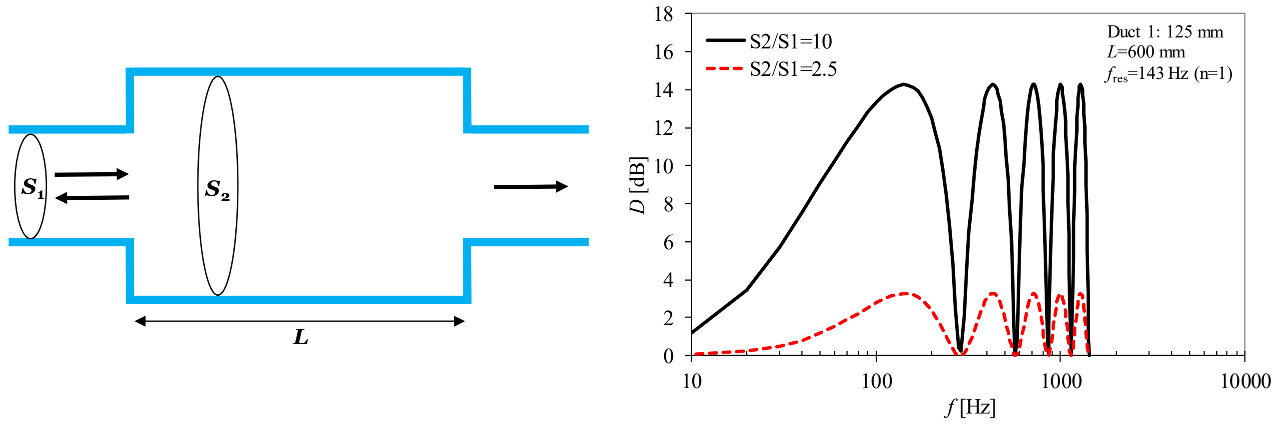


Fig. 7.19.1 (a) Chamber attenuator and (b) example of its silencing.

A side branch refers to situations where there is an element with a reactive impedance on the side of the duct as shown in Fig. **7.19.2.** Suppose that the opening of the element is significantly smaller than the wavelength, i.e. k_a<<1, where k is the wavenumber and a is the diameter of the opening. Sound absorption in this case is

(7.19.4) $$D = 10\log_{10}\left|1 + \frac{S_s\rho_0 c_0}{2S\boldsymbol{Z}_s}\right|^2$$

The equation holds for any branch whose input impedance **Zs** is known. Zs of the most common lateral branches are derived below. Sound absorption is at its highest when the impedance is zero. This prevails when there is resonance in the side branch. According to the formula, the attenuation is infinitely high for resonance, but in practice there are always losses in the element, so the sound attenuation usually does not exceed 20 dB.

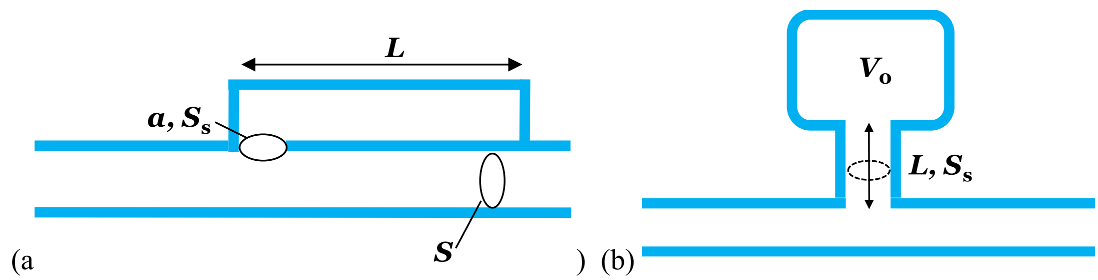


Fig. 7.19.2 (a) Quarter-wave resonator. (b) Helmholz resonator.

The quarter-wave resonator (Fig. **7.19.2a**) has a lateral branch with a length L and a constant cross-sectional area. Its input impedance is

(7.19.5) $$\mathbf{Z}_s = -i\rho_0 c_0 \cot kL$$

The maximum attenuation is obtained when this is zero, which is realized under condition

(7.19.6) $$kL = \frac{n\pi}{2}, \quad n = 1,3,5,...$$

The lowest resonant frequency occurs at wavelength $\lambda = L/4$.

A Helmholz resonator (Fig. **7.19.2b**) is a spring-mass system in which air at the entrance forms a mass and the air in the tight chamber behind it forms a spring. It is assumed that the dimensions of the resonator are significantly smaller than the wavelength, in which case all its parts are affected by constant sound pressure and the mass of the entrance behaves as a lumped mass system. The input impedance of a Helmholz resonator is of the form

(7.19.7) $$\mathbf{Z}_s = \frac{p_s}{u_s} = i\omega\rho_0 L + \frac{\rho_0 c_0^2 S_s}{i\omega V_0}$$

where L [m] is the length of the opening, S_s [m^2] is the cross-sectional area of the opening, and V_0 [m^3] is the volume of the chamber behind the opening. Maximum attenuation is achieved at the specific frequency

(7.19.8) $$f_{res} = \frac{c_0}{2\pi}\sqrt{\frac{S_s}{LV_0}}$$

Depending on how the opening of the resonator relates to the canal and the volume of the resonator, a close field is formed at both ends of the opening, which, from an acoustic point of view, increases the length of the throat by L by ΔL_i at each end of the throat. The significance of the nearfield is great if the diameter d of the throat cross-section begins to be in the order of the throat length L. The term ΔL is called *end correction*. The effective throat length L' is therefore

(7.19.9) $$L' = L + \Delta L_1 + \Delta L_2$$

End correction depends on many factors, but in a throat with a round cross section, sufficient accuracy is achieved when the throat diameter d is known. If the opening of the throat is in the wall (*baffle*), which is the most common situation in the canal, then the equation can be used

(7.19.10) $$\Delta L = 0.82\frac{d}{2}$$

Due to gable correction, the length of the Helmholz resonator opening may be non-existent, since the effective length of the opening depends on the diameter. The hole in the duct wall alone (wall thickness is usually only L=0.005 m) and the finite volume behind it form a Helmholz resonator.

In recent examinations, the components were assumed to be fully reactive, i.e. the specific real impedance part was zero. The imaginary part depended heavily on frequency, and at certain special frequencies the impedance dropped to zero.

In practice, no real system is lossless, as all walls have some absorption, sound transmission, and friction occurs on the throat walls as air flow rubs against the walls. Heat losses also occur in gas. These losses are resistive, that is, sound is converted into heat.

For example, friction in the throat of a Helmholz resonator reduces the particle velocity and the input impedance is

(7.19.11) $$\mathbf{Z}_s = R + i\omega\rho_0 L + \frac{\rho_0 c_0^2 S_s}{i\omega V_0}$$

Due to losses, maximum attenuation at the resonant frequency decreases, but attenuation is also present in antiresonances. Thus, attenuation is more broadband, i.e. the bandwidth B of the attenuator increases.

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8 SOUND INSULATION IN BUILDING

8.1. Flanking transmission

In practice, the airborne sound insulation measured between rooms in a building is always lower than the laboratory value of the building part separating the rooms, because sound is also transmitted through structures tangential to the building part (external walls, internal walls, intermediate floors), along air routes (ventilation ducts, pipe penetrations, gaps) and along the structural walls of HVAC pipelines. Lateral displacement can also occur from the outside by air through structures if, for example, identical exterior doors or windows of adjacent apartments are located close to each other and their airborne sound insulation (e.g., at resonant frequency) is low.

The same applies to impact sound insulation, with the difference that flanking transmission occurs only along structures.

It is very usual that flanking transmission carries more energy between rooms than direct transmission via separating structure. This is achieved if the sound reduction index in situ is more than 3 dB less than the sound reduction index of the separating building element in laboratory. Damping flanking transmission along air routes is usually easy (sealing, sound absorption of ventilation ductwork), but reducing structural flanking transmission in a finished site is usually difficult, if sometimes even impossible. For this reason, flanking transmission should be anticipated at the planning stage.

8.2 Airborne flanking transmission in massive structures

The modelling of structural flanking transmission takes place in the frequency range 100–3150 Hz that determines the weighted sound reduction index R'_W or the sound level difference $D_{nT,w}$. Modeling a massive single-layer structure with the accuracy required by engineering calculations is quite simple, since the oscillatory behavior of massive structures is easily predictable in this frequency range. The vibration rate is the same whether measured at any point in the structure, as long as you do not go over the structural joint. For example, the coincidence limit frequency for reinforced concrete structures 160 mm or thicker occurs in the 100 Hz range. Then, the structural vibration above 100 Hz is resonating. The resonant vibration propagates over the joints in a way that is relatively easy to model.

Modelling is still possible if a light structural layer is placed on top of the massive structure to improve airborne sound insulation. The airborne sound insulation improvement ΔR_w produced by light layers can be added directly on top of the airborne sound insulation number of the massive section if the value was determined in the laboratory.

A key role in the strength of the flanking transmission is played by the type of joint. The joint must have vibration insulation K [dB], which indicates how much the joint insulates vibration as it moves over the joint. Fig. **8.2.1** shows the vibration insulators of the most common rigid joints. For massive structures, the values do not depend significantly on the frequency in the resonating frequency range. The insulation of the joint improves as the number of associated structures increases and/or the difference between the surface masses of the joint structures increases.

When examining the intervals between apartments, the mass differences over the joints are often small and one can look at $m'_2/m'_1=1$. As a rule of thumb, the vibration insulation of the cross joint is 9 dB and that of the T-junction is 6 dB.

For comparison, Fig. **8.2.2** shows vibration insulators obtained with flexible material. Values over flexible material are up to 10 dB higher than in a rigid joint. It should be noted that a flexible joint is useful only for such a path of propagation of vibration in which the flexible material will have to be crossed. In the case of a flexible connection, the sound energy remains on the transmission side, allowing the structure in question to radiate sound more intensely to its surroundings. Rigid joints thus improve sound insulation locally but cause stronger flanking transmission than flexible joints.

Fig. **8.2.3** shows the marking of flanking transmission according to EN 12354-1. The separating structure (partition) is denoted by D when viewed from the transmission room and d by D from the reception room.

When viewed from the transmission side, there are usually four tangential structures F (2 walls, ceiling and floor). There are also four tangential structures f from the reception room side.

Normally, there are 13 first-order sound paths to be modelled. There are 1 direct audio paths (Dd). There are 4 sound paths (Ff) proceeding along purely tangential structures, 4 sound paths (Df) from the partition wall to the tangent structure, and 4 (Fd) from the tangent structure to the partition.

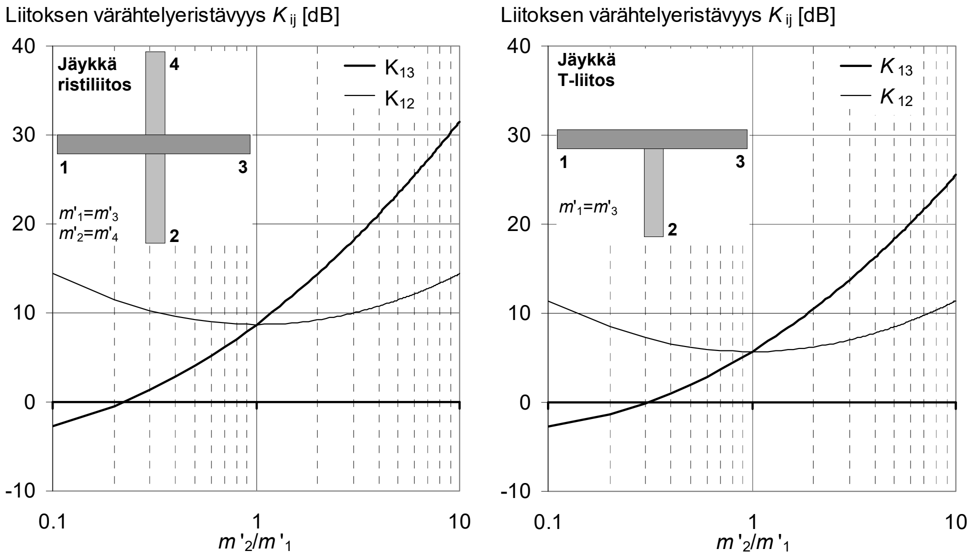

Fig. 8.2.1. Vibration insulation of rigid joints.

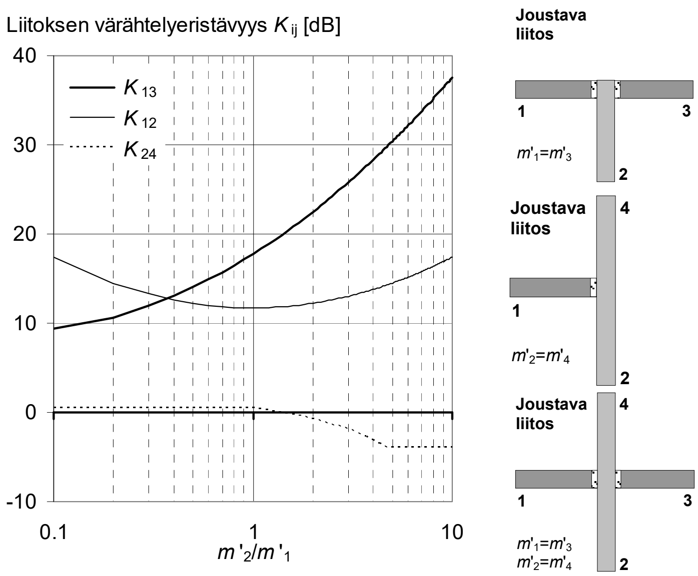

Fig. 8.2.2. Vibration insulation of flexible joints.

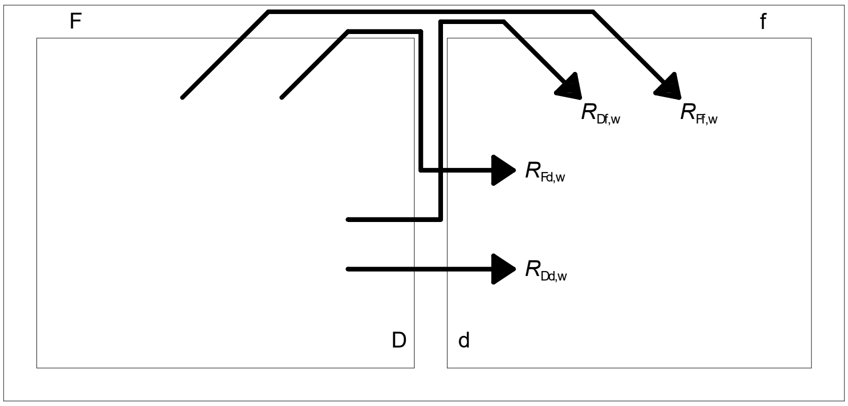

Fig. 8.2.3. Marking of flanking transmission routes according to EN 12354-1 for one tangent structure. The source room is on the left and the reception room is on the right.

Audio paths over two joints are ignored in the EN model. For example, the back wall is not viewed at all, although sound is emitted from there as well. Experience has shown that omitting sound routes of the second order and longer causes flanking transmission to be underestimated by 1–2 dB. The other sources of error are larger than this, so the approach shown in Fig. **8.2.3** is sufficient for practical calculations.

The soundproofing of the partition separating the rooms, i.e. the straight route, is given by the equation

(8.2.1) $$R_{Dd,W} = R_{s,W} + \Delta R_{Dd,W}$$

where $R_{s,W}$ [dB] is the weighted sound reduction index of the separating massive wall, $\Delta R_{Dd,w}$ [dB] is the effect of improving the weighted sound reduction index produced by either the transmission or an additional layer or lining built on the receiving room side.

The "airborne sound insulation number" of the flanking transmission route for three different types of flanking transmission is given by the equations

(8.2.2) $$R_{Ff,w} = \frac{R_{F,w} + R_{f,w}}{2} + \Delta R_{Ff,w} + K_{Ff} + 10\log_{10}\frac{S_s}{l_f}$$

(8.2.3) $$R_{Fd,w} = \frac{R_{F,w} + R_{d,w}}{2} + \Delta R_{Fd,w} + K_{Fd} + 10\log_{10}\frac{S_s}{l_f}$$

(8.2.4) $$R_{Df,w} = \frac{R_{D,w} + R_{f,w}}{2} + \Delta R_{Df,w} + K_{Df} + 10\log_{10}\frac{S_s}{l_f}$$

where

- $R_{F,w}$ [dB] is the airborne sound insulation number of structure F in the transmission room,
- $R_{f,w}$ [dB] is the airborne sound insulation number of structure f in the reception room,
- $\Delta R_{Ff,w}$ [dB] is an improvement in the airborne sound insulation Fig. produced by a light additional layer built on the surface of the tangential structure of the transmitting and/or receiving room,
- $\Delta R_{Fd,w}$ [dB] is an improvement in the airborne sound insulation Fig. produced by the adjacent structure of the transmitting room and/or an additional layer of light construction on the receiving room side of the separating wall,
- $\Delta R_{Df,w}$ [dB] is an additional light layer of airborne sound insulation built on the transmitting room side of the separating wall and/or on the surface of the tangential structure of the receiving room,
- K_{Ff} [dB] is the vibration insulation for path Ff,
- K_{Fd} [dB] is the vibration insulation for route Fd,

- K_{Df} [dB] is the vibration insulation for path Df,
- S_s [m^2] is the area of the dividing wall, and
- l_f [m] is the joint joint length of the separating partition and the associated structure.

The airborne sound insulation number R'_w is calculated from the equation

$$R'_w = -10\log_{10}\left[10^{-R_{Dd,w}/10} + \sum_{F=f=1}^{4} 10^{-R_{Ff,w}/10} + \sum_{f=1}^{4} 10^{-R_{Df,w}/10} + \sum_{F=1}^{4} 10^{-R_{Fd,w}/10}\right] \quad (8.2.5)$$

All baseline values for airborne sound insulation figures are laboratory values (no comma), but the final result is presented in comma, since the object of calculation is located in the field.

If the partition wall is lightweight, the terms Df and Fd can be omitted from the examination and only the straight route Dd and the fences tangential structures Ff are considered.

8.3 Airborne flanking transmission in light structures

Estimating flanking transmission along light building elements is significantly more difficult than secondary flanking transmission along massive building elements. For light structures, the limit frequency of coincidence is more than 2000 Hz (e.g., appr. 2500 Hz for 13 mm gypsum), so that a large part of the 100–3150 Hz frequency range under consideration occurs only in light structures. forced vibration. Forced oscillation means that a structure is forced into motion even though it does not yet exhibit vibrational forms (modes) except at some frequencies. The intensity of forced oscillation decreases with increasing mass, as predicted by the law of mass.

In addition, lightweight structures are always periodic due to various stiffeners and frames. As a result, the oscillation rate changes sharply when crossing the fulcrum or spine. Even in the direction of the spines, the forced vibration is attenuated as a function of distance, whereas in massive structures this is not the case.

The predominant form of oscillation affecting sound insulation in light structures at frequencies 100–3150 Hz (forced oscillation) is not longitudinal oscillation in the structure in the same way as resonant oscillation in massive structures. Forced vibration does not propagate over joints nearly as easily as in massive structures. Vibration insulation values are higher than for solid structures. Thus, lightweight structures do not cause flanking transmission by the same mechanism as massive structures. The lightweight structure radiates the sound of flanking transmission mainly from a corner near the joint, and even that is minor. The intensity of the vibration quickly fades when moving away from the corner. This has been demonstrated by intensity measurements in practical buildings. The most sound is emitted from the surface between the joint and the first bump or floor hammer, which is 400–600 mm wide.

Fig. **8.3.1** shows the joint sound insulation of the most common joints of light and heavy structures. It should be noted that in the junction of two heavy structures, $m'_2/m'_1=1$ prevails very often. However, in the case of a connection between light and massive structures, $m'_2/m'_1>10$. Thus, the joint sound insulation values are estimated from a completely different part of the figure, and the values are significantly higher than what is used in joints of heavy structures.

However, lightweight structures are not free from structural flanking transmission. Even if the joint of a lightweight wall dampens sound well, the vibration rate of a lightweight wall is high due to its lightness, especially at low frequencies. Vibrations in the form of forced vibrations are therefore also carried along tangent structures.

If heavy and light structures meet at the joint, then no joint sound insulation occurs in the heavy structure for the reason that the sound energy is not distributed in different directions at the joint due to low coupling.

In light structures, structural flanking transmission can also be sound energy transport through cavity, as shown in Fig. **8.3.2.** For example, there is often such a situation on the outer walls.

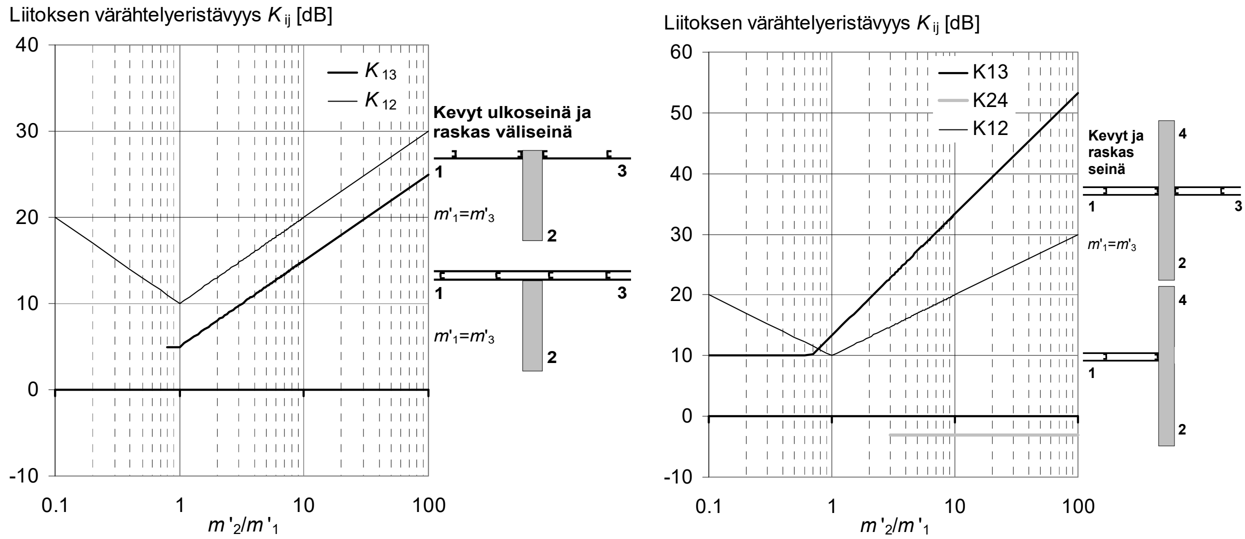

Fig. 8.3.1. Connection sound isolation for light and heavy construction.

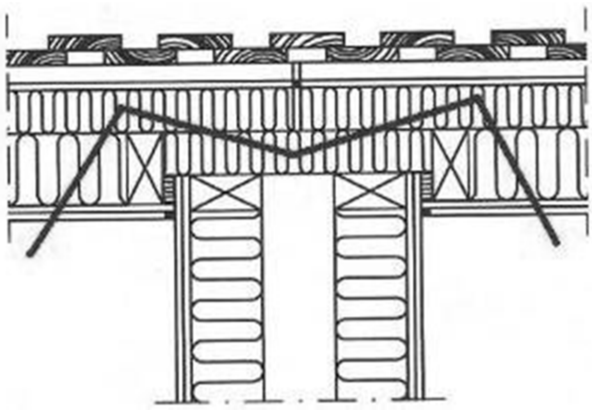

Fig. 8.3.2. Flanking transmission via the air path of the façade structure.

8.4 Modelling flanking transmission for impact sound

Lateral transitions of impact sounds are usually modelled according to EN 12354-2. Lateral displacements of impact sounds are simpler to handle than those of air sounds, since the excitation is directed only to the floor and there are fewer primary flanking transmission paths (Fig. **8.4.1**). For horizontal flanking transmission, the number of primary sound paths is limited to two and vertically to five.

The vertical impact SPL L'_n [dB] is given by:

(8.4.1) $$L'_n = 10\,\log_{10}\left(10^{L_{n,d}/10} + \sum_{j=1}^{n} 10^{L_{n,ij}/10}\right)$$

where $L_{n,d}$ [dB] is the impact SPL formed by the separating intermediate floor and $L_{n,ij}$ [dB] is the impact SPL formed through the surface j of the receiving room. Horizontally, the impact SPL is correspondingly

(8.4.2) $$L'_n = 10\,\log_{10} \sum_{j=1}^{n} 10^{L_{n,ij}/10}$$

N=4 is usually used vertically and n=2 horizontally. The method of calculation is not presented here.

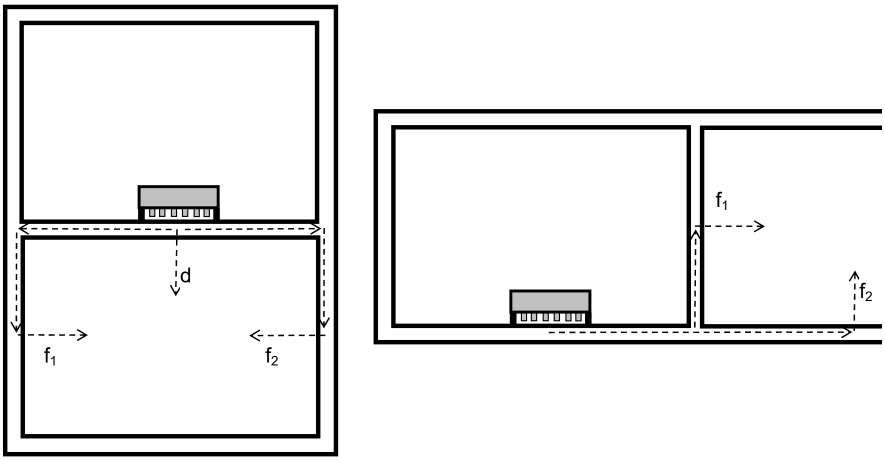

Fig. 8.4.1. Formation of vertical and horizontal impact SPL in 2 dimensions.

8.5. Example of structural flanking transmission

Floating floors are used to improve vertical air and impact sound insulation. The surface slab of a floating floor is always relatively light compared to the load-bearing structure underneath. Therefore, the floating surface structure must be cut off at the partition wall of the apartments to prevent horizontal flanking transmission. If no cut-off is made, a horizontal flanking transmission occurs (Fig. **8.5.1**).

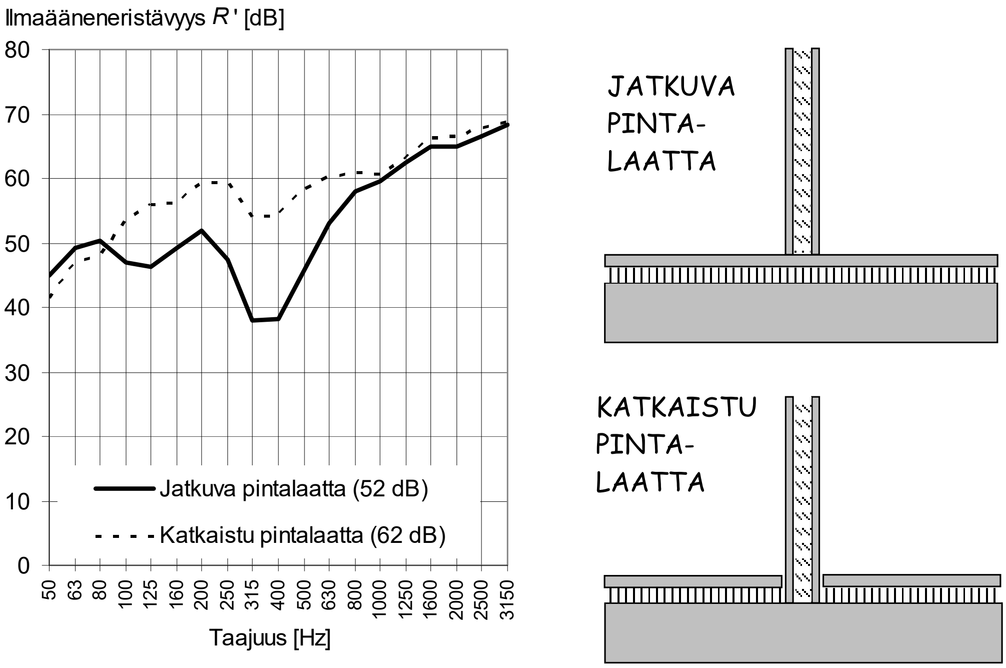

Fig. 8.5.1. Effect of cutting off the surface slab (60 mm concrete) on airborne sound insulation (RIL 243-1:2007).

8.6. Joint sound insulation

The structure separating spaces may consist of several different components, such as a wall, window, door and air penetrating element (**Fig. 8.6.1**). The total sound insulation, R_{tot}, of a wall consisting of different parts is calculated from the equation of total sound insulation (or resultative sound insulation)

(8.6.1)
$$R_{tot} = 10 \cdot \log_{10}\left(\frac{\sum_i S_i}{\sum_i S_i 10^{-R_i/10}} \right)$$

where S_i [m^2] is the area of component I, and R_i [dB] is the airborne sound reduction index of component i.

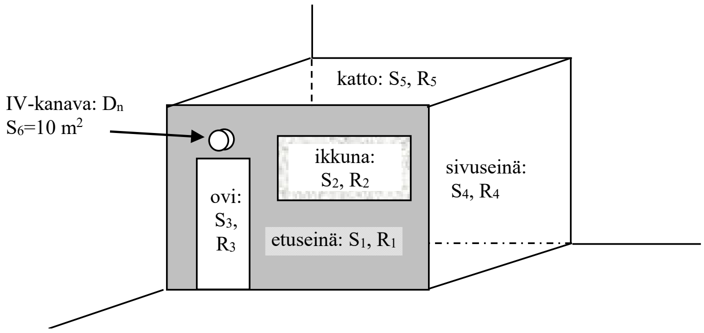

Fig. 8.6.1. Control room built of 6 structural elements.

8.7 The effect of leaks on airborne sound insulation

Most structures and products are designed to be leak-tight and laboratory testing takes place ideally in a situation where no leaks are left in the sample being tested. In practice, leaks remain at the construction stage (installation errors) or are formed during the life cycle (movement of elements, wear of seals). Fig. **8.7.1** shows the airborne sound insulation values of a door, both measured in the laboratory and in the building.

The relative effect of the leak on the airborne sound insulation of the structure increases as the airborne sound insulation of the structure increases. If the total surface area of the structure is S_1 and the sound reduction index R_1, and the leak is S_2 and R_2, respectively, the combined sound insulation of the leak and structure, R_{tot}, can be estimated using the equation of total sound insulation

(8.7.1) $$R_{tot} = 10 \cdot \log_{10}\left[\frac{S_1 + S_2}{S_1 \cdot 10^{-R_1/10} + S_2 \cdot 10^{-R_2/10}}\right]$$

If the leak is hard-walled, the airborne sound insulation of the leak can be set to zero at the first approximation, i.e. R_2=0 dB. This is illustrated in Fig. **8.7.2.** If there are bends, absorbent material or seals in the leak, the airborne sound insulation of the leak is higher.

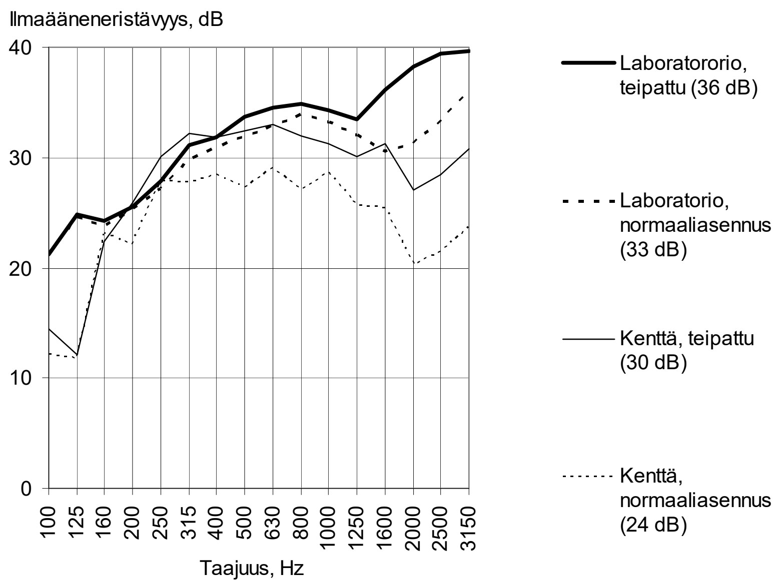

Fig. 8.7.1. Airborne sound insulation of a decibel door (door class 30 dB) in the laboratory and in the field at the construction site. In addition to the normal installation, the measurement was carried out with the seams taped, so that the proportion of leaks can be seen. In parentheses, R_w/R'_w values.

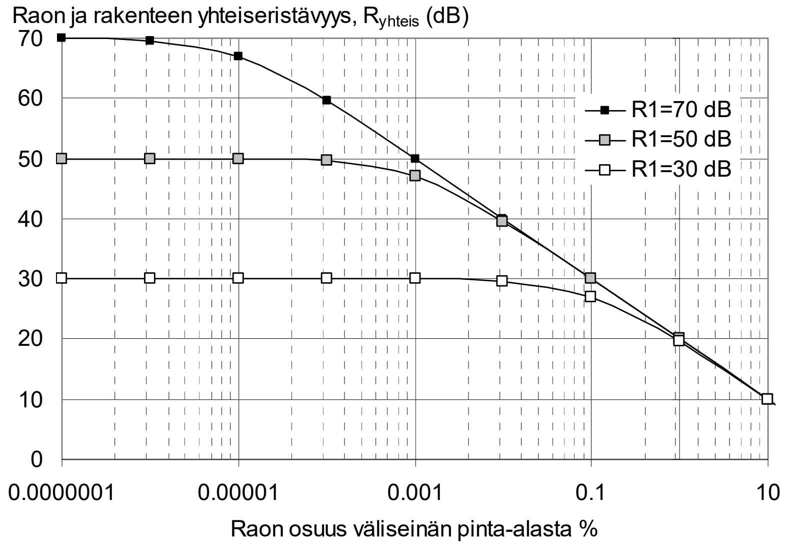

Fig. 8.7.2. R_{tot} of the leak and structure with different leak surface areas when the airborne sound insulation of a leak-proof structure is 30, 50 or 70 dB. For example, in a 10 m2 structure, 0.0001% means a leak of 1x10 mm.

8.8 Dimensioning of façade sound insulation

The required sound insulation of the façade against road traffic noise is usually determined by the planning architect and/or acoustic consultant. For zoning marking, one of the two markings in Fig. **8.8.1** is usually used. Usually, the sound insulation requirement (ΔL) for the façade is predefined in the plan if the purpose of use of the building is known. (Note: Sound insulation is a general term and does not refer to the quantity airborne sound insulation that will come up later in the chapter.) In the upper option, the requirement for the noise level inside the apartment is already subtracted from the outdoor noise level (Eq. **8.8.1**). This requires the planner to know the actual outdoor noise level $L_{A,eq,u}$ and the maximum allowed indoor sound level $L_{A,eq,s}$.

The lower option in Fig. **8.8.1** contains only the outdoor noise level value $L_{A,eq,u,}$ leaving the calculation of the sound insulation requirement ΔL to the architect. He or she must determine the maximum level of indoor noise according to the purpose of use of the $L_{A,eq,s}$ space to be decided later. This option is appropriate in situations where several uses have still been allowed for the building at the planning stage, such as a residential apartment, day-care centre, office space or business premises, but the final purpose of use will not be known until the user is known or a building permit has been applied for.

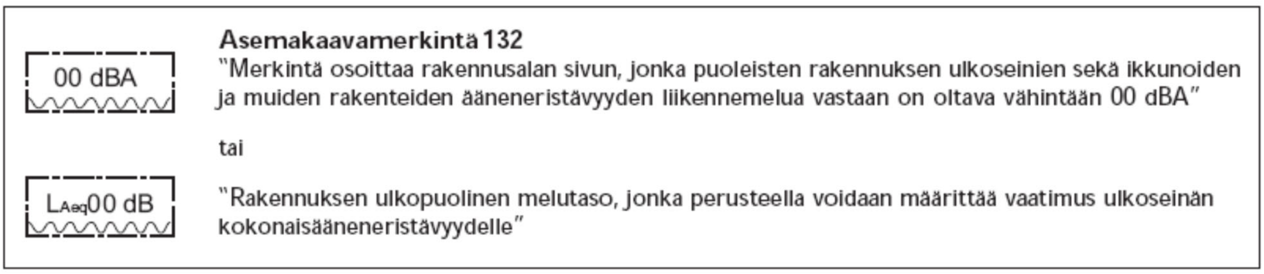

Fig. 8.8.1. Formula entries. The upper marking is the difference between the outdoor and indoor noise level, ΔL, and the lower one is the outdoor noise level $L_{A,eq,u}$.

The soundproofing of the façade should take into account all structural elements, which are usually a wall, window, replacement air valve, balcony door and ventilation window. The upper floor, flues of central ventilation or flue shall be taken into account, especially in areas of aircraft noise.

The finished facades are not subjected to sound insulation measurements in the laboratory. Instead, laboratory measurement results exist for components. If measurement results are not available, calculated values may also be used if they are sufficiently reliable.

The sound insulation of replacement air valves depends on the position of the valve. If the window has a replacement air valve, the sound insulation value should be used in the calculations, where the valve is fully open.

The sound engineering design of the façade structure includes the following objectives:

- the package meets the set sound insulation requirement and
- The components of the façade structure are selected in such a way that there are no major differences between the sound power transmitted through the different components

The Ministry of the Environment (2003) has published a model for dimensioning the airborne sound insulation of facades. A special feature of the dimensioning model is the pursuit of an optimal overall solution. The model is outlined below. The calculations must be made separately for both night and day time, as noise levels and guide values are different for these times.

The requirement for the difference between outdoor noise and permissible indoor noise ΔL (sound insulation requirement) is given by the equation

(8.8.1) $$\Delta L = L_{A,eq,u} - L_{A,eq,s}$$

where $L_{A,eq,u}$ [dB] is the equivalent A-weighted SPL on the façade surface without the reflective effect of the building and $L_{A,eq,s}$ [dB] is the maximum permissible equivalent of environmental noise A-weighted SPL indoors.

Often the environmental noise level is measured right next to the façade surface (less than 10 mm from the surface), which avoids standing waves and makes it easier to attach the microphone. The plane on the wall surface is approximately 6 dB higher than the value $L_{A,eq,u}$ measured at the same point in a free field (without the reflective effect of the building).

If $L_{A,eq,u}$ is obtained via environmental noise modelling methods/software, the buildings located close to the determination point must be non-reflecting.

The requirement for airborne sound insulation of the entire façade $R_{tr,vaad}$ [dB] is determined from the equation

(8.8.2) $$R_{tr,vaad} = \Delta L + K_1 + 7$$

where the correction term K_1 takes into account the shape of the room. Its determination is given in Table **8.8.1.** The requirement for airborne sound insulation of the entire façade, $R_{A,tr,kok}$ [dB], is given by the equation

(8.8.3) $$R_{A,tr,kok} \geq R_{tr,vaad}$$

The requirement for the airborne sound insulation of the window, $R_{A,tr,ikk}$ [dB], shall be set to:

(8.8.4) $$R_{A,tr,ikk} \geq R_{tr,vaad} + K_2$$

where the correction term K_2 [dB] (Table **8.8.1**) takes into account the relative area of windows. The requirement for airborne sound insulation of the wall section $R_{A,tr,wall}$ [dB] is determined from the formula

(8.8.5) $$R_{A,tr,seinä} \geq R_{tr,vaad} + 3$$

The unitnormalized sound level differential of small elements, $D_{n,e,A,tr}$ [dB], is

(8.8.6) $$D_{n,e,A,tr} \geq R_{tr,vaad} + 5$$

It is notable, that in all above equations, R_{Atr} denotes the laboratory value R_w+C_{tr}.

The dimensioning method already includes a safety margin of 4 dB, which means that minor errors and measurement errors during construction do not yet lead to a situation where the sound insulation target would not be reached.

The model of the Ministry of the Environment (2003) does not provide the means to assess how the fall of one component from the target value could be compensated for by a better value of another. Therefore, in practice, the so-called "sponge" is often applied. sound level difference method (RIL 243-1-2007). With fixed choices, it produces similar results to the model of the Ministry of the Environment (Kylliäinen, 2005).

Table 8.8.1. Determination of correction factors K_1 and K_2.

S/S_H	2.5	2	1.6	1.3	1	0.8	0.6	0.5	0.4
K_1 (dB)	5	4	3	2	1	0	-1	-2	-3

$(\Sigma S_i)/S$	0.1	0.13	0.15	0.2	0.25	0.3	0.4	0.5
K_2 (dB)	-6	-5	-4	-3	-3	-2	-1	0

S = julkisivuseinän pinta-ala huoneessa, m^2
S_H = huoneen lattiapinta-aa
ΣS_i = julkisivussa olevien ovien ja ikkunoiden yhteispinta-ala

8.9. Flanking transmission along ductwork

If there is a connection between the rooms via ventilation ductwork, the ductwork can carry significantly more airborne sound from one room to another than through the structures separating the rooms (**Fig. 8.9.1**).

Usually, the flanking transmission along the ductwork can be identified by the colour of the voice, because the sound sounds "tubular". The sound quality is explained by the transverse resonances of the unattenuated ductwork, the most significant of which occurs at the channel limit frequency. Lateral displacement in the ductwork must usually be taken into account if the airborne sound insulation requirement between rooms exceeds 35 dB (R'_w). Lateral displacement is prevented by a muffler.

Fig. **8.9.2** shows a measurement example of a common duct system without silencers between residential apartments. Between overlapping bedrooms, the airborne sound insulation Fig. was measured at 46 dB. When the terminals of the ducts were blocked, the airborne sound insulation number improved to 59 dB, which corresponded to the airborne sound insulation Fig. of the intermediate floor. The sound problem could be solved by installing an additional silencer in the ductwork.

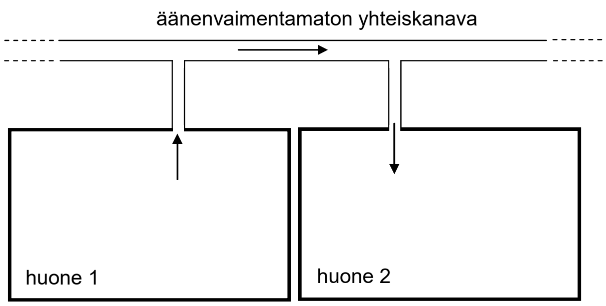

Fig. 8.9.1. Transmission of sound between rooms through ventilation ductwork. In the example, there are no terminals that dampen sound more between the room and the duct than an open channel.

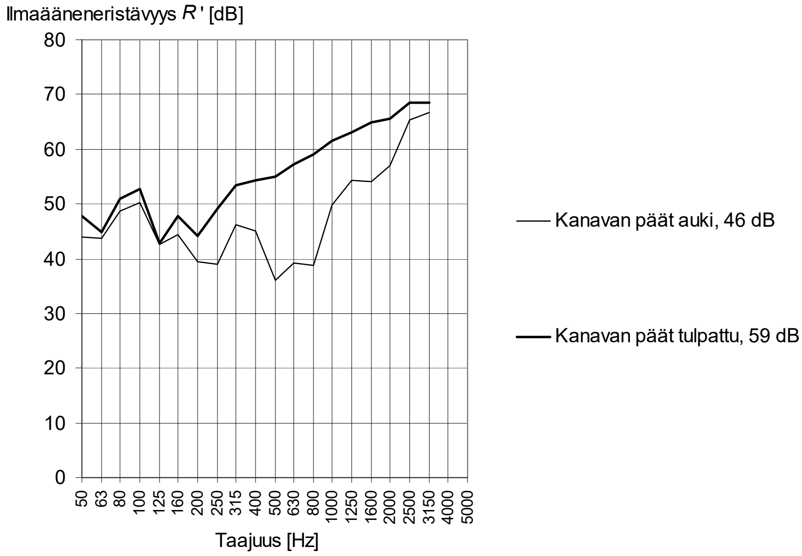

Fig. 8.9.2. The measurement result between residential apartments, where airborne flanking through ventilation ductwork weakened the airborne sound insulation number allowed by the structures by 13 dB.

The airborne sound insulation of the ventilation duct between room 1 and 2, R_d [dB], can be estimated from the equation

(8.9.1) $$R_d = D_1 + D_s + D_d + D_2$$

The output damping D_1 [dB] depends on the damping capacity of the terminal organ in room 1. Product values for output damping are not available because a test standard has not been developed. The output attenuation value may be the end attenuation value of the product in question, D_t [dB], but measured from the duct in the direction of the room, in this case in the wrong direction.

The effect of a silencer, if any, on the duct path is taken into account by the term D_s [dB]. The sound absorption of bends and branches along the duct route is included in the term D_d. When sound is transmitted from the duct to the room, the terminal attenuation value D_t of the terminal shall be applied as the value of D_2.

The airborne sound insulation determined according to equation **8.9.1** refers to the power loss in the duct cross-sectional area according to the definition of sound insulation.

Suppose there is a duct path between the rooms, the airborne sound insulation of which, calculated as above, is R_d. If the airborne sound insulation of the structure separating the spaces is R_p and the area of the structure is S_p [m^2], the joint sound insulation between the rooms is

(8.9.2) $$R_{tot} = 10 \cdot \log_{10}\left[\frac{S_d + S_p}{S_d 10^{-R_D/10} + S_p \cdot 10^{-R_p/10}}\right]$$

where S_d [m^2] is the cross-sectional area of the ductwork.

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9 HEARING PROTECTION

9.1. Noise exposure

From the point of view of hearing protection, the most important thing is to know the total energy of sound that enters the ear canal, as the total energy has been shown to be linked to noise-related hearing damage. To determine the total energy, you need to know the SPL as a function of time, i.e. the average noise level and exposure time. In order to assess the risk of noise-related hearing damage, it is necessary to know the average sound level in question over several years, because the noise level in question. The risk is usually only significant when exposed to loud noise for long periods of time. Sometimes, however, even a single impulse can contain so much total energy that permanent hearing loss is possible.

According to Government Decree 85/2006, the ear canal shall not be exposed to noise exposure of more than 87 dB. In addition, there are two lower action levels of 80 dB and 85 dB.

Daily noise exposure means the A-weighted sound level which, during a nominal working day of 8 hours, gives the same exposure as exposed noise. The daily noise exposure $L_{\mathrm{pAeq,T0}}$ [dB] is determined by the equation

(9.1.1) $$L_{pAeq,T_0} = L_{pAeq,T_e} + 10\log_{10}\left(\frac{T_e}{T_0}\right)\mathrm{dB}$$

where $L_{\mathrm{pAeq,Te}}$ [dB] is the mean sound level in the ear canal of a person over the exposure period T_e [s] and T_0=28 800 s.

Noise exposure is most commonly determined with a recording dosimeter (Fig. **9.1.1**). Determining the type of hearing protector is part of the measurement of noise exposure. If the PPE is not used at 100%, the worker is often asked to keep a PPE diary in order to assess the actual impact of the PPE on noise exposure.

In noise control and hearing protection, the following concepts/quantities are used to describe noise:

- Emission; noise emission from a noise source, which describes the energy that a sound source produces in its environment.
- Immission; the sound level at the point of stay (or workstation), i.e. the level that a noise source or multiple sound sources exert at a given point;
- Personal noise exposure; the average noise level over the entire period during which the noise has been exposed.

Emission is usually indicated in the form of a sound power level. Immission depends on the distance to the noise sources, the noise emission of the noise sources and the attenuation factors between the listening point and the noise source. Immission can vary a lot over time depending on the fermentation of noise sources. Noise exposure, on the other hand, depends on the points where the person has spent their time, how long they have stayed there and what kind of hearing protectors they have used. The protection efficiency [dB] and utilization rate [% of the time] of PPE must be taken into account.

If the worker's points of stay i during the working day and the periods of stay T_i [h] in them are well known and there is a constant sound level $L_{\mathrm{pA,i}}$ [dB], the average sound level can be estimated on the basis of short sound level measurements. This requires that the sound level is constant and the short sample represents the sound level over a long period of time with high certainty. In this case, the average sound level can be calculated using the equation

(9.1.2) $$L_{pAeq,8\mathrm{h}} = 10\cdot\log_{10}\left(\frac{1}{\sum_i T_i}\sum_i T_i \cdot 10^{L_{pA,i}/10}\right)$$

where the time periods T_i = 8 h.

If the noise is of short duration, the noise exposure level is sometimes used. The sound exposure level is the same as the equivalent noise level normalised to a one-second sound event:

$$L_E = 10 \cdot \log_{10}\left[\frac{1}{t_0}\int_{t=0}^{T}\frac{\tilde{p}^2(t)}{p_0^2(t)}dt\right] \tag{9.1.3}$$

where t_0 = 1 s and p_0 = 20 µPa. The sound exposure level can be used to compare noise events of different durations in terms of the risk of hearing damage or annoyance. The A-weighted SPL is most easily determined, in which case the designation is L_{AE}.

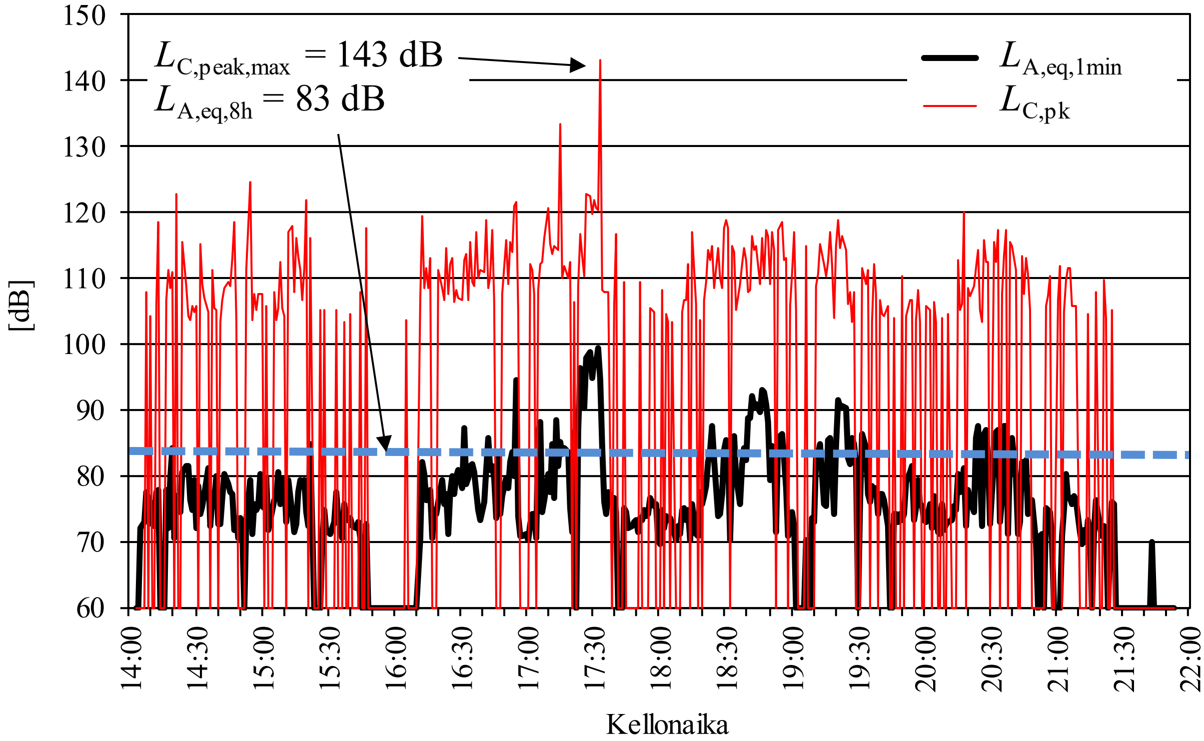

Fig. 9.1.1. Noise exposure is measured in the workplace with a dosimeter consisting of a microphone placed close to the ear and a recorder that fits in your pocket. The curve shows values for Lp,A,eq,1min as a function of time. For the measurement period, at least the equivalent level Lp,A,eq and the peak sound level LC,peak,max shall be determined.

9.2 Hearing protectors and protection effectiveness

If the daily noise exposure exceeds 80 dB, the risk of hearing damage is obvious and hearing protection should be initiated at the latest. Types of protectors have been developed for different needs, because hearing protectors should be as comfortable as possible. Improving comfort can improve the utilization rate of hearing protectors, which is critical for protection efficiency. The main types of hearing protector products are:

- muffle
- plug
- helmet

If the hearing protectors do not have electronics, the protectors are called passive. Based on electronics, protectors can be divided into four groups, for example:

- Level-dependent communicative hearing protectors: the protector amplifies external sounds when the level is below a threshold value (e.g. for hunting use).
- Safety-related communication hearing protectors: communication is possible between the protectors, or the protector is connected, for example, with a mask. Phone. The level of communication inside the protector is not limited.
- Consumer electronics hearing protectors: intended for e.g. for listening to music via radio or telephone. The SPL is limited to 82 dB L_A.
- Noise-cancelling hearing protectors: suitable for noise with strong low-frequency components.

The protection efficiency D [dB] of hearing protectors is determined in laboratory conditions in the octave bands 125–8000 Hz. It is determined by a subjective method. The subjects' hearing threshold (the lowest audible sound level) is measured for pink octave noise in a known sound field without and with protectors, and the shielding effectiveness is obtained from the difference between these hearing thresholds.

Fig. 9.2.1 shows the sound attenuation values of the most common types of PPE, i.e. cup protectors and plug protectors at different frequencies. Sound absorption usually increases with increasing frequency. However, values above 50 decibels do not seem to be achieved. This is due to bone conduction (lateral displacement through the skull and its openings to the auditory organs).

The damping ability of the cup protector is largely influenced by the mass and tightness of the cup. In practice, the interior of the protector functions as a constant pressure chamber up to a certain limit frequency. Below the limit frequency, the damping capacity of the protector is largely determined by the mass of the protector's wall. Above the limit frequency, attenuation is also affected by the absorption of the internal space of the protector.

In the laboratory, the conditions are carefully controlled and the protective equipment is carefully placed on the head. In practice, the protection effectiveness of PPE during use is reduced by several factors related to use, such as:

- incorrect installation in the head or ear canal.
- Untightness: between the protector and the skin hair, beard, headdress or eyeglass headbands.
- poor condition or freezing of protective seals

As a result of sound leaks, the protection efficiency with plug protectors can drop to almost zero. The protection effectiveness of the protector during use can be assessed with small microphones that fit into the ear canal. According to studies, workplaces where training in the use of PPE has been organised have achieved better protection efficiency than workplaces without training.

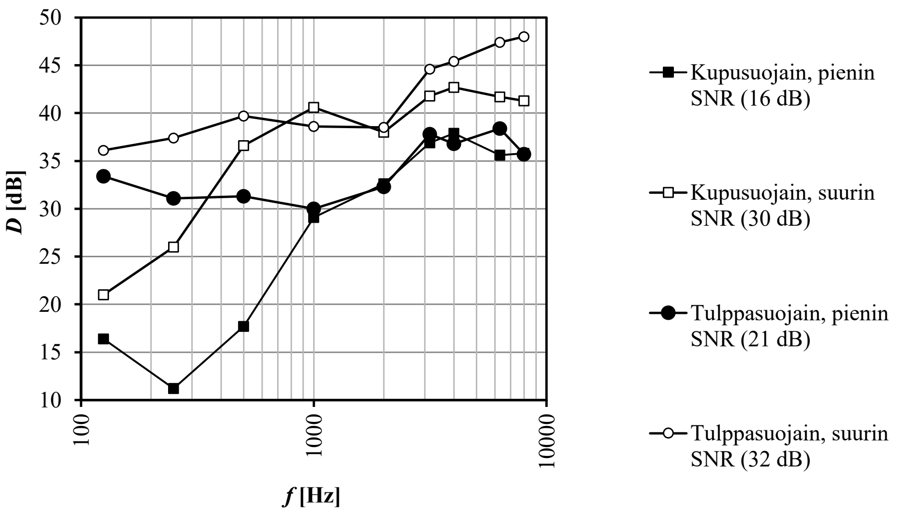

Fig. 9.2.1. Frequency-dependent shielding efficiency values for products with the lowest and highest damping performance from a product manufacturer.

9.3 Dimensioning of hearing protectors

Efforts should be made to select the hearing protector so that the sound level **L'_A=75±5 dB** prevails at the entrance to the ear canal. If the A sound level at the entrance to the ear canal drops below 70 dB, speech communication begins to become more difficult and vital signs (bite, blood circulation) begin to be heard, decreasing the comfort of use. In addition, a safety margin of 3□5 dB must be reserved to compensate for the decrease in protection efficiency caused by mild misuse.

If the workstation is variable, it would be good to have several protective equipment for different environments. This is often not possible. In this case, the dimensioning of the protector should be made according to the loudest noise emission mission.

To choose the right hearing protector, you need to know the noise you are exposed to and the protection efficiency values of the PPE options available. Fig. **9.3.1** shows a typical product fiche with information for different calculation methods. Octave band and SNR methods are commonly used.

Frequency (Hz)	125	250	500	1000	2000	4000	8000
Mean Attenuation (dB)	17.0	24.0	29.5	36.9	37.3	39.3	35.4
Std. deviation (dB)	3.2	2.0	2.6	3.3	4.9	3.2	3.9
Assumed Protection Value (dB)	13.8	22.0	26.9	33.6	32.4	36.1	31.5

EN 352-1:2002 **SNR=32 dB H=34 dB M=29 dB L=22 dB**

Fig. 9.3.1. Sound attenuation specification in the product sheet for a hearing protector.

Octave band method. The octave band method is the most accurate of the dimensioning methods. It requires the noise to be known in octave bands. This method is applied to protect against a particular type of particularly loud noise, such as noise from machinery producing a certain frequency at a fixed workstation. The SPL inside the protector is given by:

(9.3.1) $$L'_A = 10 \cdot \log_{10}\left[\sum_{i=1}^{7} 10^{(L_{Z,i}+L_{A,i}-D_i)/10}\right]$$

where $L_{Z,i}$ [dB] is the external noise level at frequency i, $L_{A,i}$ [dB] is weighting of the A filter at frequency i and D_i [dB] (APV in product descriptions) is the protection efficiency of the protector at frequency i.

The SNR method is the most inaccurate, but the easiest. In most cases, it is sufficient. The *SNR* attenuation value describes the attenuation of the protector against average industrial noise. The value inside the protector is given by:

(9.3.2) $$L'_{A,eq} = L_{C,eq} - SNR$$

9.4. Degree of use of hearing protectors

The degree of use of hearing protectors refers to the ratio of the time of wearing the hearing protector to the exposure time as a percentage. If the noise is time T, and then the period T_D *is* accompanied by protective equipment with a nominal efficiency of D [dB], the utilisation rate is U [%]

(9.4.1) $$U = 100\frac{T_D}{T}$$

The equivalent SPL is given by the equation

(9.4.2) $$L_{eq,T} = 10 \cdot \log_{10}\left[\frac{1}{T}\sum_i T_i 10^{L_i/10}\right]$$

where T_i [s] is the time spent in each noise situation and L_i [dB] is the sound level corresponding to that time. Consider the ideal case where the same sound level $L_{eq,out}$ prevails in a person'*s* environment at all times T. In this case, the equivalent level in the ear canal without protectors is $L_{eq,out}$. If a person wears hearing protectors for the time T_D, the total noise time T ($T_D<T$) gives the equivalent level in the ear canal at time T, $L_{eq,ear}$ [dB] is the sum of the two sound levels

(9.4.3) $$L_{eq,ear} = 10 \cdot \log_{10}\left[\frac{1}{T}\left(T_D 10^{\left(L_{eq,out}-D\right)/10} + \left(T - T_D\right)10^{L_{eq,out}/10}\right)\right]$$

The actual protection efficiency in the ear canal D' [dB] with time T is of the form according to the energy principle

(9.4.4) $$D' = L_{eq,out} - L_{eq,ear}$$

It can also be presented in the form of:

(9.4.5) $$D' = 10 \cdot \log_{10}\left[\frac{1}{T}\left(T_D 10^{-D/10} + T - T_D\right)\right]$$

Fig. 9.4.1 shows the actual protection efficiency achieved by the PPE at three different rated protection efficiency values. Nominal protection efficiency can only be achieved if the utilization rate is 100%. The type of PPE is of little importance if the utilisation rate is below 90%. If the utilization rate is less than 50 %, no PPE achieves actual protection efficiencies above 3 dB.

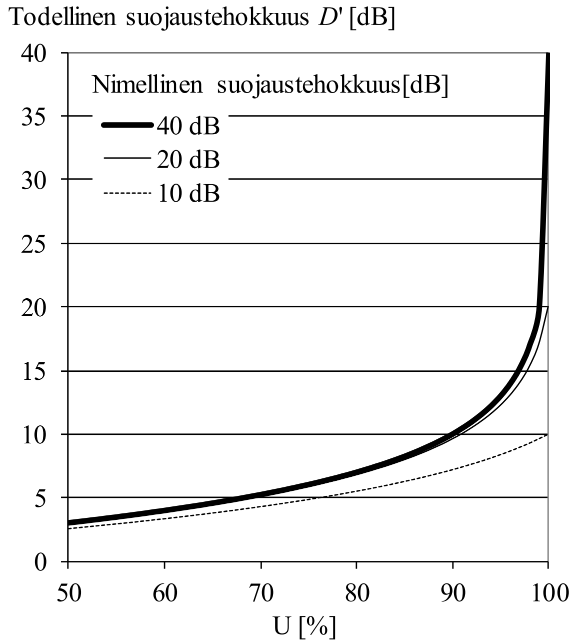

Fig. 9.4.1. Contribution of hearing protector use *U* [% of exposure time] to actual protection performance.

LITERATURE

10 INDUSTRIAL NOISE ABATEMENT

10.1 Means of noise abatement

This chapter examines noise abatement in the interior of industrial halls. Noise abatement can be targeted at the sound source, the path of sound transmission or the listening area (Fig. **10.1.1**). In the following chapters, we will examine the means of noise abatement shown in this figure.

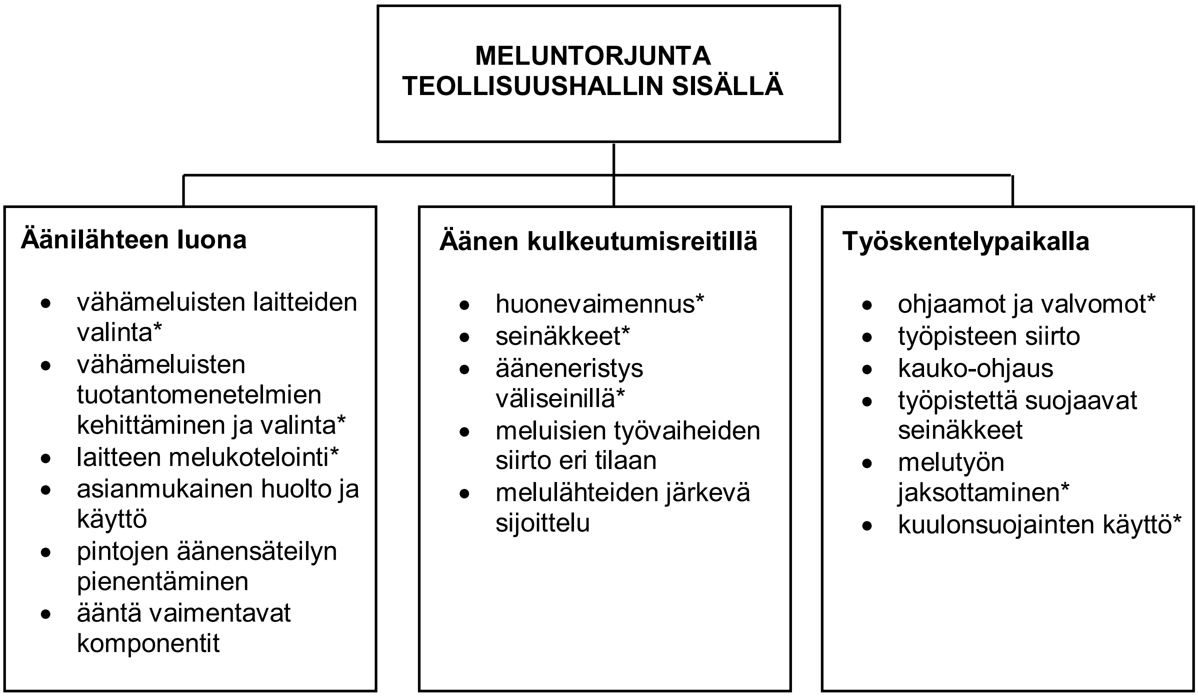

Fig. 10.1.1. Breakdown of technical noise abatement measures.

10.2. Indication of the noise emission of machinery

The purchase of low-noise machinery is probably the most effective means of noise abatement if low-noise machines are available. Reducing noise emissions also has a direct impact on noise exposure.

The noise emission of the machine is primarily indicated by the sound power level L_W. Large equipment is often used to indicate the emission SPL, L_p, in the operator's position or at another specified location.

The noise emission declaration shall be drawn up in accordance with ISO 4871. The noise emission declaration may be submitted as a one- or two-digit noise emission notification value.

The measured noise emission L is either the A-weighted sound power level, the average A-weighted emission SPL or the C-weighted peak emission SPL determined from measurements. The measured values can be determined either for a single machine or as an average value for several machines.

Uncertainty K is the numerical value of the measurement uncertainty associated with the measured noise emission value. The numerical value of K can be 1–5 dB depending on the number of devices measured and the accuracy of the measurement standard.

The one-digit noise emission declaration value L_d is the sum of the measured noise emission and the associated uncertainty, rounded to the nearest decibels: $L_d=L+K$. A two-digit noise emission declaration value means that L and K are reported separately. Fig. **10.2.1** shows an example of a noise emission declaration.

Machine model number, operating conditions, and other identifying information: **Analyzer, serial number: xxxxxx**				
DECLARED DUAL-NUMBER NOISE EMISSION VALUES **in accordance with ISO 4871**				
operating mode	**mode 1**	**mode 2**	**mode 3**	**mode 4**
A-weighted emission sound pressure level, L_{pA} (ref. 20 μPa) at operator's position, in decibels	52	54	54	55
Uncertainty K_{pA} in decibels	4	4	4	4
A-weighted emission sound pressure level, L_{pA} (ref. 20 μPa) at bystander position (behind the analyzer), in decibels	55	56	56	56
Uncertainty K_{pA} in decibels	4	4	4	4
Mode 1: Startup and initialization Mode 2: Action 1 Mode 3: Analyzation process Mode 4: Shutdown and sleep Values determined according to noise test code given in ISO 11202.				
NOTES: 1. The sum of measured noise emission value and its associated uncertainty represents an upper boundary of the range of values which is likely to occur in measurements. 3. The declared L_{pA} values include the background noise correction K_{1A}= 0 dB (no correction needed) 3. The declared L_{pA} values include the local environmental correction K_{3A}= 2 dB which was determined using the measured reverberation time of the test room (ISO 11202 Annex A).				

Fig. 10.2.1. Example of a noise emission declaration for a chemical analytical instrument. A noise emission declaration can indicate the combined effect of the sound level of several devices. The results are presented as a two-digit declaration value.

10.3. Machine noise analysis

If low-noise machinery cannot be procured, noise abatement measures must be applied on site. Technically and economically sound noise abatement planning requires knowledge of which sound sources are the loudest and to what extent exposure to which sound source. Noise abatement measures are easiest to justify if they reduce the personal noise exposure of as many workers as possible. In most cases, the most effective noise reduction is achieved if the machine itself can be damped. In this case, you need to know where the sound radiates into the workspace and what causes the noise. From the source of noise, it is necessary to find out:

- main internal sources;
- the main pathways of advancement from these sources;
- the main sound-emitting parts of the machine.

10.3.1 INTERNAL SOURCES

The first analysis of internal sources can be carried out on the basis of the spectrum of the sound emitted, combined with information about the structure and operation of the machine (possible internal sources, rotational speeds, number of wings, poles and gear teeth, etc.). The cause of spikes in the spectrum can often be identified in this way. Additional information can be obtained from the time history of the emitted sound. In both approaches, varying the operating parameters of the machine, and especially the running speed, can be a very useful way to understand the source of the signals. Auditory assessment is often a faster way to obtain information about the noise source than measurement.

If it is not yet possible to draw a conclusion about the internal sound source from the spectrum and time history, the next impacts may be

- sequential switching off of internal sources (if possible);
- sequential interruption of sound propagation from internal sources to machine interfaces (by isolating air, liquid or hull sound)

For approaches, see Table **10.3.1.**

10.3.2. PATHWAYS

The most straightforward way to obtain information on the relative importance of different sound paths is to increase insulation successively along these routes, for example by adding more efficient upholstery, insulation or dampers. However, these are not effective means at low frequencies. Quantitative information on pathways can be obtained experimentally using theoretical models or using a mixed approach (Table **10.3.2**).

10.3.3. RADIATION

The analysis may relate to radiation to air, gas in the tube, liquid in the tube, structure or a combination of these. For the analysis of radiation into the air surrounding the machinery, the methods in Table **10.3.3** may be applied .

Table 10.3.1. Analysis of internal sources.

	Designation	Description	Comments
1	Frequency spectrum analysis	Determination of the spectrum of sound at an arbitrary point in the far field.	If the mechanical structure and operating principles of the machine are known, frequency analysis can show which internal sources are important.
2	Time analysis of radiated sound-pressure	Determination of sound pressure as a function of time at some point in the far field.	In addition to frequency analysis, space-time analysis provides additional information related to excitation mechanisms.
3	Shutting down component sources	Sometimes it is possible to operate the machine while keeping certain internal sources turned off. If sound pressure is measured in a distant field (as a function of time and/or frequency), the proportion of the subsource to the whole can be determined.	
4	Sequential covering or cutting off of internal sub-sources	It is often possible to reduce the propagation of sound from a specific subsource to the interfaces of the machine. Measure the effect of the procedure on sound pressure (either as a function of time and/or frequency) in an appropriate environment.	An airborne partial sound source can be installed inside a suitable enclosure. The part source that generates body noise can be isolated from the machine structure with a suitable flexible element.
5	Variation of operating parameters	Varying the load or speed. Measure the effect of the procedure on sound pressure (either as a function of time and/or frequency) in an appropriate environment.	
6	Prankster analysis	Measure the acceleration at the point of the machine interface, synchronize the measurement with the rotating axis. Use appropriate software for analysis.	For rotating machines, this is a very suitable method for separating internal sources from the standard response of the machine structure.
7	Direct substitution method	Replace the partial source with an artificial source whose properties are known. Measure the radiated sound level.	Method for determining the volume (air sound or body sound) of a subsource (equivalent) source;
8	Reverse substitution method	The reverse method in step 7, using a speaker at the receiving point and a microphone or accelerometer at the location of the internal source.	A comfortable option when the internal source cannot be replaced by an artificial source. The default is the linearity of the system.
9	Correlation or coherence method	Determine the correlation between the radiated sound pressure and the reference signal from the subsource.	The usefulness of this method may be limited, since in practice it can be difficult to obtain a suitable reference signal
10	Sound intensity measurements	Determine partial audio power from any internal audio sources.	A suitable method for determining the radiation of direct air sound of various components.

Table 10.3.2. Methods for analyzing sound propagation inside the machine.

	Designation	Description	Comments
1	Direct transfer function measurement	Use an artificial source at the site of the internal source and measure acceleration at machine interfaces or sound pressures in the far field	
2	Reverse transfer function measurement	The reverse method of step 1. The locations of the source and recipient are exchanged.	Useful when it is impossible to install an artificial source inside the machine
3	Sequential displacement of displacement routes inside the machine	Use isolation techniques for different sound propagation paths	The method is suitable for identifying important sound paths
4	Power flow measurement	Use special techniques to determine the flow of air, liquid or body sound power along a particular route	Complex methods; requiring special expertise.
5	Mode analysis (experimental)	Measure the oscillatory behavior of the structure deterministically.	Not suitable for the resonant response frequency range

Table 10.3.3. Methods of analysis of sound radiation.

	Designation	Description	Comments
1	Masking selected parts that emit sound	Cover the radiating machine parts in succession and measure the radiated sound pressure	Suitable for determining the direct proportions of air noise of various machine parts
2	Sound intensity measurements	Take samples from the surface of the machine with the intensity sensor	
3	Near-field sound pressure measurements	Measure sound pressure close to the radiating surface	Easier to perform than 2 but less accurate
4	Directional sound pressure measurements	Measure sound pressure in a free field with a strongly directional microphone or microphone mesh	Straightforward method, but rarely used
5	Vibration measurements on surfaces emitting air sound	Measure the vibration on the radiant surface. The radiated power is given by the equation $W = \rho c S \overline{v}^2 \sigma\omega$	Suitable if σ there is a good estimate of radiation efficiency
6	Mode analysis	Analyze the deterministic behavior of external interfaces	Suitable at low frequencies if radiation efficiency is known

10.4. Noise abatement of machinery

To make it easier to understand the mechanism of sound generation and propagation, it is necessary to create simple acoustic models. The models form the basis for noise abatement measures at the design stage. A universal approach is to differentiate as follows:

- internal sources;
- propagation routes inside the machine;
- radiation from its interfaces.

Internal sources and transmission pathways can each be divided into three categories according to the medium:

- air sound
- liquid sound
- Body sound

For the purposes of this chapter, radiation is considered to be limited to air.

Fig. 10.4.1 shows the principle of acoustic modelling of the machine. The engine is the only internal source. It generates air sound and trunk sound. Three internal pathways of penetration can be distinguished:

- through the air inside the body into the opening;
- through the air inside the body to the walls of the frame;
- through wall brackets to the frame.

Radiation occurs from the opening and walls of the body. The total sound power emitted by the machine is the sum of these three components.

A systematic approach begins with an assessment of the relative importance of the above-mentioned elements. The next step is to examine the blocks in Fig. **10.4.1,** looking for opportunities to reduce the intensity, transmission and/or radiation of the source. This should be done taking into account other elements of the machine design process.

When considering machine noise, two modes of noise generation must be distinguished: flow dynamic (gas and/or liquid) and mechanical generation. Flow dynamic noise is caused by random fluctuations in fluid pressure and speed. Examples include combustion processes, fans, discharge pipes and hydraulic systems. Mechanical noise is caused by vibrations of machine parts, which are caused, for example, by dynamic forces caused by shocks or unbalanced masses. The vibrations are transmitted to noise-emitting surfaces, such as the machine housing or the workpiece. Examples include gears, electric motors, hammers, shakers and mechanical printing presses.

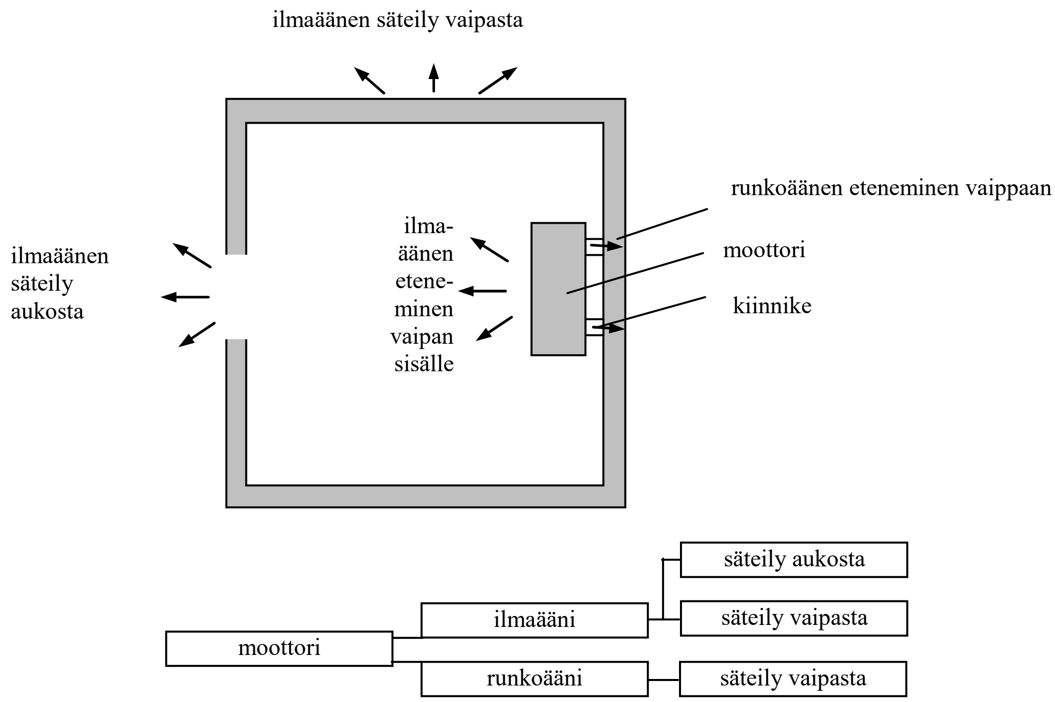

Fig. 10.4.1. A simplified machine consisting of an electric motor and a frame with an opening.

Examples of fluid dynamic noise reduction include:

- attenuation of periodic pressure variation at the excitation point
- reduction of flow rates
- Avoiding sudden pressure fluctuations
- Efficient design of flow-through components

Examples of mechanically generated noise reduction include:

- reduction of excitable dynamic forces, e.g. using elastic layers to extend the duration of the impulse caused by the impact
- reducing the vibration rate of the machine at the excitation point with a certain dynamic force, for example, by using stiffeners or additional masses (inertia body);
- damping the transmission of vibration (body sound) from the tuning point to sound-emitting surfaces, for example, using elastic materials with strong internal damping
- Using soundproofing cladding
- attenuation of sound emitted by an oscillating structure, e.g. by using:
    - thin frame walls instead of rigid thick walls
    - damping layers on the surface of thin metal sheets
    - perforated sheets of metal, provided that sound insulation is not required;

10.5. Low-noise production processes

The following procedures are recommended for noisier production processes:

- Avoid shocks and fast movements using smooth motion (slow progressive motion) and limit impact noise by reducing impact speeds (e.g. reduce drop height, use smaller masses) and use damping materials on impact surfaces (Fig. **10.5.1**)
- Avoid using ductwork with flow restriction barriers; Choose large-radius bends or design a continuous cross-sectional system instead of discontinuous
- Use multiple nozzles in discharge openings instead of one large nozzle to reduce turbulence at the flow margins
- Avoid speeds close to the speed of sound and prevent cavitation by using multiple pressure relief valves
- Use plastic drive wheels instead of metal ones if the mechanical load requirement allows this.

- Install a gear with crooked teeth instead of a straight gear
- Make sure all rotating masses are balanced
- Choose low-noise bearings (friction bearings are usually quieter than roller bearings)
- Choose materials that provide the best combination (e.g., plastic/steel) and surface coating of frictional contact elements
- Use materials with high internal damping (e.g., gray cast iron, sandwich sheets, plastics)
- Limit the propagation of hull noise to sound-emitting surfaces

In some cases, it is more appropriate to replace an old noisy machine with a low-noise one instead of implementing these retrospective noise abatement measures (Table **10.5.1**).

There are also noisy activities that are not associated with stationary machines, for example, sounds from the operation of hand machines. These can often be the dominant sources of sound in the office. By carefully selecting tools or work arrangements (e.g. muffled hammers, padded workbenches, low-noise grinding panels, magnetic damping mats, etc.), significant noise reduction can be achieved (Fig. **10.5.2**).

Table 10.5.1. Examples of alternative low-noise processes

Voimakasmeluiset prosessit	Vähämeluisemmat prosessit
lyöntiniittaus	paine- tai rullaniittaus
paineilma- tai polttomoottorivälitys	sähkökäyttöinen välitys
reikien tekeminen tai leikkaaminen esim. kiveen tai betoniin iskukoneella	poraan tai timanttihampailla varustetun pyörösahan terään sovitettavien koneiden käyttö
tyssäysmeisti	kartio- tai suorapuristus
työntöleikkaus	vetoleikkaus
virtauskuivaaminen	säteilykuivaaminen
plasma-happileikkaus	vedenalainen plasmaleikkaus
iskuleikkaus, stanssaus	laserleikkaus
perinteinen TIG/TAG hitsaus	TIG/TAG-suojakaasuhitsaus
liekkikarkaisu	laserkarkaisu
niittikiinnitys	painekiinnitys
iskumuotoilu	hydraulinen puristus
pistehitsaus	saumahitsaus
hionta kivilaikalla	nauhahionta
ketjuvälitys	hihnavälitys
niittaaminen	liimaaminen
sahaaminen	leikkaaminen
leikkaaminen	poraaminen
takominen	valaminen
vierittäminen tai pudottaminen	nostaminen tai laskeminen
sovittaminen lekalla	tuotannon suunnittelu siten, ettei lekaa tarvita

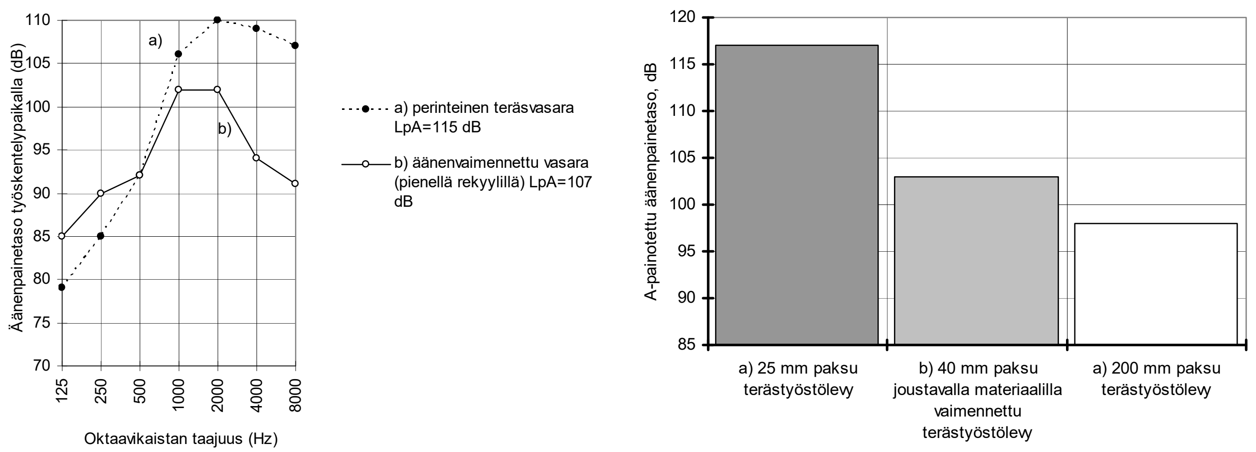

Fig. 10.5.1. Examples of hammer noise reduction by changing the hammer type or increasing the mass of the object to be machined.

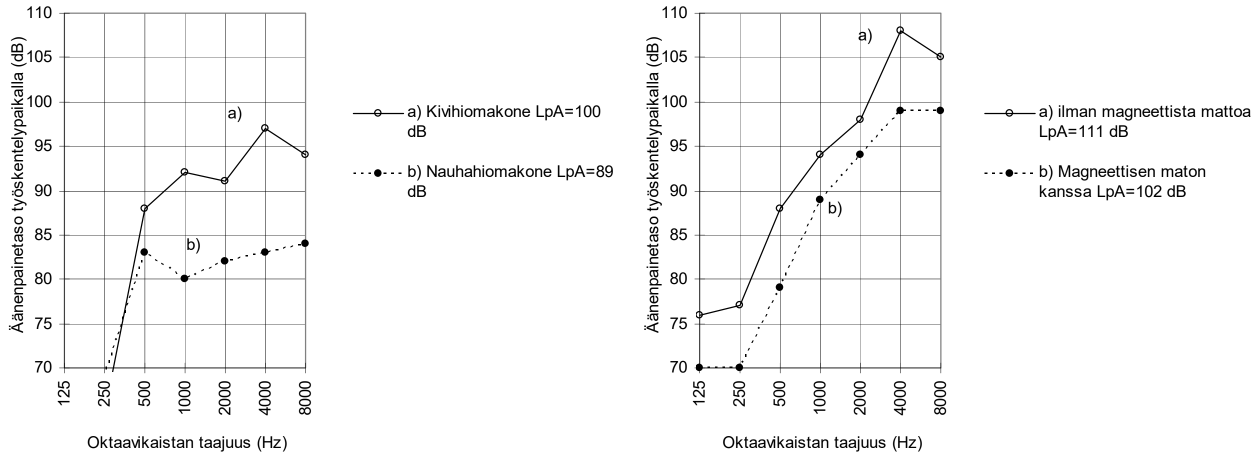

Fig. 10.5.2. On the left, an example of the SPL during grinding when cleaning the body of a cast-iron electric motor. On the right, an example of the effect of increasing the internal damping of the piece to be sanded.

10.6. Noise protection with enclosures

Soundproofing a machine by enclosure is a very common means of sound absorption. Since enclosure is often an expensive and bulky solution, it should be considered whether a more economical end result can be achieved by renewing, moving or subdividing the machine. The following boundary conditions must be taken into account in the design of the enclosure:

- Impact of enclosure structure on production, speed and quality
- ease of maintenance; Dismantling and opening
- ergonomics, for example, in terms of lighting conditions or moving parts of the enclosure
- difficulty in drift
- ventilation to prevent overheating
- Other safety factors
- Choosing a rational location for the sound source
- Effects of enclosure on the environment
- Electrical and automation design
- Locations and quantities of grommets
- the need and places for doors and windows

The enclosure should be designed so that the sound attenuation achieved is sufficient at all frequencies. In order to assess adequate sound attenuation, information is needed on the background noise level of the environment at a typical time of use. The chassis sound absorption should not be dimensioned more than 10 dB below the ambient sound level.

Sound absorption is influenced by the following factors:

- soundproofing R_w of a solid wall structure and the resulting transmission coefficient, τ_w
- average absorption ratio of the inner surface of the housing α_w
- the ratio of the total area S_a of the openings (apertures) to the wall area S_w of the enclosure
- Aperture transmission coefficient τ_a

Since the enclosure usually has at least one window and door in addition to the wall, the combined sound insulation R_{tot} is assessed.

A simple model for estimating the sound absorption generated by the housing is shown below (Fig. **10.6.1**). Suppose the machine is mounted on vibration insulators, in which case there is no need to pay attention to body noises. In addition, it is assumed that a diffuse sound field prevails inside the case. In this case, the wall surface is hit by sound intensity I

(10.6.1) $$I = \frac{p^2_{rms}}{4\rho_0 c_0}$$

where p [Pa] is the average sound pressure inside the case. If the power of the sound source is W [W], the sound power absorbed by the wall surfaces, penetrating the wall surfaces and penetrating the aperture at equilibrium is equal to the power of the sound source:

$$W \approx I\left[S_w(\alpha_w + \tau_w) + S_a\tau_a\right] \tag{10.6.2}$$

The hole is permeated by sound power W_a

$$W_a = S_a I \tau_a = \frac{S_a W \tau_a}{S_w(\alpha_w + \tau_w) + S_a\tau_a} \tag{10.6.3}$$

The wall is permeated by sound power

$$W_w = S_w I \tau_w = \frac{S_w W \tau_w}{S_w(\alpha_w + \tau_w) + S_a\tau_a} \tag{10.6.4}$$

The transmission coefficient for the entire enclosure is

$$\frac{W_t}{W} = \frac{S_a\tau_a + S_w\tau_w}{S_w(\alpha_w + \tau_w) + S_a\tau_a} \tag{10.6.5}$$

and the total level reduction due to the enclosure is

$$D = 10\log_{10}\frac{W}{W_t} = 10\log_{10}\frac{S_w(\alpha_w + \tau_w) + S_a\tau_a}{S_a\tau_a + S_w\tau_w} = 10\log_{10}\frac{\frac{S_w}{S_a}(\alpha_w + \tau_w) + \tau_a}{\frac{S_w}{S_a}\tau_w + \tau_a} \tag{10.6.6}$$

In general, in the calculations it is safe to assume that the sound insulation of the openings will be negligible, i.e. τ_a=1. This is very true for openings with hard-surfaced walls.

Efforts should be made to implement openings so that the walls are sound-absorbing. In the best case, there are also bends in the opening, which means that more sound hits the walls.

The transmission coefficient of the aperture is obtained from the level reduction of the aperture D_a by the equation

$$\tau_a = 10^{-D_a/10} \tag{10.6.7}$$

The longer and more winding the trough, the better sound absorption is achieved. The sound absorption of the sound-absorbing and winding gland improves with increasing frequency. If there are no bends in the penetration, which absorbs from its walls, the sound absorption is slightly reduced at high frequencies.

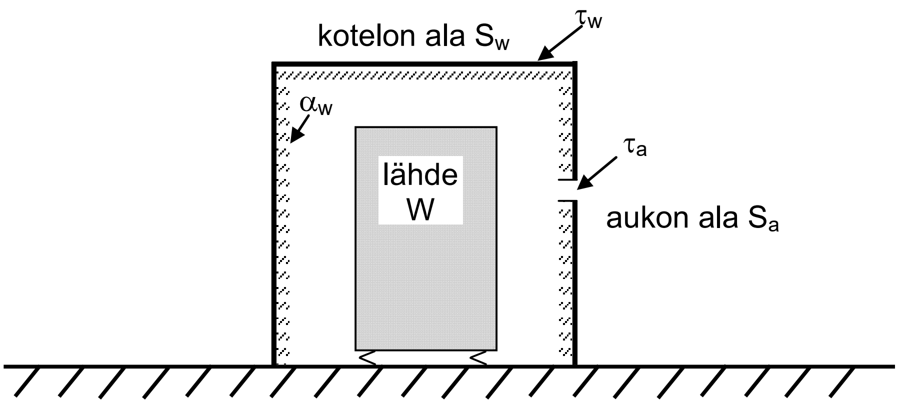

Fig. 10.6.1. Parameters of a simplified noise reducing enclosure.

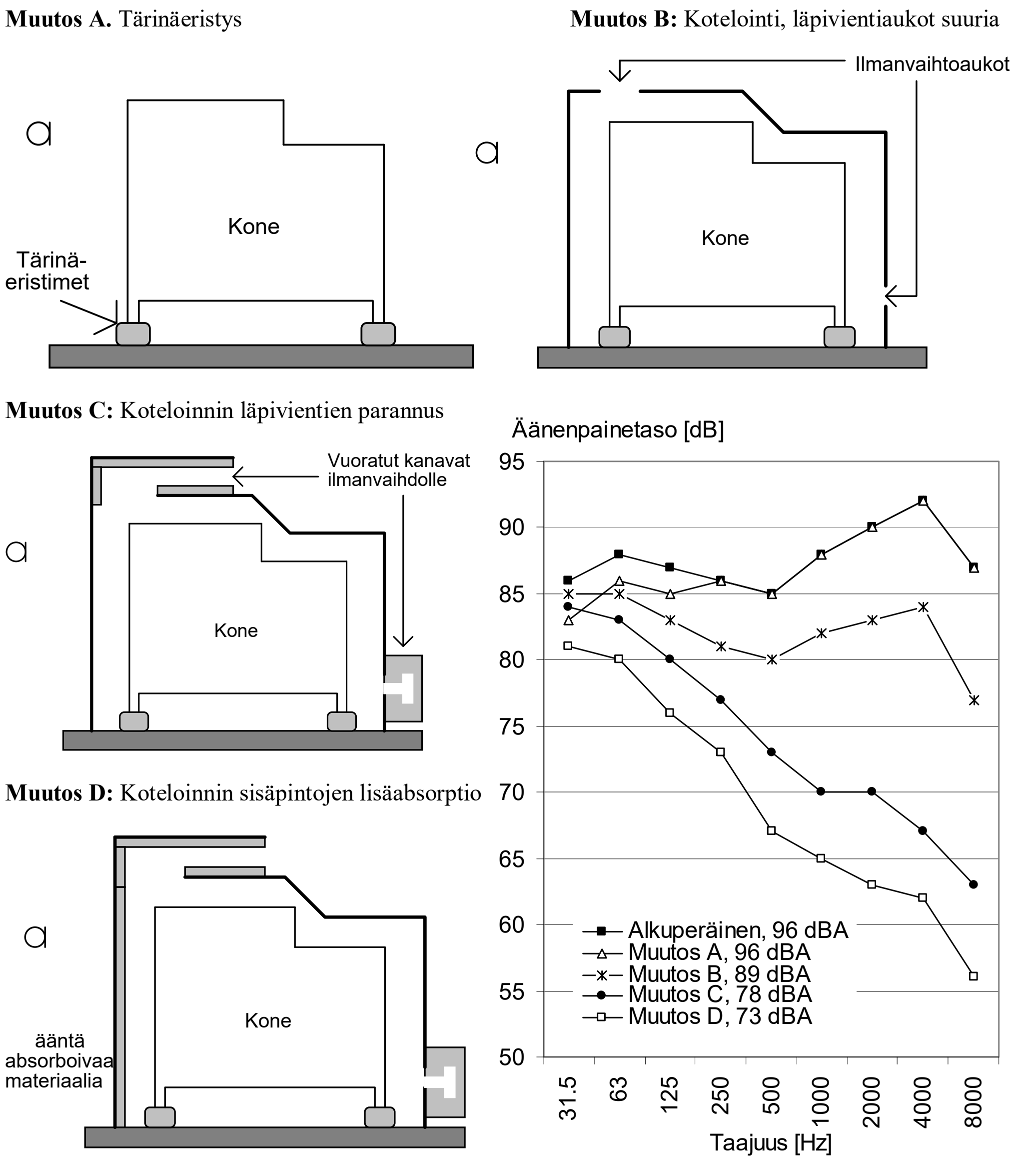

Fig. 10.6.2. Typically achievable noise reduction with various machine and enclosure arrangements.

In reality, the sound field inside the enclosure is nowhere near the diffuse sound field, because the walls can be absorbent and the volume of the enclosures is small. In addition, the wall surfaces of the case are hit by uneven sound intensity. If possible, the grommets should be placed at points where the sound intensity is low.

Fig. **10.6.2** shows an example of sound absorption achieved by different housing solutions. In the example, sound absorption starts with vibration isolators and proceeds through a hard-surface housing to an absorbent housing, including sound-absorbing openings. In the best case, the sound absorption is 25 dB L_A. In practice, it is seldom possible to achieve a better result than this if there are leaks in the case.

The enclosure should be dimensioned to a reasonable level. The sound attenuation should not be dimensioned higher, which is the background noise level of the room without a housing machine. Excessive sound absorption unnecessarily increases costs. For example, a thin sheet metal structure is usually sufficient for soundproofing a wall and a sound-absorbing coating for the inner surface.

10.7. Building acoustic measures

Building acoustic measures include screens, coating of room surfaces with absorption materials and compartmentation of walls. Loudly noisy machinery should be soundproofed in a separate compartment from low-noise machinery where production processes allow.

The cost of building acoustic procedures is usually high due to the large number of materials or structures. The design of walls and screens must take into account, among other things, the management of material flows in production.

Often, hangar cranes or forklift traffic prevent the construction of entire walls or screens and leave leaks in them, so the sound absorption rarely exceeds 10 dB.

Building acoustic measures are discussed in other chapters of the textbook:

- Increase in room absorption
- Airborne sound insulation of wall structures
- Noise screens
- Vibration insulation

As a rule of thumb, room absorption and noise walls can affect sound levels between 3 and 15 dB, depending on the distance to the sound source. Near the sound source, noise cancellation may be negligible. If there are plenty of sound sources, the sound absorption effect will not be perceived as well. Examples of this are given in RIL 243-4-2011.

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11 VIBRATION AND SHOCK

11.1 Introduction

Vibration usually refers to oscillations that occur in a solid body when viewed from a technical point of view. Body sounds are vibrations, but body sounds are spoken of when an object acts as a transmitter or emitter of air sound. Vibration turns into vibration when it can be sensed as the movement of an object. Vibration is a subjective experience of vibration. Body sound must be heard by sound generated by vibrations directly applied to the structure. Rattling is called a secondary body sound caused by vibration, the frequency of which is higher than that of vibration. For example, there is the jingling of wine glasses in the cupboard as the train passes by.

Even the slightest vibration can damage buildings, machinery and equipment. Vibration is harmful to health only if it can be sensed as vibration. Low-frequency vibrations below 1 Hz are experienced as wobble or wobble. Strong vibration exposure can cause direct bodily health hazards to humans. Weaker vibrations in buildings can cause discomfort and distraction.

This chapter covers the basics of vibration, its isolation, and vibration assessment and control.

11.2 Fundamental quantities

Vibration is the back and forth harmonic movement of a piece. In harmonic oscillation, the deviation *s* [m] of the body, corresponding to the distance to the equilibrium position *s*=0, occurs according to the equation

(11.2.1) $$\mathbf{s}(t) = \hat{s}e^{i(\omega t+\varphi_s)}$$

where φ_s is the phase shift, ω is the angular frequency [Hz], and *t* is the time [s].

Depending on the application, one of the following quantities is used for vibration: deviation *s* [m], velocity *v* [m/*s*] or acceleration *a* [m/s^2]. There is a traditional Newtonian dependence between these quantities:

(11.2.2) $$\mathbf{a}(t) = \frac{d}{dt}\mathbf{v}(t) = \frac{d^2}{dt^2}\mathbf{s}(t)$$

so

(11.2.3) $$\mathbf{a} = i\omega\mathbf{v} = -\omega^2\mathbf{s}$$

Thus, the same vibrational signal can be represented in three different ways, which can be derived from each other. Fig. **11.2.1** shows the relationship between quantities in the case of an oscillatory signal.

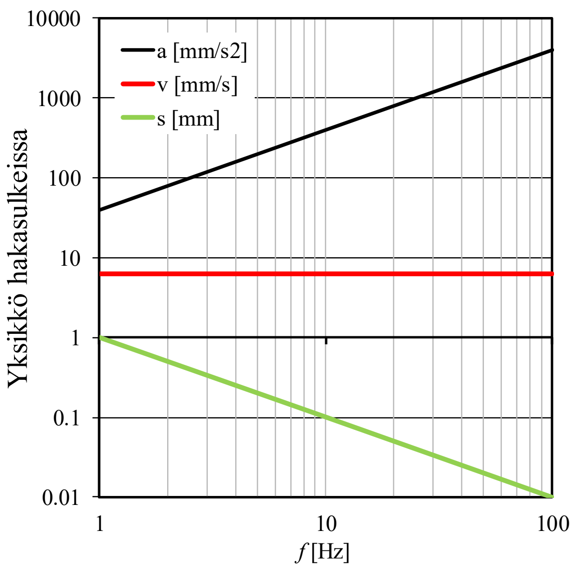

Fig. 11.2.1. The spectra of deviation, speed and acceleration of the oscillation depend on each other.

11.3. Measurement and assessment of vibration exposure

Guideline values for vibration are usually given as absolute values. Decibel representation is mainly used to estimate vibration insulation.

Human vibration exposure is measured by acceleration. The measurement is made using 3D sensors or by determining alternately with a 1D sensor in different directions. The vector sum of the three dimensions is obtained by the Pythagorean theorem from the x, y, and z components of the effective values of acceleration

(11.3.1) $$\tilde{a}_v = \sqrt{\tilde{a}_{w,x}^2 + \tilde{a}_{w,y}^2 + \tilde{a}_{w,z}^2}$$

The effective value of oscillation acceleration is given by the equation vibration acceleration

(11.3.2) $$\tilde{a} = \sqrt{\frac{1}{T}\int_0^T a^2(t)dt}$$

Whole-body vibration refers to vibrations applied to the whole body. Vibrations are directed at a person either from the floor via feet or from the seat via buttocks and thighs. When assessing the effects of whole-body vibration, the effective value is weighted in spectral space, so that the result corresponds to the subjective experience of vibration (compare the A-weighting of the SPL):

(11.3.3) $$\tilde{a}_w = \sqrt{\sum_{n=1}^{N} (W_n \tilde{a}_n)^2}$$

where W_n denotes either the weighting factor W_d or W_k at n as shown in Fig. **11.3.1.** The constant n applies to the frequency bands to be summed.

Hand-arm vibration refers to vibrations applied exclusively to the hands. In the assessment of the effects of hand-arm vibration, emphasis is also placed on:

(11.3.4) $$\tilde{a}_{h,w} = \sqrt{\sum_{n=1}^{N} \left(K_n \tilde{a}_{h,n}\right)^2}$$

where K_n is the weighting factor used to estimate hand-arm vibration (**Fig. 11.3.1**). Hand-arm vibration exposure shall be assessed by determining the equivalent weighted acceleration for the duration of an 8-hour working day:

(11.3.5) $$\tilde{a}_{h,w,eq(8)} = \sqrt{\frac{1}{T_8}\int_0^{\tau} \tilde{a}_{h,w}^2(t)dt}$$

where is $\tilde{a}_{h,w}(t)$ the effective value of the weighted vibration as a function of time and T_8 is 8 hours. If the duration of exposure is not 8 hours per day but time T, estimate the vibration lasting 8 hours equivalent from the formula

(11.3.6) $$\tilde{a}_{h,w,eq(8)} = \tilde{a}_{h,w,eq(T)}\sqrt{\frac{T}{T_8}}$$

The detection threshold is shown in Fig. **11.3.2.**

Human-related vibration is usually measured within 0.1–80 Hz.

	W_k	W_d	K_n
0.5	0.418	0.853	
0.63	0.459	0.944	
0.8	0.477	0.992	
1	0.482	1.011	
1.25	0.484	1.008	
1.6	0.494	0.968	
2	0.531	0.890	
2.5	0.631	0.776	
3.15	0.804	0.642	
4	0.967	0.512	
5	1.039	0.409	
6.3	1.054	0.323	0.727
8	1.036	0.253	0.873
10	0.988	0.212	0.951
12.5	0.902	0.161	0.958
16	0.768	0.125	0.896
20	0.636	0.100	0.782
25	0.513	0.080	0.647
31.5	0.405	0.063	0.519
40	0.314	0.049	0.411
50	0.246	0.039	0.324
63	0.186	0.030	0.256
80	0.132	0.021	0.202
100			0.160
125			0.127
160			0.101
200			0.0799
250			0.0634
315			0.0503
400			0.0398
500			0.0314
630			0.0245
800			0.0186
1000			0.0135
1250			0.0089

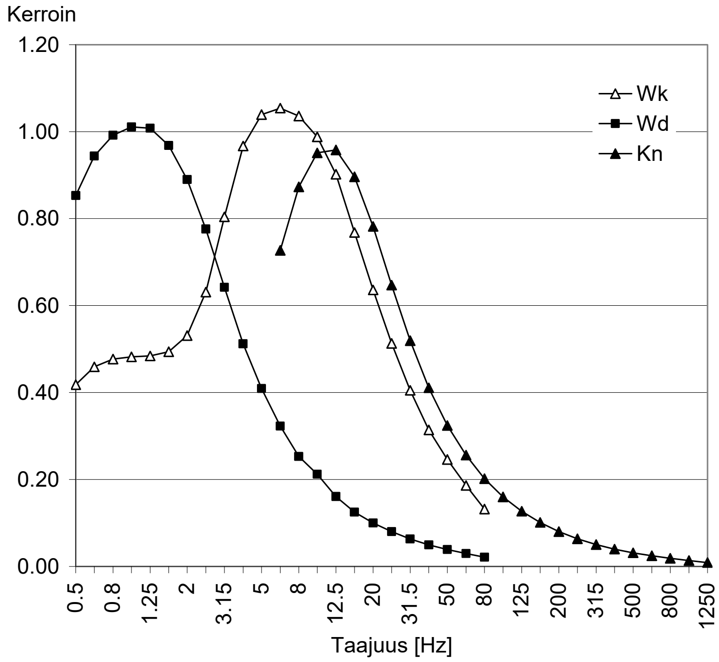

Fig. 11.3.1. Weighting terms for the estimation of whole-body and hand-arm vibration exposure according to ISO 2631 and ISO 5349. In standards, there are even more weighting factors. The weighting factor W_d is used in the x and *y* directions. The weighting factor W_k is used in the z-direction.

	a [mm/s^2]	v [mm/s]
5	15	0.48
10	18	0.29
20	20	0.16
40	25	0.10
80	28	0.06
160	40	0.04

Fig. 11.3.2. The average sensing threshold of a sedentary person for vertical oscillation based on several laboratory experiments (Schwendicke &; Altinsoy, 2020).

11.4. Occupational vibration exposure limit values

Hand-arm vibration causes health hazards especially to the circulatory, musculoskeletal system or nervous system. Hand-arm vibration is exposed when using various handheld machines. Whole-body vibration causes lower back diseases and spinal injuries in particular. Exposure to whole-body vibration occurs in vehicles and machinery cabs, for example.

The assessment of occupational vibration exposure is based on Government Decree 48-2005, which is consistent with the European Community "Vibration Directive" (2002/44/EC). The procedure is uniform across Europe.

Action and limit values for vibration exposure are laid down in the decree. If the action values are exceeded, the employer is obliged to take certain preventive measures, whereas the limit value is a so-called "limit value". an absolute upper limit, which must not be exceeded at all.

The hand-arm vibration exposure limit value is 5 m/s^2 and the action value is 2.5 m/s^2 in relation to the eight-hour reference period. The limit value for exposure to whole-body vibration is 1.15 m/s^2 in relation to the eight-hour reference period and the action value is 0.5 m/s^2, respectively. If exposure to vibration is very short-term, a limit value of 7 m/s^2 for whole-body vibration and 35 m/s^2 for hand-arm vibration shall be used.

The estimation of hand-arm vibration exposure is based on the calculation of the eight-hour daily exposure, i.e. the nominal 8-hour average of hand-transmitted vibration A(8). The daily exposure is expressed as the vector sum of the effective values of the frequency-weighted acceleration, i.e. the so-called frequency weighted acceleration. as a total value. The effective values *ahwx,* ahwy, and ahwz are determined in directions perpendicular to each other. The exposure shall be assessed as specified in ISO 5349-1 (2001), chapters 4 and 5 and Annex A. Exposure may be estimated on the basis of vibration emission data provided by the manufacturers of the work equipment used, observations of working times and working methods, or by measurement.

The estimation of whole-body vibration exposure is based on the calculation of the nominal average daily exposure over eight hours. Exposure may be estimated on the basis of vibration emission data provided by the manufacturers of the work equipment used, observations of working times and working methods, or by measurement.

In the case of whole-body vibration, longer measurement times are possible, especially if the person sits in the cab of the vehicle throughout the working day, but the measurement is basically the same.

Fig. 11.4.1 shows the hand-arm vibration measurement event and the hand-arm vibration limit value for different exposure times.

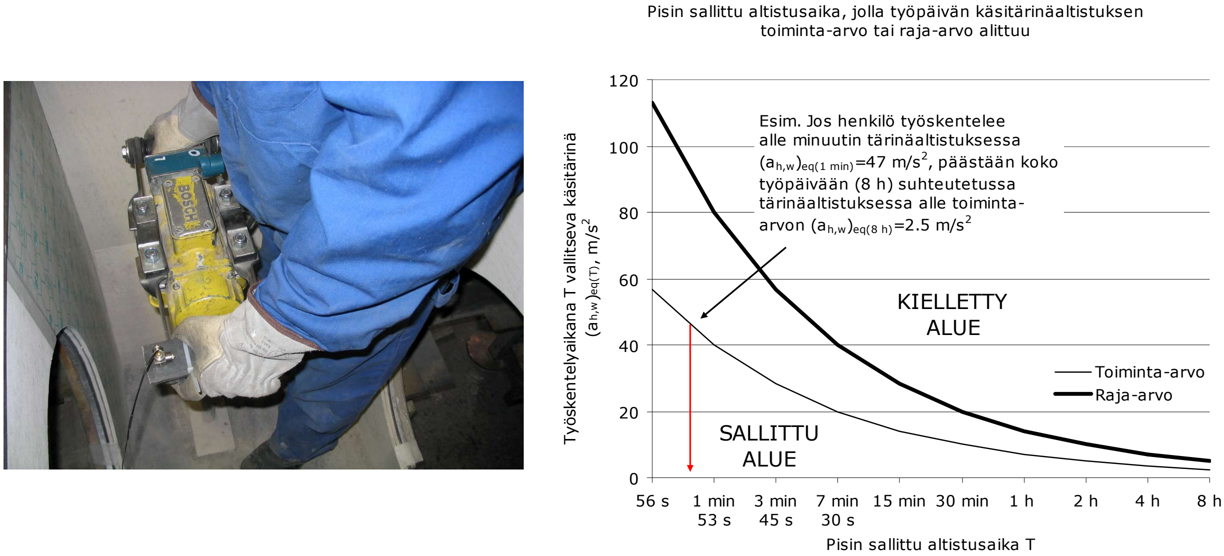

Fig. 11.4.1. Hand-arm vibration exposure is measured by inserting an iron between the hand and handle to which the 3D sensor is attached. On the right is the hand-arm vibration exposure action value calculated at different vibration exposure times.

11.5 Effects and assessment of road or railway vibration

Environmental vibrations detected in buildings are most often caused by rail or road traffic. Construction vibration is measured by vibration velocity.

There are no vibration limit values for buildings in Finland. In order to achieve the living comfort required by building regulations, the vibration intensity should be such that complaints about vibration are kept to a minimum. VTT has compiled foreign guideline values and they are generally applied in the design of roads and buildings (Törnqvist and Talja, 2006; Talja et al., 2008; Talja and Saarinen, 2009).

Vibration can impair living comfort and damage buildings. The disadvantages of vibration include:

- reduction in living comfort
- disturbed concentration and sleep
- fear of structural damage or a decrease in the value of the property.

As a rule, disadvantages associated with living comfort are manifested at significantly lower vibration rates than structural damage.

Traffic vibrations are mainly caused by irregularities in the road lane. When the tires hit irregularities, the vehicle's shock absorbers begin to resonate. The resonance is strongest in the 10–16 Hz range, where the strongest vibration amplitudes are usually observed. The intensity of vibration in a building strongly depends on the construction method of the building and the resonances of the structures themselves. The most difficult situation is when there are resonances under construction in the same frequency range where traffic vibrations are strongest. For example, the resonances of intermediate floors are located between 5 and 20 Hz, which is also where road traffic vibrations occur the most.

The lighter the building, the more sensitive it is to vibrate. The most difficult are unpiled buildings built on clay mattresses with light intermediate floors. In an apartment building, the amplitude of vibration decreases by 4–10 dB per floor, so the greatest disadvantages are usually experienced on the lowest floors. On the top floors, however, wobble is more easily detected, which can be harmful to structures.

Vibration control in road traffic includes:

- alignment of roads at a sufficient distance from houses, especially in clay-based areas,
- maintaining the evenness of the road surface,
- thicker than normal road surface,
- speed limitation,
- diverting heavy goods traffic onto roads with less vibration nuisance,
- strengthening the foundations of the building, or in extreme cases,
- "vibration insulation" of the building in the so-called. ground break.

When assessing traffic vibration, the vibration sensor is placed in the appropriate place:

- at the object of vibration complaint, to the floor of the room where the vibration nuisance is experienced;
- at the building permit stage, the location where the building will be located and/or a nearby building with similar traffic vibrations.

The oscillation velocity is measured in xyz directions. The upper limit frequency normally considered is 100 Hz. In each direction, the oscillation rate is considered separately, that is, the xyz resultant value is not determined. For the oscillation velocity, frequency weighting is first made using the equation

(11.5.1) $$W_v(f) = \frac{1}{\sqrt{1+\left(\frac{f_0}{f}\right)^2}}$$

where f_0=5.6 Hz, after which we obtain the weighted oscillation rate in the frequency band under consideration vW by the equation

(11.5.2) $$v_W = \sqrt{\sum_i \left(W_{v,i} v_i\right)^2}$$

where i comprises the number of frequency points (e.g., one-third octabe bands) in the band under consideration.

The oscillation is followed by a period during which a representative number of oscillation events is obtained. For example, daily monitoring is usually sufficient for road transport, while a one-week period may be required for rail transport. The 15 strongest vW values per second ($v_{W,1s}$) shall be selected for the period. From these, the mean value, M, and the standard deviation SD are calculated. As a result, the upper limit value of the 95 % percentile of frequency-weighted oscillation velocity maximums, $v_{W,95}$, is given by:

(11.5.3) $$v_{W,95} = M + 1.8 \cdot SD$$

In Finland, guidelines are applied to new buildings, according to which $v_{W,95} \leq 0.30$ mm/s (Ministry of the Environment, 2018). Under such circumstances, it is estimated that no more than 15% of the population would find vibration events disturbing.

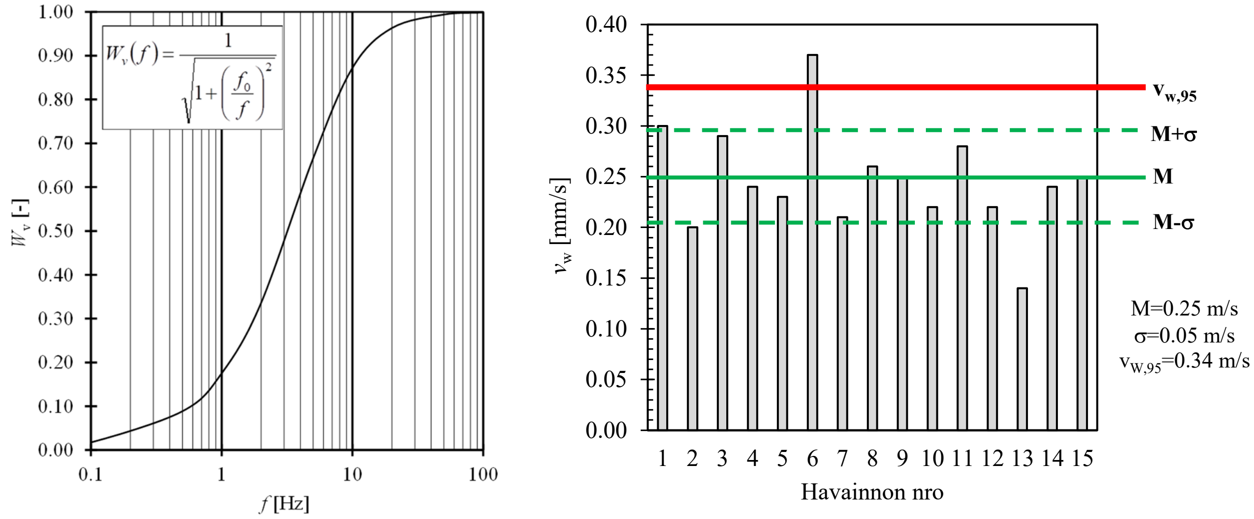

Fig. 11.5.1. On the left, frequency weighting according to Eq. 11.5.1. On the right, a statistical examination of the fifteen most powerful oscillatory events.

11.6 Vibration isolation

Fig. **11.6.1** shows a situation where a sinusoidal force F with angular frequency ω is exerted on the mass m. If the device is installed rigidly attached to the floor, the vibration of the device will be transmitted directly to the floor and may cause inconvenience in the near quarters. The purpose of vibration insulation is that the rigid installation is replaced by a flexible installation, thereby increasing the vibration of the device, but applying less forces to the floor.

Insulation is based on a spring-mass formation, where the device represents the mass and the insulator represents the spring. There are 4 forces acting in the system. The sum of the inertial force of the mass, the spring force and the damping force due to the speed of movement must be equal to the interference force of the rotating machine:

(11.6.1) $$m\frac{d^2x}{dt^2} + kx + C\frac{dx}{dt} = F(t)$$

where m is the mass [kg] of the whole system, k is the spring constant [N/m], and C is the damping of the spring [Ns/m].

The characteristic frequency f_0 of the system is determined by setting the interference force to zero, deviating the system from its sleep state, and allowing the system to vibrate freely. The system then resumes a damping oscillatory motion at a constant frequency f_0. Suppose there is no attenuation. Assuming that the oscillation is sinusoidal, $x = \hat{x}\sin\omega t$, we obtain

(11.6.2) $$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

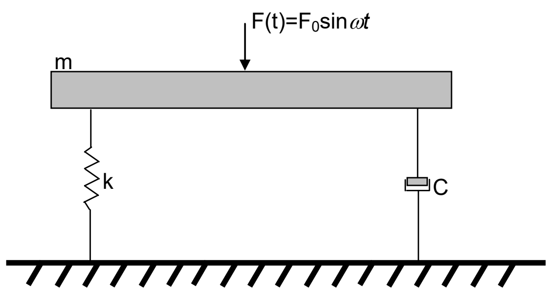

Fig. 11.6.1. A system consisting of a machine with a mass of m and its chassis structure. The machine exerts a force F on m and F' on the chassis.

For ordinary steel springs, there is a relationship between spring force and compression $\Delta\xi$ [m] due to mass

(11.6.3) $$k \cdot \Delta x = m \cdot g$$

that is, compression is obtained from the equation

(11.6.4) $$\Delta x = \frac{mg}{k}$$

So, the specific frequency is

(11.6.5) $$f_0 = \frac{1}{2\pi}\sqrt{\frac{g}{\Delta x}} \approx \frac{1}{2\sqrt{\Delta x}}$$

Thus, the specific frequency of the system can be visually estimated by observing the compression of the spring or insulator relative to the equilibrium position.

If the mass is affected by the perturbing force F and the force F' acting on the base via the springs, the energy transmission ratio is

(11.6.6) $$e = \left(\frac{F'}{F}\right)^2$$

Its inverse is level reduction:

(11.6.7) $$D = 20 \cdot \log_{10}\left(\frac{F}{F'}\right)$$

If there are no losses in the system, then the level reduction is given by the formula

(11.6.8) $$D = 20 \cdot \log_{10}\left(1 - n^2\right)$$

where n is the frequency ratio

(11.6.9) $$n = \frac{f}{f_0}$$

As a rule, there are losses in the springs or the springs have shock absorption, which is proportional to the speed of movement

(11.6.10) $$F = kx + C\frac{dx}{dt}$$

The damping ratio is defined as:

(11.6.11) $$\gamma = \frac{C}{C_c} = \frac{C}{2\sqrt{km}}$$

where C_c is the critical damping coefficient. Insulation in this case takes the form

(11.6.12) $$D = 10 \cdot \log_{10}\left[\frac{(2\gamma n)^2 + \left(1 - n^2\right)^2}{1 + (2\gamma n)^2}\right]$$

In practice, however, vibration insulation is closer to the curve **in Fig.** 11.6.2.

The damping ratio, or loss factor, can be determined from the insulation curve using the half-value width method

(11.6.13) $$\gamma = \frac{\Delta f}{f_0} = \frac{f_2 - f_1}{f_0}$$

where f_0 [Hz] is the resonant frequency at which the greatest negative isolation is achieved, and f_1 [Hz] and f_2 [Hz] are the frequencies where the insulation is 3 dB better than the frequency f_0 ($f_1 < f_2$).

The main features of vibration isolation are as follows:

- Find out the number of revolutions of the machine and calculate the operating frequency F, alternatively carry out a measurement and decide on the lowest frequency F, which you still want to isolate
- determine the reasonable number of insulators that can be installed under the unit;
- determine the mass and weight distribution of the system
- Consider each insulator and the mass applied to it: select the insulator so that the lowest frequency *F* to be insulated does not coincide with the specific frequency of the mass-spring system, but at least at 2F0, allowing an insulator of more than 10 dB to be achieved.

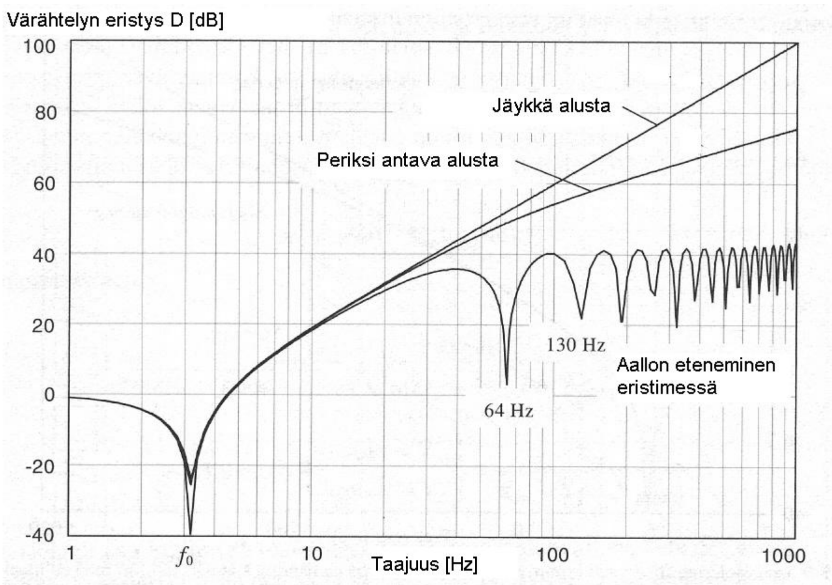

Fig. 11.6.2. In practice, vibration insulation rarely increases above 30 dB due to the presence of resonances in the insulator (Boden et al., 1999).

It has been assumed above that there is no movement in the base due to the excitation force, i.e. the spring constant of the foundation is infinitely large. In practice, the base can be considered sufficiently rigid if the mass of the foundation is ten times the mass of the object to be insulated and the foundation does not wobble.

If the oscillating machine is massive, it cannot be insulated with insulators. In practice, "insulation" is carried out by casting a concrete foundation under the machine that is at least ten times heavier and to which the machine is rigidly attached. The foundation is insulated from the rest of the building with an expansion joint.

If rigidly installed piping leaves the machine, the piping must be isolated from the frame according to the same principles as above, or a flexible connection must be installed between the piping and the machine.

According to equation **11.6.5**, the resonant frequency of an oscillating system can be estimated by compression alone. Fig. **11.6.3** shows an example of poorly and well-executed vibration insulation. With a thin insulator, the resonant frequency is high, due to which the sound of rotation of the vibrating machine is transmitted almost undamped to the frame of the building, causing a noise downstairs. When the compression of the insulator is increased, the resonant frequency of the system drops significantly below the rotational frequency of the machine, resulting in an insulating capacity of 20 dB.

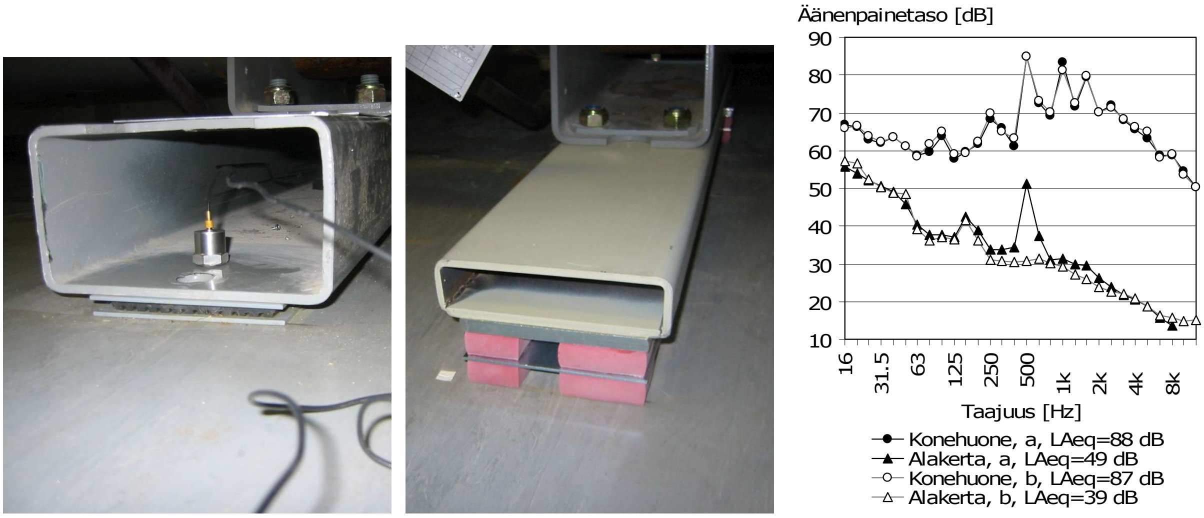

Fig. 11.6.3. The photographs show elastic vibration insulation between the frame and floor of the steel unit. (a) The wrong type of vibration isolator that does not compress sufficiently. (b) The correct type of vibration isolator with sufficient compression. (c) SPLs in the engine room and below office for each type of insulator. The type b insulator achieves the desired attenuation at 500 Hz.

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12. EFFECTS OF NOISE

12.1. Summary

Noise is unwanted, unnecessary or harmful to health.

The effects of noise can be divided into auditory or non-auditory. The auditory effects of noise, i.e. directly related to the hearing organs, include:

- noise-related hearing impairment,
- tinnitus and
- temporary hearing loss.

Non-auditory effects of noise include:

- Noise annoyance
- Difficulty communicating with speech
- Voice disorders, i.e. damage to speech organs
- Changes in nervous system and alertness
- Physiological effects
- Sleep effects
- Changes in work performance
- Behavioural changes (coping)

The perception of sound as noise depends on the environment and the requirements required for the current function. The effects of noise, both bodily and psychological, are individual and difficult to predict on an individual level. However, the average noise effects of the normal hearing population can be predicted with moderate accuracy.

Only the difficulty of speech communication can be predicted fairly well at the sound level L_{Aeq}. All other effects are better explained by other factors (so-called non-acoustic factors) than sound level.

Noise annoyance is the most common and frequently measured noise effect. Objectively measurable acoustic characteristics explaining annoyance are discussed in Chapter **13**.

12.2 Temporary hearing loss

In addition to or instead of tinnitus, high noise exposure can cause a temporary threshold shift (TTS). The greater the temporary hearing loss, the longer it takes to recover. If noise exposure continues before recovery, all or part of the hearing loss may become permanent. This is referred to as noise-induced permanent threshold shift (NIPTS). NIPTS is usually a precursor to noise-related hearing impairment if the hearing impairment is the result of years of noise exposure rather than, for example, a sudden impulse. Fig. **12.2.1** shows empirical data related to TTS. Note the logarithmic time axis.

Fig. **12.2.2** shows an example of the hearing threshold for a person with permanent noise-induced hearing loss within two working days. The hearing threshold does not have time to return to normal and the hearing loss is likely to continue to worsen.

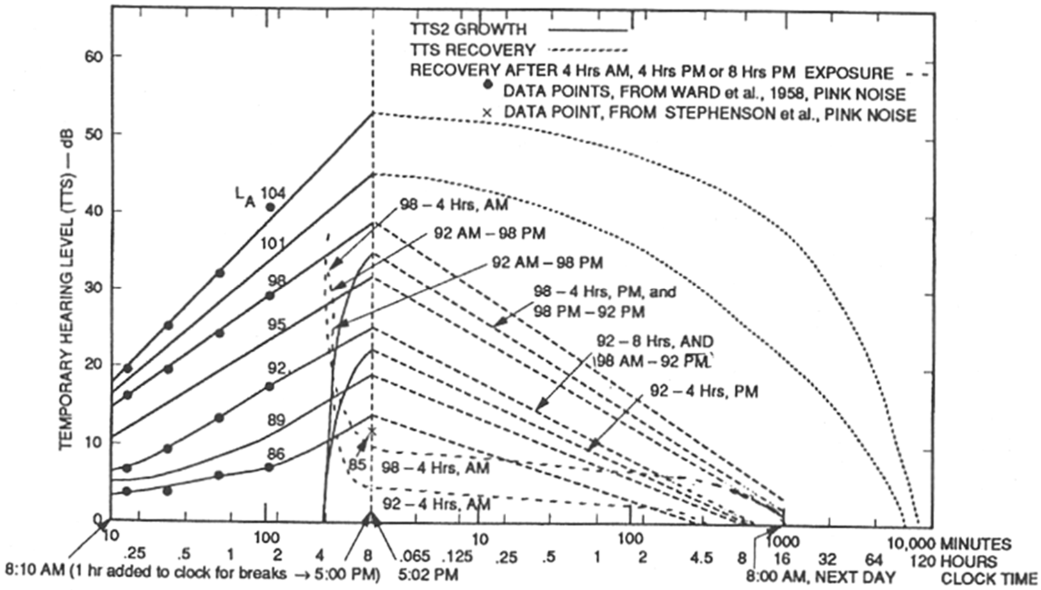

Fig. 12.2.1. Formation and recovery of temporary hearing loss after constant noise exposure throughout the working day without breaks. Y-axis TTS is defined at 4 kHz (Photo: Kryter, 1994).

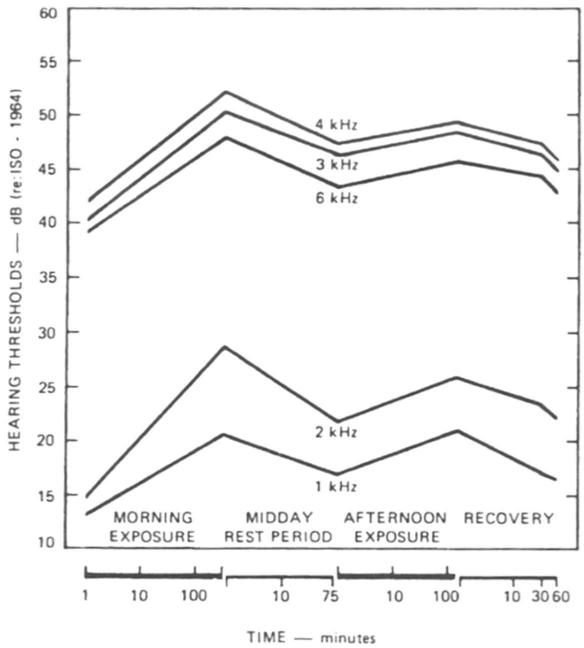

Fig. 12.2.2 Change in the hearing threshold of a person with permanent hearing loss measured at different times of the working day. A person has NIPTS because hearing loss is already significant in the morning (Photo: Kryter, 1985).

12.3 Tinnitus

Tinnitus means ringing in the ears for no external reason. Tinnitus is most often heard as a high-frequency sine sound, the SPL of which can correspond to an external sound level of 10–100 dB. There can be several frequencies of tinnitus at the same time. Usually the frequency is above 2000 Hz. Real sounds that occur at the frequency of tinnitus and have a level below the tinnitus level are difficult to hear because they are obscured by tinnitus.

It is estimated that there are approximately 700,000 people with hearing loss in Finland. Approximately 30–80% of people with hearing loss are estimated to have permanent tinnitus. Tinnitus is often associated with sensitive hearing. However, there is no clear link between tinnitus intensity and the degree of hearing impairment.

Tinnitus is common even in people with normal hearing after temporary heavy noise exposure. Usually, tinnitus disappears after a few minutes or hours after intense noise exposure has stopped, but if noise exposure continues, tinnitus can become permanent. It can become a chronic problem for life. Tinnitus is often accompanied by hyperacusis, or sensitive hearing.

One explanation for tinnitus may be that sensory cells have died and the nerve cells beneath them remain "idle".

Tinnitus is most often caused by ear-related diseases/injuries, but also by circulatory problems, muscle tension in the neck and shoulder area, or improper bite.

Tinnitus has three degrees of severity:

1. Mild tinnitus, where the ailments can be overcome without major problems.
2. Moderate tinnitus, in which case there are already disadvantages such as difficulty falling asleep.
3. Severe tinnitus, which disrupts daily life, causes a lot of difficulty concentrating and sleeping, and makes work and social life more difficult, with depression and can have serious consequences.

You can try to treat tinnitus with a masking voice, a hearing aid, or by coming up with activities that distract attention from tinnitus. In addition, individual adaptation training methods have been developed.

12.4 Noise-related hearing impairment

The sensation of hearing is caused by the vibration of the basilar membrane of the inner ear. The vibration moves the hair cells that act as auditory sensory cells. At birth, a person has about 16,000 sensory cells. Hearing is at its most sensitive around the age of 12. Every year, a small part of the hair cells self-destructive. This so-called. Programmed cell death destroys an estimated 10–100 cells per year during adolescence, while the pace accelerates and hearing deteriorates with age. Hair cells do not regenerate. When exposed to a rifle shot without protective equipment, for example, hair cells are permanently destroyed and the sense of hearing is based on the remaining hair cells. That's why it's important to start hearing protection at a young age.

Individual hearing threshold refers to the lowest volume a person can hear. The hearing threshold is determined by audiometric examination. The hearing test takes place in a quiet environment so that the quietest test sounds can be heard.

At any age, the hearing threshold depends on the frequency. In the case of an audiological examination, the hearing loss of a healthy 18-year-old person is recorded as 0 dB and acts as a so-called "hearing loss". as a reference in hearing tests. A person whose hearing loss in both ears is less than 20 dB is considered normal hearing.

Hearing tests are performed at workplaces at two different levels of accuracy. An accurate hearing test must be performed by the company's occupational health care before noise work begins. The study covers frequencies 250, 500, 1000, 2000, 3000, 4000, 6000, and 8000 Hz. Periodic checks are carried out at the beginning of noise exposure once a year for four years and thereafter once every three years (Savolainen, 2006). In the beginning, the purpose of frequent inspections is to find the most noise-sensitive people. The periodic examination is a screening type, which is a slightly less accurate method and only occurs at frequencies of 500–8000 Hz. In this case, the hearing threshold is not tested at levels below 20 dB, because hearing at level 20 dB is still classified as normal hearing ability. Based on the hearing screening, hearing class I–IV is chosen, which determines the degree of disadvantage in working conditions.

In the case of age-related hearing loss, the hearing threshold increases with increasing frequency (Fig. 12.4.1), so it can be distinguished from noise-related hearing loss (Fig. 12.4.2) when the damage is sufficiently advanced.

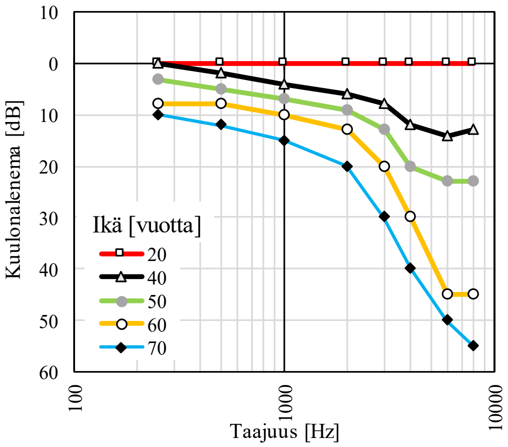

Fig. 12.4.1. Frequency dependence of hearing loss (presbycusis*) typical of age-related hearing loss* (Photo: Toivanen, 1976).

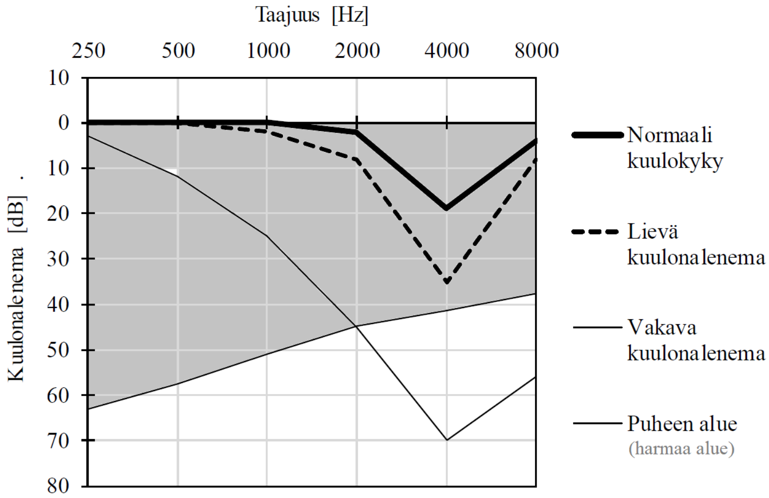

Fig. 12.4.2. Noise-related hearing loss (*sociocusis*). The bottom edge of the grey area represents the SPL of speech at a distance of 1 m from the speaker. In the case of severe hearing loss, some of the frequencies contained in speech are cut off because the hearing threshold level is higher than the speech level.

Permanent hearing loss, or hearing loss, can be caused by exposure to loud noise, ageing, unfavourable genetics, congenital hearing loss, exposure to certain chemical substances, high blood pressure or other individual factors. Of all hearing impairments, it is estimated

- 50% can be explained by hereditary factors such as age-related hearing loss or hereditary hearing loss,
- *20 %* is explained by noise exposure,
- 30% can be explained by other risk factors, the most common of which are high cholesterol, hypertension, smoking, painkillers, circulatory problems or co-exposure to toxic substances or vibration to the inner ear with noise exposure.

Noise-related hearing loss can occur from prolonged exposure to noise, slowly, or from a sudden strong impulse immediately, such as a shot or explosion. According to ISO 1999 (1990), the probability of hearing impairment for fairly steady noise can be predicted based on exposure time and noise exposure (Fig. **12.4.3**). However, there are large individual differences in the susceptibility of noise-related hearing impairment to formation. Individuals with the lowest noise tolerance experience hearing impairment even with very low noise exposure, whereas those who tolerate a lot of noise exposure may not experience hearing loss even with high noise exposure. Therefore, hearing tests focus on identifying the most noise-sensitive individuals through screening tests.

Noise-related hearing loss always causes the strongest hearing loss first at frequencies 3□6 kHz (Fig. **12.4.4**), regardless of the frequency content of the noise. As noise exposure continues, hearing loss increases and gradually extends to low frequencies. When the hearing loss is large enough, speech communication becomes more difficult because some speech frequencies remain below the hearing threshold.

Year after year, noise-related hearing impairment (noise impairment) is the single most common occupational disease (Fig. **12.4.3**), although the proportion of noise workers may have decreased over the years and hearing protection is receiving increasing attention. However, the degree of harm caused by hearing impairment has decreased, probably due to increased noise abatement in the workplace.

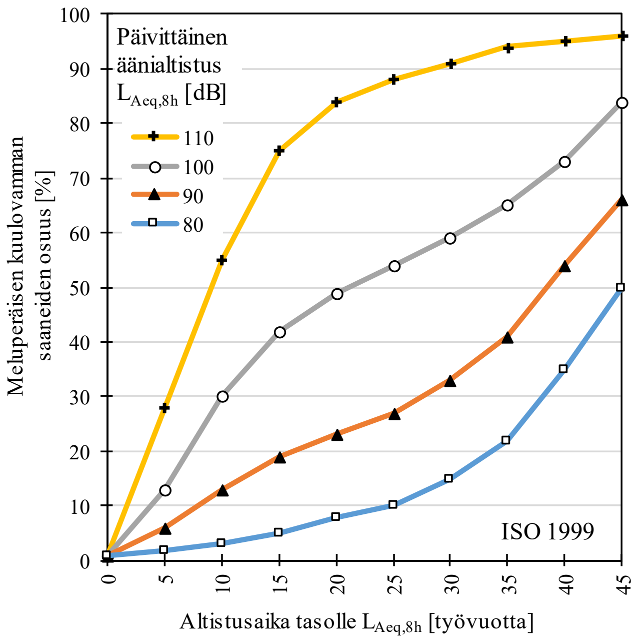

Fig. 12.4.3. Proportion of hearing impaired and noise exposure (ISO, 1999).

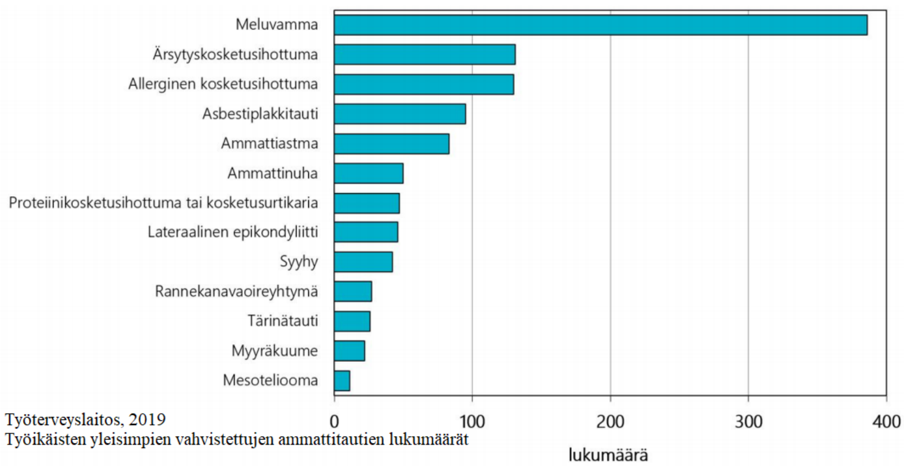

Fig. 12.4.4. Confirmed most common occupational diseases in 2015 (Finnish Institute of Occupational Health, 2019). Only 10 of the noise injuries were found in women, the remaining 370 in men.

12.5. Noise annoyance

From the point of view of psychological well-being, noise can be defined as sound that interferes with a task in progress, is unnecessarily loud or otherwise contradicts a perceived or desired soundscape. Distraction is the feeling of discomfort caused by a particular sound when it is known or believed that a sound negatively affects an individual or group.

Distraction is not the only feeling that noise causes. Other emotions caused by noise include anger, disappointment, dissatisfaction, withdrawal, helplessness, depression, restlessness, distraction, arousal or exhaustion. However, the term annoyance is commonly used to link these factors.

The annoyance of noise is measured with a questionnaire, usually using a 5-step verbal scale or an 11-step numerical scale. Chapter **13** discusses this in more detail.

Classifying a sound as noise or pleasant sound does not happen by evaluating the volume alone, because the perception of sound is always subjective:

- A loud rock concert can be the most terrible noise for one person, while for another it is a fulfillment of expectations, both musically and socially. Explanations for different reactions can include differences in people's values and attitudes towards the sound itself and/or its producer.
- The voice of colleagues in the office is not perceived as noise when participating in the discussion. The same conversation can be just noise for an outsider, especially if they are simultaneously trying to focus on a demanding job or another conversation.
- In the factory, the operator of the production machine 1 receives feedback on the success of the work and the functionality of the machine from the various nuances of the machine's sound, and the sound does not disturb the operator. The user also knows how to protect their hearing at the right moment. On the other hand, an employee of the quieter production machine 2 next door perceives the sound of production machine 1 as noise, because it requires him to wear hearing protectors as well, and the sound masks the sound of production machine 2, reducing the quality of his own work. A business owner walking by, on the other hand, does not perceive either sound as noise, but as a sign that the work is progressing on schedule (machine sound = cash flow sound).

Fig. **12.5.1** shows a model of what factors influence the experience of a sound or soundscape. The boxes in the picture are illustrated below it.

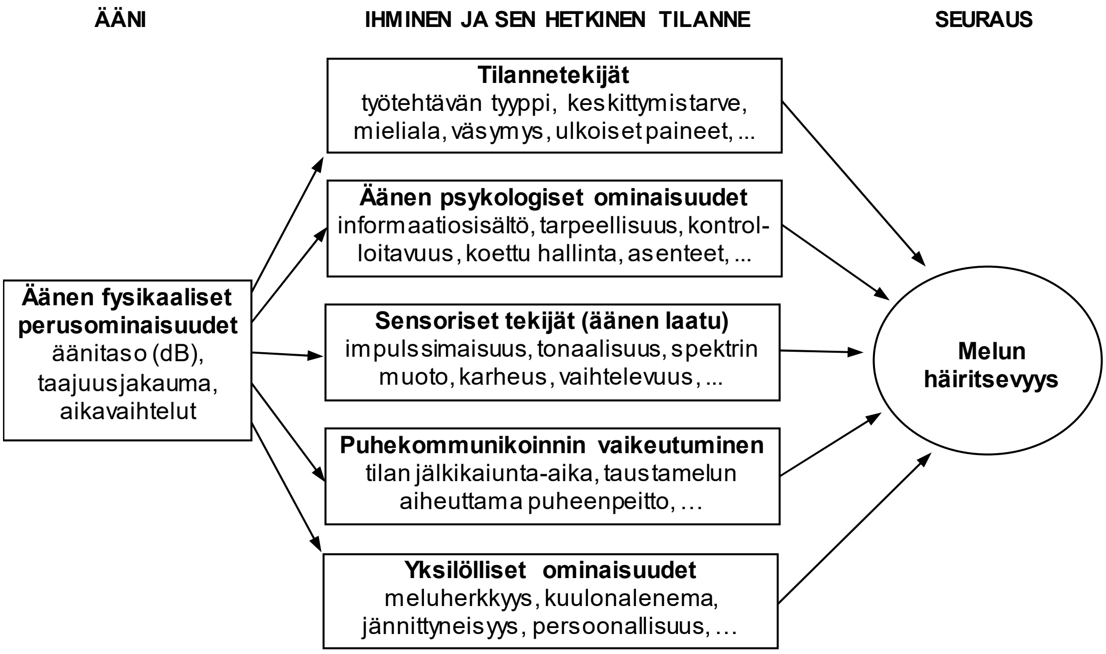

Fig. 12.5.1. The decision whether noise is annoying is made by a person, not a noise meter. The physical properties of noise explain only part of the experience.

Situational factors. A person's work task or activity sets requirements for an appropriate sound environment. When sleeping, a low sound level is required, while at a rock concert the opposite is true. Tasks that require concentration require a smooth soundscape, while routine tasks run better if something happens in the environment. If the task or situation is under time pressure, the requirements for an appropriate sound environment are high. Expectations can also vary depending on fatigue and mood.

Psychological characteristics of sound. In offices, most disturbances are caused by voices that are unnecessary to oneself. From time to time, people talk or listen to themselves, so that the speech heard in the space is no longer noise. Sound is often classified as noise if its information is not useful. For example, the reversing sound of a truck is considered useful when being outside near the vehicle, but at the same time very disturbing to the person sleeping in a nearby bedroom, even though its level is more than 50 dB lower. Unexpected sounds are perceived as disturbing because they take up all the attention for a while. In this case, stress reactions are common, since a person can tense up ahead of time due to sound. If the noise source can be controlled, annoyance is usually reduced. For example, knowing that noise is attenuated by pointing it out to the person who causes it usually means that the noise is less disturbing than if the noise cannot be attenuated by mentioning this to the source of the noise. A negative attitude towards the sound producer usually translates into greater distraction of the sound. It is easier to accept a voice if the reason for the sound is known and it is generally or for your benefit. The noise of a neighbour with an unpleasant personality can be more harmful than that of a nice neighbour. A person living on the first floor may be particularly disturbed by the sound of the elevator because he does not use it himself. The sound of a wind power area does not usually disturb the residents of the area who own a share of the power plants. People who know that noise can be suppressed but not done for some reason find noise more disturbing than those who know or think that everything possible has already been done. For example, there are usually no complaints about noise inside passenger cars, because it is known that it is impossible to develop silent cars.

Sensory factors (sound quality). The sound may have peculiarities that add distraction. This is discussed in Chapter **13**.

Difficulty communicating with speech. The voice becomes disturbing if communication becomes more difficult, i.e. the level of one's own voice has to be raised to an unnecessarily loud level in relation to the situation in order to get a message to another person or the speech of the other person cannot be understood.

Individual characteristics. The effects of sound conditions may depend on factors such as noise sensitivity, personality traits, hearing ability and previous experiences.

- *Noise sensitivity:* Noise sensitivity predicts higher noise annoyance and is often an indicator of sensitivity to stressors in general. Noise sensitivity is measured by surveys. Gender or age do not predict noise sensitivity. Noise sensitivity is linked to some personality traits.
- *Personality traits:* There is some consensus among psychology researchers that the basic characteristics of the psyche can be described using five basic factors (Big 5). These are neuroticism (emotional balance), extraversion/introversion (extroversion or introversion), amiability (positivity/negativity of interaction), conscientiousness (ability to focus on the essentials and self-control) and openness (intellectual curiosity, openness to new experiences). Two of these have been linked to noise experiences. Neurotic personality implies lower stress tolerance and anxiety, which predicts a certain sensitivity to noise as well. It is particularly difficult for neurotic and painful people to perform memorization and learning tasks in noise (Nurmi and von Wright, 1983; von Wright and Vauras, 1980). Noise is an environmental stressor that affects alertness. Extroverts have a better ability than introverts to regulate their alertness, which makes it easier to cope with noise. Extroverted individuals are more dependent on signals from outside than introverts (Eysenck, 1967). Extroverted individuals find it more difficult to maintain their activation levels high enough without external stimulation than introverts (Green, 1984). For example, a car radio helps extroverts stay awake on long car journeys, whereas for introverts radio had no effect (Fagerström and Lisper, 1977). Introverts have been found to perform at their best at lower sound levels than extroverts. Extroverts often tolerate noise better than introverts. Some researchers associate temperament with the endocrine system, while others associate temperament with the characteristics of the nervous system. Personality is a characteristic of the brain that is characterized by a certain way of reacting and is reflected in bodily functions. With age, reaction patterns are shaped by experience and feedback from the environment. Personality is at least partly hereditary, but personality can change with age. Personality factors explain some of the differences in individual experiences of distraction.
- *Hearing loss:* The single largest group complaining about noise being annoying are people with hearing loss, who make up 10% of people. People with hearing loss, for example, are more likely to complain about sound conditions in offices because it is more difficult for them to assess the directions in which sound is input. In addition, hearing aids work poorly in noisy environments. The complaints are caused by ear sensitivity to sounds, distortion associated with hearing impairment, negative attitude towards noise, especially if one is aware of one's own hearing impairment, and poor speech intelligibility in background noise. People with moderate to severe hearing loss have a harder time understanding speech because the hearing curve tends to have a strong bump in the 4 kHz range. Consonants in this case stand out poorly. Directional hearing is worse than that of people with normal hearing, especially if the hearing loss is not the same in both ears. Speech is less understandable in a noisy state. Hearing loss increases with age.
- *Cultural factors:* Some cultures accept noise better than others. For example, densely populated areas are used to higher noise and can therefore be better tolerated in sparsely populated quiet areas.

12.6 Difficulty in voice communication

Speech communication becomes more difficult as background noise levels increase, reverberation time lengthens and speech noise levels decrease. Difficulties in speech communication cause avoidance of communication, learning difficulties, misunderstandings, accident risks, decreased work capacity, feeling isolated, deterioration of social interaction and relationships, and various stress reactions. Speech expression has to be condensed in strong background noise, which highlights the importance of non-verbal communication, such as gestures, emotions, attitudes, workplace culture and experience, in interaction and increases the possibility of misunderstandings.

Speech communication in background noise can be facilitated by increasing the volume or moving closer to the speaker/listener (Fig. **12.6.1**). In a quiet environment, the normal speech level is 50 dB L_{Aeq} when 1 m away from the speaker directly in front. Skilled people can speak 30 dB louder than this for long periods of time without loss of voice.

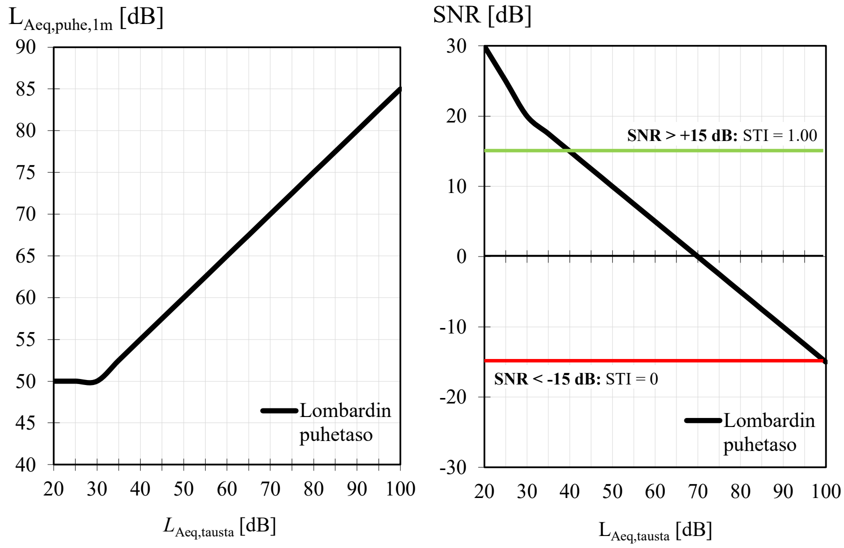

Fig. 12.6.1. On the left, the A-weighted SPL, L_{Aeq}, of speech is shown, speech (1 m away from the speaker in the free field) with different background noise A sound levels LAeq, background. On the right is the corresponding speech-to-noise ratio SNR ($L_{Aeq,speech}$–$L_{Aeq,background}$) when the speaker is 1 m away directly in front. Speech communication becomes impossible at this distance when the background noise level rises above 100 decibels because the speech-to-noise ratio drops below -15 dB. By moving 10–20 cm from the mouth of the shouting speaker, the sound level increases by up to 20 dB. This is often the case at rock concerts and dance restaurants.

12.7 Voice disturbances, i.e. damage to speech organs

Constant communication at elevated speech volume can cause voice disturbances and laryngeal diseases (Sala, 1995). The symptoms caused by a voice disorder include the following: the voice becomes tired or strained, the voice becomes hoarse or lower, the voice fails, breaks, disappears, the voice is difficult to hear, or singing is difficult. Throat symptoms include mucus, need to rumble, burning sensation, roughness, burning and pain. Voice disorders are most often organic in nature (e.g. vocal cord nodule), i.e. there is some structural defect in the vocal cords that becomes apparent when using the voice. In this case, the mucous membrane of the vocal cords no longer vibrates quickly and regularly with a little effort. In functional voice disorders, the vocal cords are healthy, but there are shortcomings in the regulation of the functioning of the vocal system. In neurological disorders, the vocal cords also look normal, but the vocal cords are deficient due to neurological reasons. Distinguishing between functional and neurological sound disorders is often a headache in clinical work. Voice disorders of predominantly psychological origin also occur.

Voice disorders are common in speech professions (teachers, singers, kindergarten staff), but can also occur in industrial environments where constant speech communication is required. Rontal et al. et al. (1979) found physiological changes in the larynx in 8% of those working in industry.

Work-related laryngeal diseases can include vocal cord nodules and polyps, laryngitis and chronic ulceration, and granulation. Vocal cord nodules arise as a result of prolonged speaking in loud voices. In voice disorders and diseases of the larynx, the function of the larynx as a source of sound is impaired. The sound cannot be heard enough, and the sound does not last long enough. The voice is poor and hoarse, and speech intelligibility deteriorates, especially in difficult communication environments. In this case, the speaker often consciously or unconsciously tries to change their working methods to make the use of voice less burdensome and more demanding of speech communication. If these measures are not enough to restore the balance between demands and performance, job performance will start to suffer. In addition to poor sound conditions, sound disturbances can also be caused by sensitisers, chemicals, dry room air and other air pollutants. Incorrect speaking can also be the cause of voice disorders, in which case rehabilitation of voice use helps.

The most common treatment for voice disorders is sick leave and rehabilitation. From the operator's point of view, reducing the risk of sound disturbance is an important justification for the acoustic design of teaching and care facilities, since sick leave due to sound disturbances is usually long and entails high costs.

Abnormal voice quality can attract the listener's attention. The listener begins to listen to the voice itself and not the message. At worst, a deviant voice causes the speaker to be underestimated and even discriminated against.

The adverse effects of a voice disorder depend on how demanding the use of speech is at work or during free time. A change in the vocal cord may render a teacher completely incapacitated, whereas a construction worker with a similar change in his vocal cord, for example, may not notice any harm to work because the need to speak is not constant.

In occupations where the work requires the use of a heavy or amplified voice, vocal cord nodules can be considered to meet the criteria for an occupational disease. However, the replacement of voice disorders as occupational diseases has not been established.

Sound ergonomics refers to all those measures that improve the possibilities of good sound production and speaking, hearing and distinguishing speech, i.e. speech communication (Sala, 2004).

12.8 Changes in nervous system and alertness

The information contained in physical stimuli, such as sound, can trigger many functions of the nervous system, which in turn can affect physiological and mental functions and behavior. The overall chain of action of neurological origin is as follows:

Sound stimulus *→ Sensory nerve → pathway → central nervous system → neural pathway → □response organ response →* ***Action***

Depending on the characteristics of the voice and the individual, the length of the chain of influence varies from a simple reflex arc to complex information processing involving several different areas of the brain.

The startle reflex, orientation and defensive reactions are instinctive reactions caused by momentary sudden information. In humans, these instinctive reactions are regulated or restrained by a remarkable adaptability based on learning and will. The reaction depends on its intensity and significance. The response controlled by the central nervous system is transmitted to the body via stress hormones from the autonomic nervous system and the HPA axis. The body's response can be broad or narrow, e.g., panic response or muscle twitching. Most often, the response is just either the initiation of some glandular activity or just processes occurring in the brain.

Only part of the sensations produced by sound stimuli reach the level of consciousness and are perceived. When perception is still associated with previous experiences, the information contained in the stimulus is understood in one way or another. The response to a stimulus can be mediated from different levels of consciousness. Which stimuli are perceived depends on external and internal factors. External factors include the intensity and qualitative characteristics of the stimulus. They affect whether attention is directed to sound. Other environmental conditions, i.e. competing stimuli (sight, touch, lighting, other sounds), also affect sound perception.

It depends on the adaptation of the sensory organ what kind of sensation the stimulus produces. A person's alertness and other emotional state, previous experiences of stimuli and state of need are also internal factors that affect the significance of sensation in the nervous system and the response caused by the stimulus.

In the stochastic dominance model, all stimuli enter the central nervous system. Relevant stimuli are perceived and others are constantly competing for attention. The total capacity of such a system is divided as a function of alertness into environmental monitoring, work and concentration, so that the capacity remaining at work has its optimal value at a certain state of alertness. In the model, the performance of the observation system follows the so-called "performance of the observation system" (Fig**. 12.8.1**).

As a rule, extroverted people have lower alertness than introverts, which means that performance can improve when alertness is increased, for example, by background noise. As a rule, introverts have a higher level of alertness, which means that background noise leads to a decrease in performance. Because of this, different personalities can experience the same sound environment in different ways while working or otherwise.

Attentiveness, i.e.,, the way of selecting information from the environment, depends on alertness. Alertness is lowest at night, low in the early morning and high in late afternoon. Alertness can also be adjusted by yourself. Noise increases alertness, but according to the Yerkes-Dodson model, endless increases in alertness lead to a decrease in performance.

Different tasks require different alertness levels in order to perform well. In an excessively high state of alertness, a person is at his most attentive and thus sensitive to processing all external signals, and the ability to concentrate on the main task begins to decrease. In hyperactive mode, you can no longer work at all. With optimal alertness, you can select the most important information to process, and performance is at its best. The more difficult and varied the task, the lower the alertness should be in order to maintain a sufficiently broad and selective level of attention. Since a person automatically focuses on signals coming through the senses, especially variable sound takes up part of the brain power and attention changes.

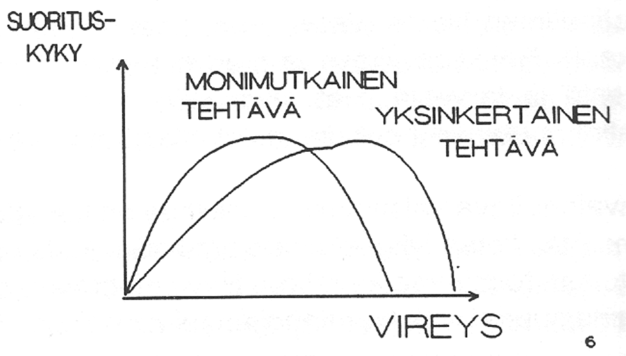

Fig. 12.8.1 According to the Yerkes-Dodson hypothesis, alertness increases e.g. when the sound level increases. Performance is at its best when alertness is appropriate. Simple tasks can be handled well even in noise, but complex tasks require a lower noise level and alertness.

12.9. Physiological effects

The effects of sound on physiological functions are transmitted through the auditory system. The effect is based on the numerous connections between the auditory pathway of the central nervous system and the motor, cognitive and autonomic centers that control vital functions. Sound perceived as noise triggers a vigilance reaction in the reticular activation system that affects all functions of the central nervous system (Heinonen-Guzejev et al., 2012). Signal modulation in the central nervous system can amplify or attenuate the effects of sound on physiological functions, depending on the importance given to sound by the central nervous system.

Disturbing noise, i.e. noise that the central nervous system gives a negative meaning to, can affect the functioning of the autonomic nervous system and endocrine glands. Noise can have both short-term and long-term effects on the cardiovascular system. The effect of noise-induced stress can be seen mainly through two physiological mechanisms. Sudden noise activates

- the sympathetic part of the autonomic nervous system, and
- the adrenal nucleus, or sympathicoadreno-medullary axis (SAM axis).

As noise persists, the hypothalamic-pituitary-adrenal cortical axis (HPA axis) is activated (Fig. **12.9.1**). As a result, physiological changes occur:

- acceleration of the pulse,
- vasoconstriction of the skin and internal organs,
- increase in blood pressure,

- increase in stress hormone levels in the blood,
- effects on lipid metabolism and blood clotting factors.

The activation of stress-mechanisms caused by noise increases blood pressure, heart rate, susceptibility to arrhythmias through activation of the autonomic nervous system and stress hormones, which can contribute to the risks of cardiovascular diseases. Disturbing sound can lead to disturbed sleep, anxiety, functional symptoms and similar health effects. Therefore, prolonged noise exposure can increase the risk of cardiovascular diseases such as hypertension, coronary heart disease and myocardial infarction.

The increase in stress hormone levels (cortisol, adrenaline, noradrenaline) due to increased noise-related alertness can be measured in blood plasma and saliva. Long-term stress can also be analyzed from the hair follicle. In a recent Finnish study, it was possible to observe an increase in stress hormone levels in noise of 65 dB after only 45 minutes of exposure (Fig.).

Noise exposure can originate, for example, from night-time traffic noise or workplace noise exposure. Daytime traffic noise levels of more than 60 dB (outside the façade of the apartment) are associated with an increased risk of coronary heart disease and myocardial infarction. Long-term workplace noise above 85 dB (ear canal level) has been found to be associated with higher myocardial infarction mortality. Higher individual noise sensitivity increases these risks.

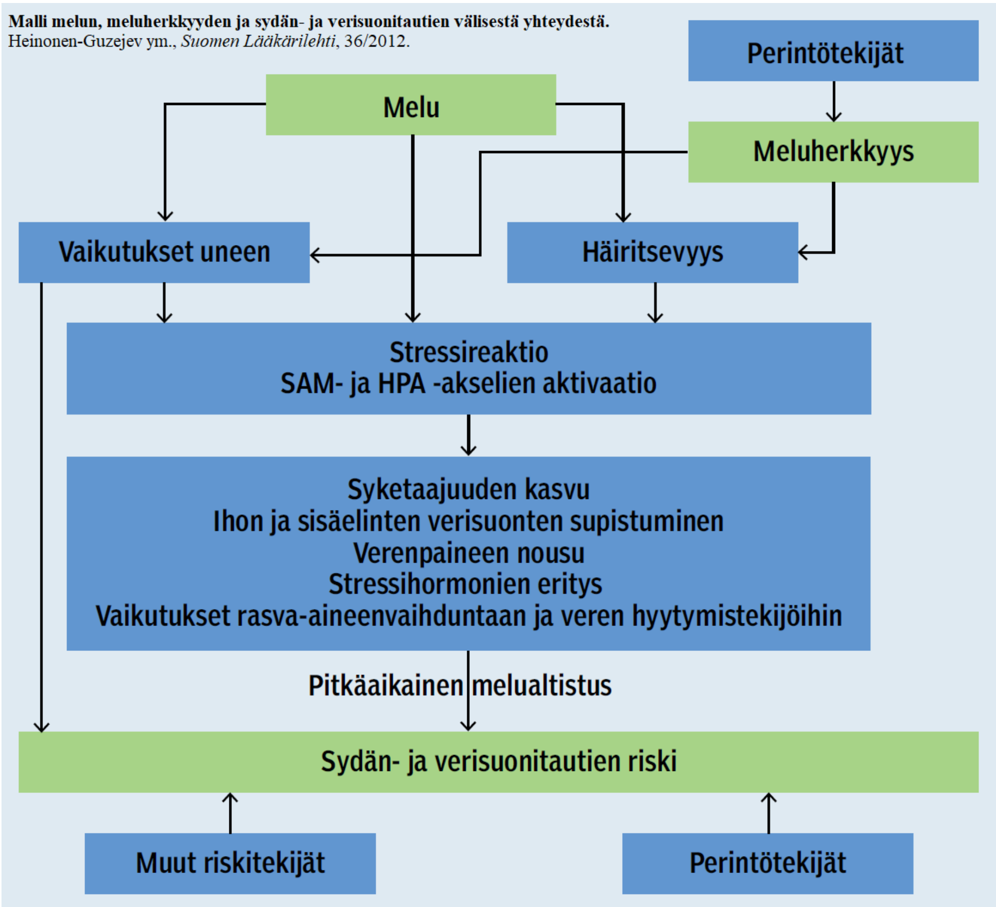

Fig. 12.9.1. Model of the link between noise, noise sensitivity and cardiovascular disease.

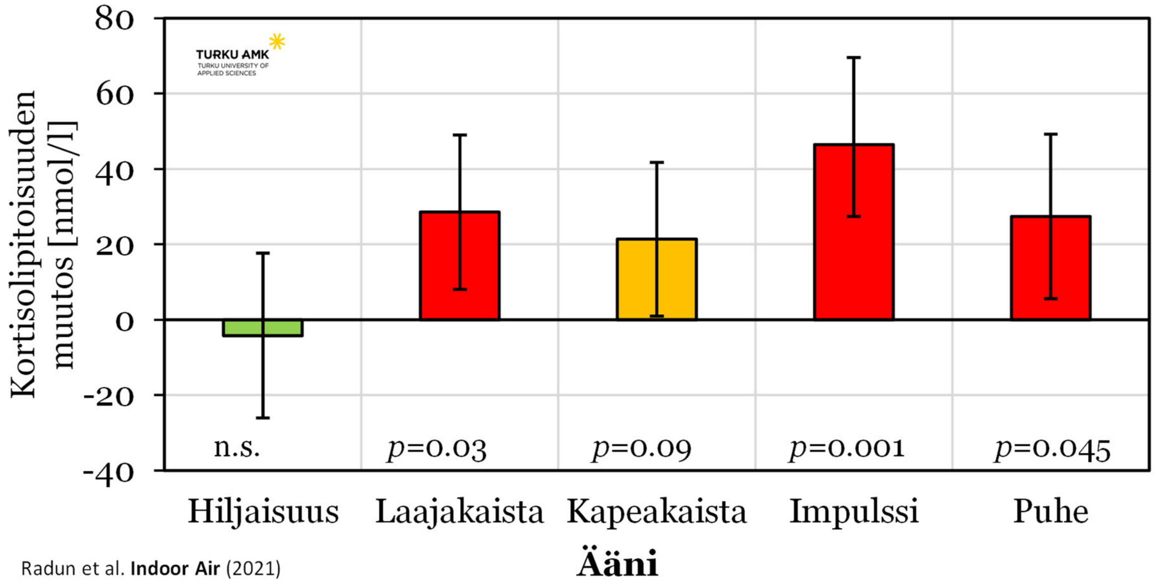

Fig. 12.9.2. Radun et al. (2020) found that exposure to noise for as little as 45 minutes significantly increased ($p<0.05$) blood plasma cortisol levels compared to the silent period following the experiment. When exposed to silence for 45 minutes, no effect was found. Silence: 35 dB L_{Aeq} ventilation sound. Broadband: 65 dB L_{Aeq} ventilation sound. Narrowband: 65 dB L_{Aeq} drilling machine. Impulse: 65 dB L_{Aeq} sound of piling rammer. Speech: 65 dB L_{Aeq} talk program from radio.

12.10 Sleep effects

Sleep can be divided into the following categories:

- Valve
- sleep (Non-REM sleep) or
- flash sleep (REM sleep, the phase of dreaming).

For well-being, it is important that the number and duration of these conditions during the day is suitable for the person. Long-term deviation from this balance is harmful to health.

It usually takes 5–20 minutes to fall asleep. After that, the person falls into deep (N3–N4) sleep through light sleep phases (N1–N2). The first phase of REM sleep usually begins in healthy adults about 90 minutes after falling asleep. In particular, the REM sleep phase has a revitalizing effect on brain activity. REM sleep is linked to the processing of learned information, memory functions and mental health regulation. After intense physical exertion, the relative amount of deep sleep increases.

The most important for recovery are deep sleep (N3–N4) and REM sleep, the former for physical recovery and the latter for mental and emotional as well as learning.

Noise exposure during sleep causes activation of the nervous system and endocrine systems (endocrine system). Variable noise causes intermittent sleep, while steady noise makes sleep superficial. In both, deep sleep (N3 and N4) and REM sleep are reduced. Falling asleep after each waking period becomes more difficult. When waking up, the mood can change and we are often tired during the day and suffer from headaches. Fig. **12.10.1** shows an example of the depth structures of good quality night's sleep.

In particular, noise reduces the duration of deep sleep. A Finnish study found an average 10% reduction in deep sleep N3 in young healthy adults (Fig. **12.10.2**) when exposed to normal nighttime road traffic noise (Myllyntausta et al., 2020; Varjo et al., 2015). This happened even though the subjects had been selected for the experiment on the basis that there was road traffic noise in their living environment. The level of road traffic noise studied was such (37 dB L_{Aeq}) that is commonly observed in apartments in city centres.

The average sound level alone does not indicate the intensity of the effect of noise on sleep, but the information content of noise and the quality of sound (variability, familiarity) also affect. The sense of hearing can put the body on alert and cause a person to wake up even at very low sound levels. For example, a mother who has just given birth may wake up to even the slightest sound or movement of the baby.

It should be noted that fluctuations in sound levels reduce sleep quality more than just a high sound level. For example, an adult can easily catch up with sleep even in loud noise (over 75 dB L_{Aeq}) if the noise is steady and predictable. Quality sleep can continue for a long time. This happens, for example, in a car if the sleeping position is otherwise comfortable. Waking up usually occurs even if the sound level changes.

In Finnish living environments, the effects of noise on sleep have been investigated, for example, in a survey conducted in 28 apartment buildings in different environmental noise areas. According to the responses, noise produced by neighbours and road traffic causes difficulty falling asleep and waking up only occasionally on average. The degree of harm was by no means alarmingly high (Hongisto *et al.*, 2013). The sleep nuisance of noise is likely to be greater in large cities in warmer countries, where the insulation of façade structures is naturally smaller due to the low need for thermal insulation and, at the same time, the noise levels in the environment are higher due to the higher volume of traffic.

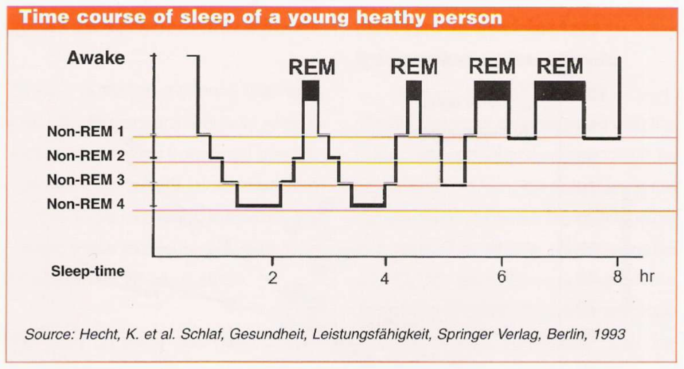

Fig. 12.10.1. An example of an 8-hour night's sleep structure with no noise disturbances.

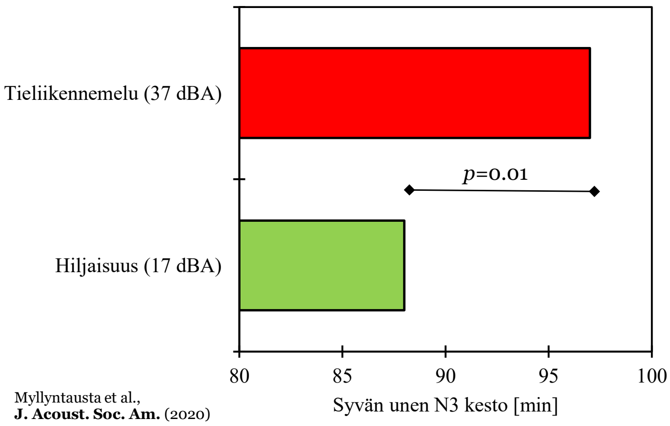

Fig. 12.10.2. In a Finnish laboratory study, the duration of deep sleep was reduced by 10% due to noise.

12.11 Changes in work performance

Work performance can be measured in laboratory tests using various psychological tests. The most common tests are those that focus on a specific memory function, such as attention, creativity, working memory length, or long-term memory function.

The effects of even (and loud) sound on work performance were studied extensively until the 1970s. The studies found that steady loud noise does not directly affect work performance. It mainly affects strategy choices. Noise is an additional stress factor, and to compensate for it, the ability to concentrate on the main task can even improve. On the other hand, wide-ranging attention is impaired, which may impair performance in secondary tasks. Prolonged struggle in noise causes fatigue, etc. disadvantages to be listed later.

In the 1970s, it was discovered that unwanted speech sounds have a much stronger effect on work performance than even sound. The effect was seen in working memory tasks. In routine tasks, there is no effect. Since then, the focus of research has shifted to the effects of speech sounds.

The effect of speech sounds on work performance has been studied extensively in Finland (see review: Hongisto and Keränen, 2018). All Finnish and foreign studies support Hongisto's (2005) model that speech that stands out in cognitively demanding tasks, especially those that require working memory, decreases work performance by at least 10%, depending on the studies. The change in work performance depends on speech intelligibility, which can be measured using the speech transmission index STI (Fig. **12.11.2**). Work performance begins to deteriorate when the STI exceeds 0.20 and the maximum decrease is reached when the STI exceeds 0.50.

Of the different types of noise, speech has the greatest impact on work performance. There are several reasons for this:

- Speech may contain information. Human attention is directed to it sensitively and permanently.
- Speech is highly variable and complex in sound level compared to any other sound.
- The content of speech information cannot be predicted.
- Even if the same words are repeated in speech, the working memory still registers the words automatically, taking resources away from a task that requires working memory.

According to current knowledge, the brain has two central memory blocks: working memory and storage memory. The size of storage memory, i.e. long-term memory, is unlimited, but information always travels to and from there through working memory.

In working memory, three main blocks (cognitive function) can be distinguished: short-term visuospatial memory (later: visual memory), short-term word memory and central processing unit. The central unit processes information, directs mental resources to the functions required by the task, and retrieves information from the storage memory. Thus, the central unit maintains attention and guides thinking. The function of word and visual memory is to temporarily store information so that individual things can be formed into wholes, i.e. things can be understood. In this context, people are most interested in word memory, because speech has been found to hinder this particular part of working memory the most.

4□8 words remain in the word memory at a time. This amount of information can be used to form a normal sentence and substance, to which an understanding or association of the matter can be attached, if necessary. The latter are needed to make a decision about whether to remember it or whether the information is superfluous. Word memory consists of two parts: the word volume and the repetition mechanism. All information presented in the form of speech automatically enters the word volume, where it remains like "internal speech" automatically for a few seconds. With the help of a repetition mechanism, the content of the word volume can be refreshed. The repetition mechanism is basically "chanting" with internal speech or out loud. With repetition, the word is re-entered into the word volume.

Information presented in the form of speech proceeds directly from the ear to the word volume, while written information first requires it to be encoded into internal speech. Therefore, the speech heard ignores the read text and interferes with reading or verbal tasks in general. Since the sense of hearing cannot be switched off, speech noise is a major disadvantage for work tasks that require continuous and efficient use of working memory.

With learning, the need to use a CPU when performing certain tasks decreases or is used only from time to time. Because of this, tasks that are routine are usually not disturbed by speech noise. The same applies to all tasks that do not require special concentration from a person.

A person can regulate his attention to some extent. A person can try to ignore noise, and they usually succeed in this effort with certain limitations. Noise has been found to change the strategies used to perform work tasks. Noise can favor the choice of certain strategies at the expense of others or cause inflexibility in the choice of cognitive strategy. As noise reduces the resources allocated to the task, the remaining capacity is allocated to the most important parts of the task at the expense of less important parts. Depending on the task, such a strategy can either maintain task performance or weaken task performance.

As a rule, it can be noted that as the cognitive difficulty of the task increases, it becomes more difficult to do it in speech noise. These include tasks whose demands are not predictable or which require complex reasoning and shared attention.

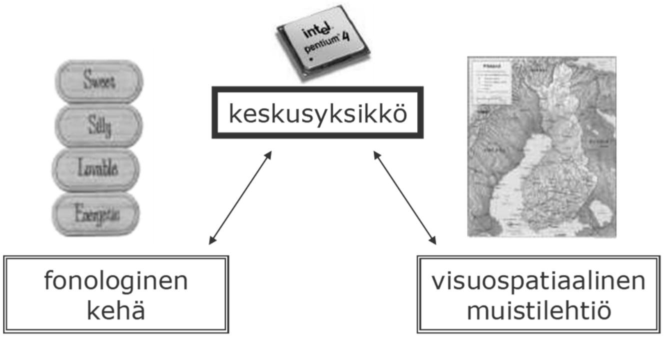

Fig. 12.11.1. According to one model, working memory would be divided into three units.

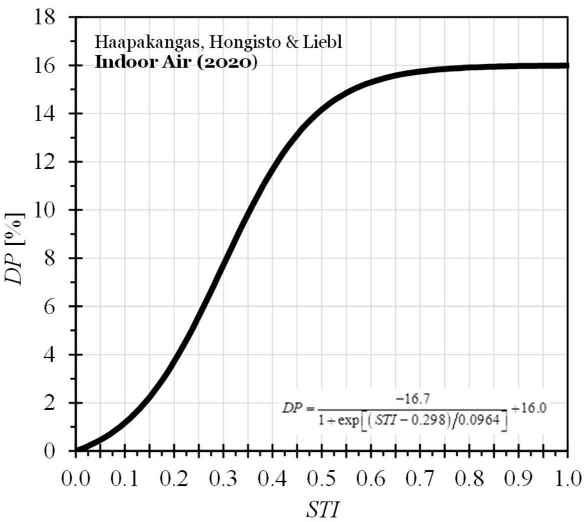

Fig. 12.11.2. Performance of tasks requiring working memory is reduced by more than 15% if background speech sounds stand out well (STI>0.50) compared to a situation where speech does not stand out (STI<0.20) (Hongisto, 2005; Haapakangas et al., 2020).

12.12 Behavioural changes (coping)

People have at their disposal coping and control methods related to self-regulation, i.e. so-called self-regulation. coping means. They can be used to regulate emotional reactions and control or change an external problem. These self-regulation methods have been found to compensate for the annoyance of noise.

In office work, for example, disturbing noise has numerous social and behavioural effects. These include changes in everyday activities such as closing doors, increasing the volume of the radio, being unfriendly, less willing to help, changing workstations and complaining about noise. Disturbing noise causes irritation, shifting work outside working hours or home, and leaving the workstation. These factors naturally have an impact on work efficiency. Struggling and changing work strategies are common compensatory management methods in noisy workplaces. This is due to the fact that reading speed or concentration usually decreases due to background sounds containing information, for example, a conversation with colleagues. Talking about the noise problem and possible noise abatement planning also takes up workplace resources. Those who are highly motivated to work (type A personality) are able to compensate for noise nuisances by exerting effort. Even if a person working in noise is able to maintain his attention for the time required to complete the task, he will still be burdened. In this case, the effects may be visible only after the task has been completed. The consequences range from fatigue to decreased cognitive performance and impaired social interactions. A noisy working environment frustrates employees if they do not achieve their own goals or those set by their supervisor due to noise.

Disturbing environmental noise can cause people to move, close windows, avoid using balconies, turn up the sound on the TV, change the location of the bedroom, degrade social relationships or make noise complaints. Noise or attempts to avoid its nuisance can cause aggression, unfriendliness, idleness, decreased willingness to cooperate or lack of willingness to participate.

12.13. Environmental sensitivity

Idiopathic Environmental Intolerance refers to a condition in which a person experiences symptoms that cause functional impairment or lifestyle limitation in a certain work or living environment, even though the same environment does not cause symptoms for the majority of people. Environmental sensitivity explains organ effects that are not caused by the direct vital effects of environmental factors. The key mechanisms are reactions in the central nervous system and bodily responses derived from it. Therefore, the symptoms are very diverse and general, and they often coincide in different environmental sensitivities. Environmental sensitivity can develop for almost any factor that has acquired a negative or even threatening significance and can thus appear in small amounts or even hints of adverse factors such as smells, electromagnetic fields, indoor air, food, sounds and wind turbine infrasound.

Environmental sensitivity occurs if the symptoms could not be demonstrated by direct physical, chemical or biological organ effects of the exposures, and no actual disease has been found that would explain the symptoms. The symptoms are explained by the responses of the body's defence systems in the central nervous system, the involuntary nervous system and the immunological system. Similar symptoms occur in many other conditions where the body's alarm systems are activated.

Some of those with symptoms from wind turbines associate their symptoms with infrasound. When assessing the effect of exposure on humans, it is essential to determine the level, duration and mechanism of action. **According to Unit 1**, infrasound also has a hearing threshold above which infrasound can be heard at decibel values. This has also been observed in a Finnish study (Rajala *et al*., 2022). Infrasound has only been found to cause negative health effects when the level is above 140 dB. Similar effects are also observed with sounds of 140 dB, the frequency of which exceeds 20 Hz. However, infrasound at the level of 140 dB is already audible. The infrasound level of wind turbines is tens of decibels below the infrasound hearing threshold. In a normal living environment, infrasounds that are much stronger than the infrasound of wind turbines are constantly detected and no health hazards have been observed (Hongisto and Oliva, 2017).

When developing environmental sensitivity to any factor, eg. For infrasound from wind turbines, the symptoms are triggered reflexively when a person assumes exposure to a harmful factor. This has been demonstrated in numerous double-blind experiments in electrosensitive subjects who could not detect electrical exposure but exhibited symptoms when they assumed they were exposed to it. In environmental sensitivity, the central nervous system is conditioned to certain conditions, environmental factors or their cues, which explains the reactions that arise and the body's responses. Environmental sensitivity can be recovered through persistent training and unlearning.

Currently, there is no research data to support the notion that non-audible infrasound or infrasound from wind turbines causes health hazards (Hongisto and Oliva, 2017; Ministry of Economic Affairs and Employment, 2017). Despite this, many people living near wind power sites (or those near the future wind power site) have the impression that infrasound from wind turbines impairs health. This negative perception of wind turbines and the health hazards they cause creates a risk of environmental sensitivity, which, at worst, can significantly restrict life and habitat. Today, such symptom images can be identified better than before. Environmental sensitivity has been added to the Finnish ICD-10 classification of diseases from the beginning of 2015 under the heading *R68.81: Continuous or repeated exceptional sensitivity to normal environmental factors.* As identification improves, recovery can be supported on behalf of healthcare professionals.

Environmental sensitivity to infrasound is increased by the concept of wind turbine syndrome launched abroad (Hongisto and Oliva, 2017). It has been accompanied by the following nonspecific symptoms, similar to those of other environmental sensitivities and not explained by any disease:

- migraine or headache including nausea, vomiting and sensitivity to light and sounds,
- dizziness
- ringing, whistling or other sound in the ears for no reason whatsoever (e.g. tinnitus),
- hearing impairment,
- locking of the ears or a feeling of pressure in the ears,
- rash or itching of the skin,
- back pain or backache,
- recurrent upset stomach,
- blurred vision,
- frequent heartbeat or palpitations,
- problems concentrating or remembering things,
- panic attacks or similar sensations.

The emergence of environmental sensitivity is illustrated in Fig. **12.13.1.** Environmental sensitivity develops as a physiological stress response to a perceived threat, and it can be caused by various kinds of stress and individual susceptibility. Environmental sensitivity usually starts with a disadvantage of comfort. Harm to comfort may be caused by some real exposure and simply by the perception that some close factor would cause health hazards. Harm to comfort is followed by avoidance of exposure (aversion), activation of the central nervous system (stress) caused by the threat, and thus the emergence of stress symptoms. It's a brain conditioning that can be dismantled through cognitive-behavioral therapy.

Environmental sensitivity is strongly associated with the so-called environmental sensitivity. The nosebo effect, which is the opposite of placebo. The Nosebo effect means that negative expectations influence our interpretation and the responses that follow. Negative perceptions can arise from one's own or others' experiences, the media, other people's speeches or behavior. Environmental sensitivity can also follow when one's own symptoms are explained by a supposed adverse factor, after which this factor or its presumed presence begins to generate reactions and responses.

Doctors and experts, as well as politicians and the media, have a great responsibility in risk communication related to environmental factors, because unjustified emphasis on harm increases public concern and provokes environmental sensitivity. In this case, the worry itself is the cause of the symptoms and not the predisposition itself.

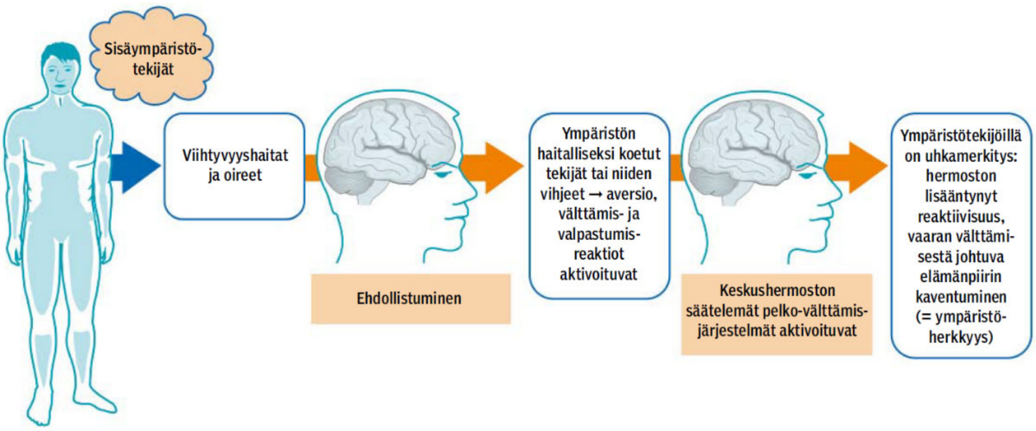

Fig. 12.13.1. Development of environmental sensitivity (Sainio and Karvala, 2017).

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13 NOISE ANNOYANCE

13.1. Definition

Sound is perceived as noise if it is unnecessary or disturbing. Sound is commonly referred to as noise if it is estimated to cause health hazards or hearing damage.

Noise annoyance is the most common harmful effect of noise. The World Health Organization (WHO) defines "*health as a state of complete physical, mental and social well-being and does not simply mean the absence of illness or weakness*." According to the WHO definition, experiencing noise disturbance is a health hazard that should be avoided in order to maintain well-being. The annoyance of noise is a health hazard because mental well-being is not balanced due to distraction. For this reason, measuring the annoyance of noise plays a key role in research into the health effects of noise. It is measured both in studies of the habitat and in psychological laboratory studies.

Annoyance is measured by questions according to ISO/TS 15666 (2003) with three verbs attached: *bother*, *disturb* and *annoy*. The standard questions and answer options are shown in Fig. **13.1.1.** When studying noise experiences in the living environment, a time period may be included (e.g. 12 months according to the standard, or shorter if necessary). In laboratory studies, the time period is omitted.

The dose-response ratio *of noise annoyance* refers to a relation that presents the proportion of all respondents who find noise highly annoyed *(%HA) or* annoyed *(%A) as a* function of sound level. Dose-response relationships are determined by conducting surveys in the living environment and by attaching to the responses a modelled sound level to which the respondent is exposed in their yard or balcony. If the response rate is high enough, the dose-response relationship derived from the responses is probably representative of the experience of the population as a whole. For the determination of the dose-response relationship %A and %HA the method shown in Fig. **13.1.2** shall be applied .

Fig. **13.1.2** shows some Finnish dose-response relationships for environmental noise in residential environments. It is normal for the sound level presented in dose-response relationships to refer to the yard sound level, because the indoor sound level cannot be determined. This naturally causes great inaccuracy in the results, since the sound insulation of the façade and the location of the living rooms in relation to the direction of noise input vary from one respondent to another. In different countries, due to the need for thermal insulation or differences in environmental noise regulations, the sound insulation of facades is at different levels, and the amount of windows kept open varies. These errors are characteristic of environmental epidemiological studies and therefore the dose-response relationship obtained in one country may not be directly applicable to another. Dose-response relationships also vary between different areas and housing types, because different areas have different land use histories and residents have different expectations regarding the sound level of their living environment.

Kun olet kotona/pihalla, kuinka paljon äänilähteen X melu kiusaa, häiritsee tai ärsyttää sinua?
Ajattele viimeistä 12 kuukautta.

5-portainen verbaalinen vastausasteikko

Ei lainkaan	Hieman	Kohtalaisesti	Paljon	Erittäin paljon

11-portainen numeerinen asteikko

Ei lainkaan										Erittäin paljon
0	1	2	3	4	5	6	7	8	9	10

Fig. 13.1.1. Measurement of noise nuisance according to ISO/TS 15666 (2003).

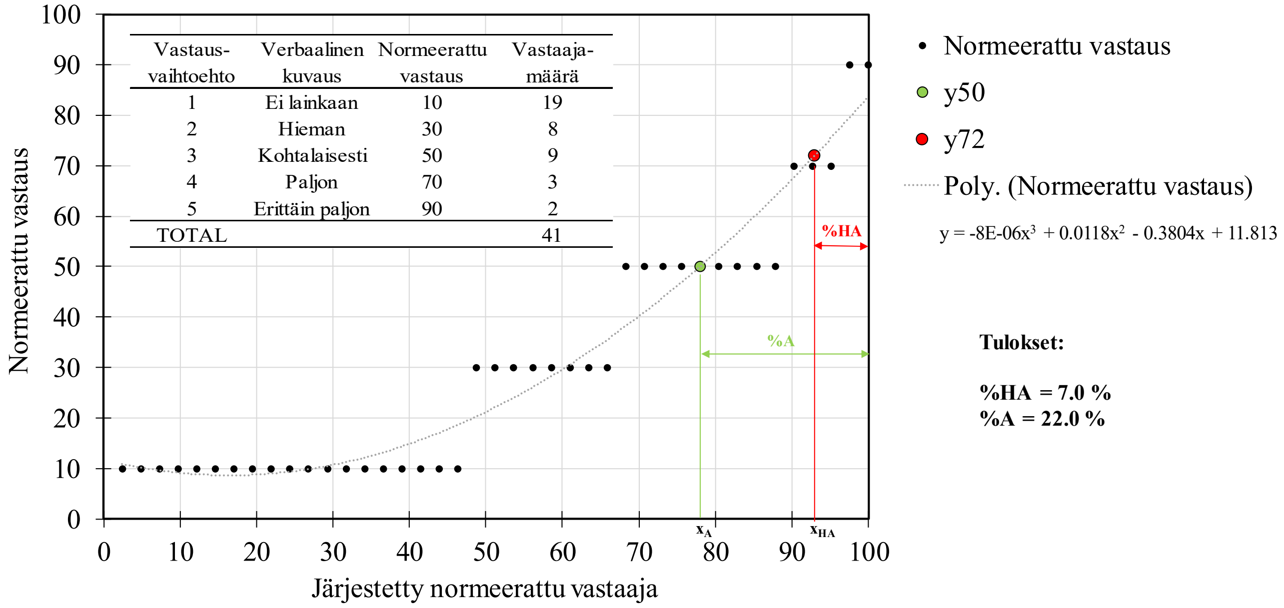


Fig. 13.1.2. Example of determining a dose-response relationship using a 5-step response scale. In the example, we use imaginary data from 41 respondents whose yard has a sound level of 40–45 dB *LAeq* (sound level zone). A similar procedure applies to all sound level zones separately. Respondents are first ranked according to the increasing response number. The answers are given a normed value on a scale of 0–100. In standardization, 100 is divided by the number of answer options. With the five-step answer option, the width is 20, so that answer option 1 is placed in the middle of it at 10. After that, we get the graph above, showing the black balls. To the graph obtained, the third-order polynomial function is fitted.

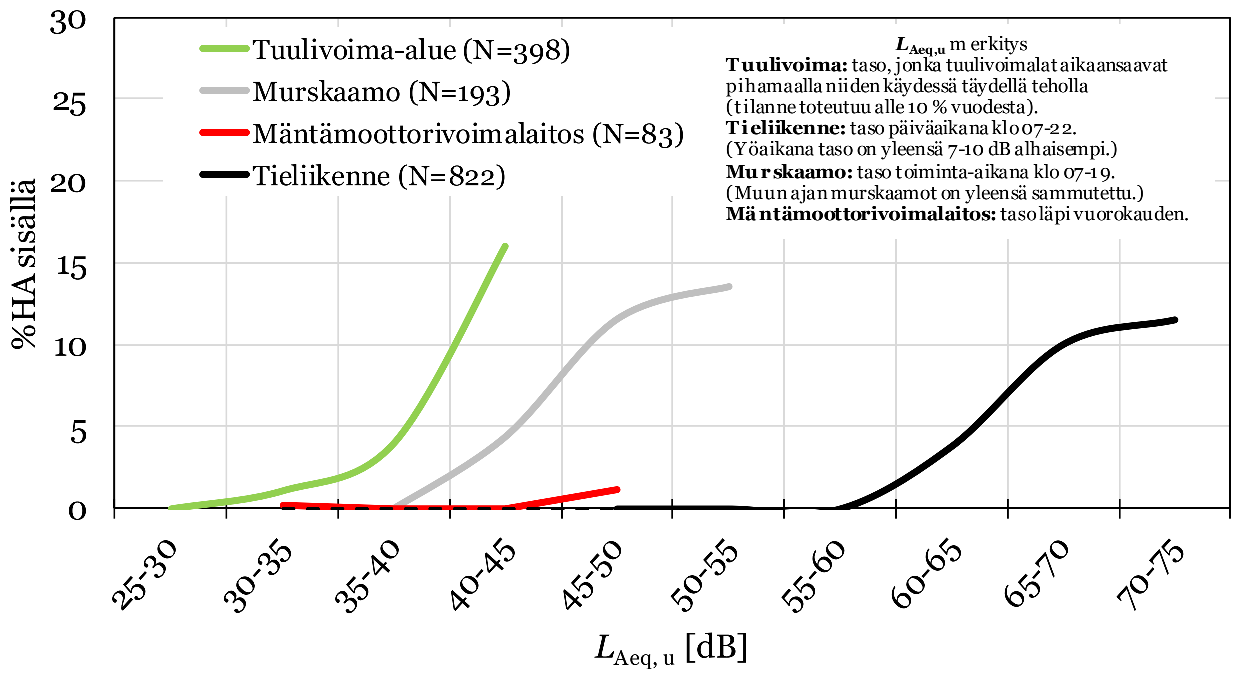


Fig. 13.1.3. Dose-response ratios of environmental noise for four different types of environmental noise. The verbal question shown in Fig. **13.1.1 was used as** the question, but the response scale was four steps (option 1 had been replaced by the description "No sound is heard"). On the horizontal axis is the A-weighted equivalent level in the yard of the respondent's residential building, LAeq,u. On the vertical axis, the probability (in percent, scale 0–100%) that the resident will find noise very disturbing indoors, %HA. Each study was conducted in different areas where the relevant environmental noise producer was nearby (Hongisto et al., 2017; Maula et al., 2019a; Maula et al., 2019b). In parentheses is the number of survey participants.

13.2 Special features and sanction

Sound can be characterized by various adjectives or technical features. The most commonly mentioned sound qualities or special features in psychoacoustics include:

1. **Frequency band:** the frequencies that occur above the hearing threshold (or background noise) in the sound being studied at all, e.g. usually 100–10000 Hz for speech or whistle only whistle frequencies.
2. **Sound frequency distribution:** SPL as a function of frequency. The colour of the voice can be experienced, for example, as rumbling, hissing, hissing or sharp.
3. **Narrowband sound (tonality):** sound has one or more distinctive frequencies that dominate.
4. **Impulsivity:** sound has shocks or rapid ascent points that dampen more slowly than rise.
5. **Amplitude modulation:** the volume of sound varies either periodically or in a random period with a frequency of less than 15 Hz. This is sometimes referred to as significant pulsation or *fluctuation strength*.
6. **Roughness:** the pulsation is dense (above 15 Hz), e.g. rain.
7. **Intermittency:** the volume of sound varies but the period is long, such as road traffic grouped by traffic lights.
8. **Informational content:** the longer entities of sound form verbal information, emotion or other association, such as speech or music.
9. **Soundscape:** a set of sounds characteristic of a particular environment that produces strong memories but does not necessarily contain specific information. These include, for example, the sounds of a shopping centre, seashore, forest, flame or kindergarten.

The most important features in terms of building acoustics and noise abatement are features 3–4, as they are subject to a penalty of 3–10 decibels in Finnish legislation, depending on the case (Table **13.2.1**). Sanction means that the sanction k and measurement uncertainty K are added to the measured A-weighted average sound level L*Aeq* before comparing the result with the guideline value for the sound level. In the case of housing health guidelines, the sanction is only applied to the period during which the special feature occurs. In the case of the Sound Environment Regulation and the guideline values for environmental noise, the sanction shall be applied in full as soon as it can be demonstrated that the characteristic is caused by the noise source under consideration.

In addition, action levels for low-frequency noise are presented in the housing health guidelines. Therefore, feature 2 is also important. There is a 5–10 dB sanction for night-time (22–07) environmental noise, but it is excluded in this review, as well as a correction for 5 dB evening time (19–22) and weekend correction for heavy weapons and blasts (Government, 2017).

The following chapters discuss research data on the verification of key characteristics and their effects on noise nuisance.

Table 13.2.1. Sanctions in key legislation for the specifics of sound.

	Sanktio *k* [dB]	
	Kapeakaistaisuus	Impulssimaisuus
Ääniympäristöasetus		
Ympäristöministeriön asetus 796/2017	3-5	3-5
Ympäristömelun ohjearvot		
Valtioneuvoston päätös 993/1992	5	5
Ympäristöministeriön asetus 1107/2015 (tuulivoimalat)	5	5
Asumisterveysasetus		
Sosiaali- ja terveysministeriön asetus 545/2015	3 tai 6	5 tai 10

13.3 Narrowband audio

13.3.1. VERIFICATION

In Finland, narrowband is established in buildings and surroundings using one of the following methods:

- Valvira (2016): Health hazard surveys in accordance with the Housing Health Decree (STM, 2015)

- Sound environment guidelines (2018): sounds of building services equipment in accordance with the Sound Environment Decree (YM, 2017)
- ISO 1996-2 (2007, 2017) and ISO PAS 20065 (2016): measurements according to noise level guideline values (VN, 1992)
- IEC 61400-11: Measurements according to wind turbine noise guideline values (VN, 2015)

Valvira's (2016) method states narrowband as follows:

1. If narrowband stands out weakly, then the correction is 3 dB. A subjective assessment is sufficient for this.
2. If it is clearly audible, then the correction is 6 dB. Narrowband can generally be considered clearly audible if the following conditions are met:
   - the component or components cause the intensity of the component to exceed the hearing threshold,
   - In the non-frequency-weighted terse spectrum, the terse spectrum in question. the terse pressure level of the component is at least 5 dB higher than the average of the terse pressure levels of the lower and upper ters, and
   - the average sound level of the total noise prevailing at the same time is less than 55–60 dB.

The problem with Valvira's method is that a sanction of 3 dB can be imposed on subjective grounds, which can lead to conflicting results. In addition, a vote requiring a penalty of 6 dB will not result in a sanction if the vote is located on the boundary of two one-third octave bands, in which case the vote will raise the level of both bands equally.

In the Sound Environment Guideline (2018), narrowband is excluded by auditory observation or objectively determined in accordance with ISO 1996-2.

In the ISO 1996-2 (2017) method, narrowband is established either with a rough survey method or with a precise engineering method, which in turn is done according to ISO PAS 20065 (2016). In the survey method, the existence of tonality is established by performing a tersic band analysis of the sound. It is used to determine the difference between the level of the steel band of the suspected narrowband and the average level of the surrounding bands. Narrowband occurs if the difference is greater than

- 15 dB (25−125 Hz)
- 8 dB (160–400 Hz)
- 5 dB (500−10000 Hz).

This means that narrowband is formed more easily at high frequencies than at low frequencies. The Survey method has the same flaw as Valvira's method.

Method ISO 1996-2 (2007), which preceded ISO PAS 20065 (2016), works broadly as follows (**Fig. 13.3.1**):

- the SPL is analysed with FFT in a narrowband manner (e.g. at 1 Hz resolution) to obtain the spectrum, i.e. the SPL as a function of frequency,
- spectrum A-weighted,
- the spectrum is searched for noise bags, i.e. candidates who are at least 6 dB high,
- the analysed frequency band is divided into critical bands that model the width of the hearing bands, and the critical frequency band mid frequencies (*FC*) are placed on the most distinctive voting candidates,
- the candidate shall vote if the -3 dB bandwidth is less than 10 % of the critical band width,
- adjust the linear regression to frequencies not included in the sound from which the SPL of the masking sound in the critical frequency band (*Lpn*) is obtained by summing the agreed SPLs throughout the critical band (including the noise range),
- calculate the total sound level L_{pt} from the -6 dB bandwidth of the tone,
- if the SPL of the sound is above the hearing threshold, the sonic insensitivity, AT, shall be determined by the equation,

(13.3.1) $$A_T = L_{pt} - L_{pn} + 2 + \log_{10}\left[1 + \left(\frac{f_c}{502}\right)^{2.5}\right]$$

The ISO method cannot detect (without additional self-defined steps) sounds below 50 Hz, so its application to low-frequency narrowband audio is questionable. The ISO method presents a step-by-step sanction model (Fig. **13.3.2**), but it is not applied in Finland. The narrowband determination method is currently presented in ISO PAS 20065 (2016), but is similar to that presented here.

The IEC method is similar in principle to the ISO method, but it has been modified to meet the needs of wind power noise measurement.

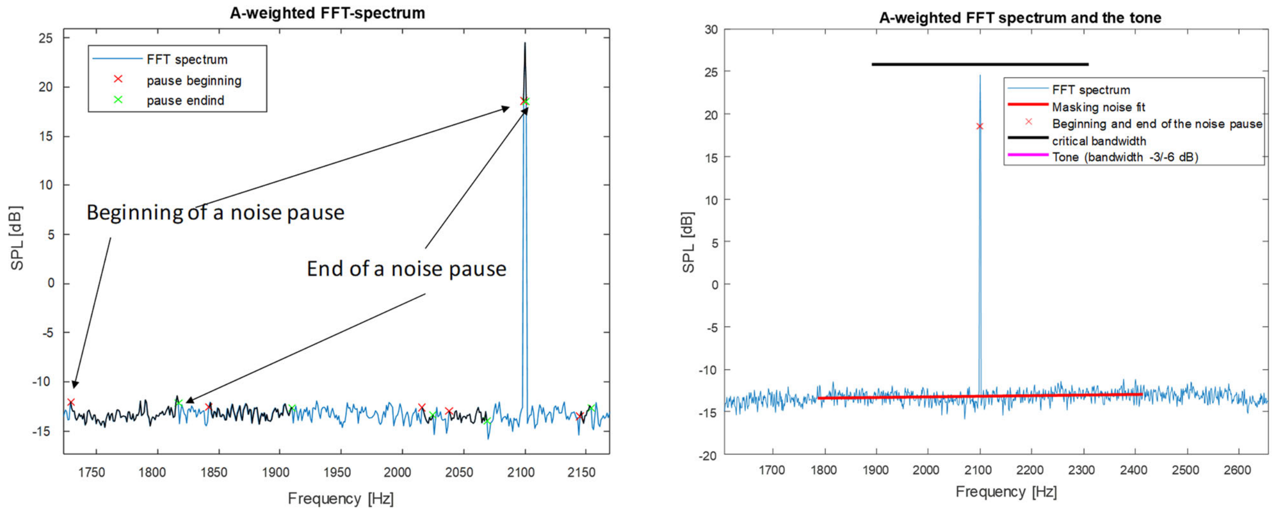
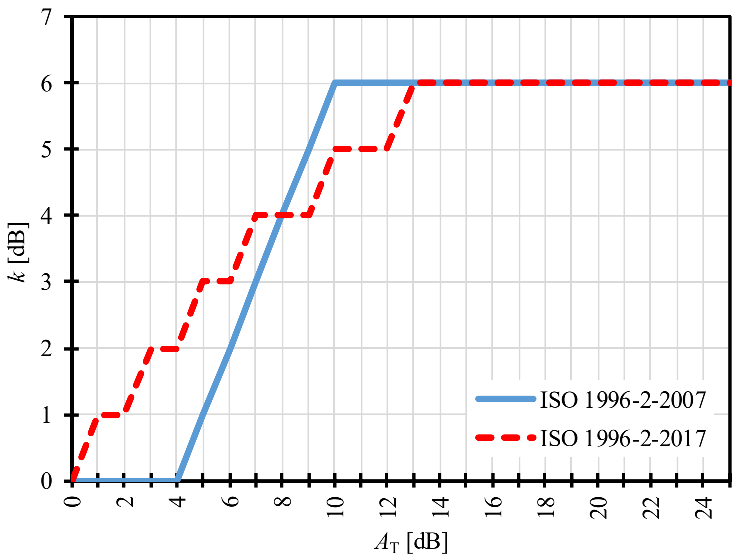


Table C.1 — Widths of critical bands

Centre frequency, f_c, Hz	50 to 500	Above 500
Bandwidth, Hz	100	20 % of f_c

Fig. 13.3.1. ISO method. Search for noise bands, overlay sound fitting, and definitions of critical frequency bands.



13.3.2. The stepwise sanction model of the ISO 1996-2 standard was not supported in psychoacoustic research (Hongisto et al., 2019).

13.3.2. ANNOYANCE

Below is a summary of the largest psychoacoustic laboratory study to date on narrowband sound distraction sanction (Oliva et al., 2017; Hongisto et al., 2019). According to that provision, the narrowband penalty ranged from 0 to 12 dB and strongly depended on the values of objective parameters according to ISO 1996-2, namely sound frequency, *fT* [Hz] (mean peak frequency) and sound inseparability, *AT* [dB]. Short sounds were used in the experiment, so the sanctions received may not be the same as in a living environment where exposure can occur continuously. The results are likely to apply well to the mutual significance of the objective variables *fT* and *AT* for the sanction.

Need. ISO 1996-2:2007 provides a detailed method for determining narrowband in the FFT spectrum of sound. In the standard, narrowband is described by two variables: tone frequency, f_T [Hz] (average peak frequency), and sound inseparability, A_T [dB] (frequency peak height). According to the standard, the sanction would depend only on the value of A_T, and standard sanctions in national regulations would not be justified. The scientific basis of the standard's sanction model is lax. There are no studies in the literature on the factors on which narrowband noise sanction depends, especially at low noise levels (25–to 35 dB L_{Aeq}). This is an essential sound level range when indoors and assessing the health effects of noise.

Target. The aim was to find out how narrowband noise should be sanctioned so that it corresponds to the subjective experience of annoyance. The detailed objective was to determine how *fT* and *AT* affect the sanction of narrowband sound when the sound level is low, corresponding to the typical sound level of a residential apartment.

Methods. 40 subjects were recruited for the psychoacoustic experiment. There were 20 narrowband sounds studied. The sound frequencies studied were 50, 110, 290, 850 and 2100 Hz. The sound was separated by 5, 10, 18 and 25 dB and was caused by wideband noise with inverted A-weighting. The level of all tonal sounds was L_{Aeq} = 25 dB. In addition, the experiment had 14 broadband reference tones at levels 19– to 45 dB L_{Aeq}, which could be used to determine the sanction. The reference sounds were also broadband noise with an inverted A-weighting spectrum. The method of determining the sanction is shown in Fig. **13.3.3.** The subjects rated the annoyance of each sound on a scale of 0–10.

Results. The sanction obtained in the experiment depended on different f_T and A_T values as shown in Fig. **13.3.4.** At its highest, the sanction was as high as 12 dB. This means that certain narrowband audio with a sound level of 25 dB L_{Aeq} was perceived as disturbing as wideband audio with a sound level of 37 dB L_{Aeq} that does not contain narrowband. The highest sanctions were observed with high values of f_T. At low f_T values, no sanction was observed, although A_T was large. The study does not support standard sanctions for narrowband noise. The penalty *k* obtained in Fig. **13.3.4** could be calculated using the equation

$$k = A_T\left[-0.361 + 0.326 \cdot \tan^{-1}\left(\frac{6 f_T}{1000} - 0.5148\right)\right] \tag{13.3.2}$$

Conclusions. Narrowband sound can increase the annoyance of the sound, but the penalty *k* [dB] describing the annoyance increase depends on the values of f_T and A_T. The results suggest that narrowband at high frequencies is a greater risk factor for annoyance than narrowband at low frequencies. However, in a residential environment, noise disturbance is also affected by factors other than the characteristics measured by sound. For example, if the sound is varied, comes from an unnecessary source, is clearly distinguishable from other background sound, or is completely new and unprecedented, it may be perceived as disturbing in different ways than found in this study.

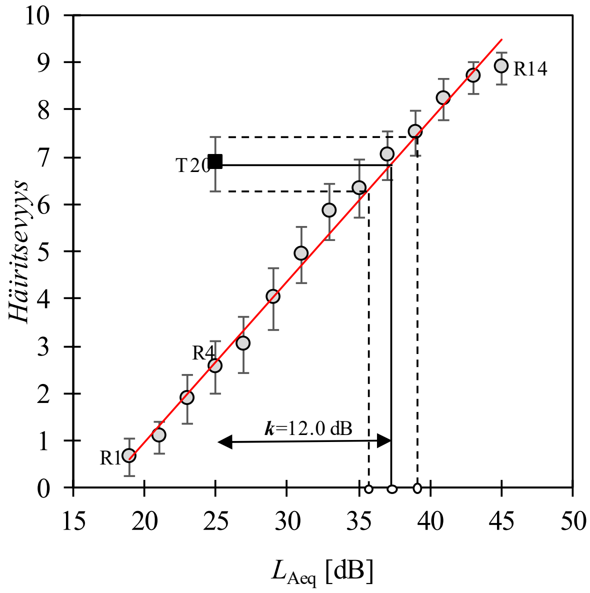


Fig. 13.3.3. Sanction *k* was determined as described herein. R1–R14 (circles) indicate the average annoyance of the reference sounds and the mustache the 95% confidence interval. The red line represents the first-order adaptation to reference sounds. T20 (black square) represents one narrowband sound with a sound level of *LAeq* = 25 dB. For the sound T20, the annoyance average is about 7. The reference sound was considered equally disturbing if its sound level is *LAeq* = 37 dB. Due to this, the sanction is *k* = 37□25 = 12 dB. Dashed lines are used to determine the 95% confidence interval of the sanction.

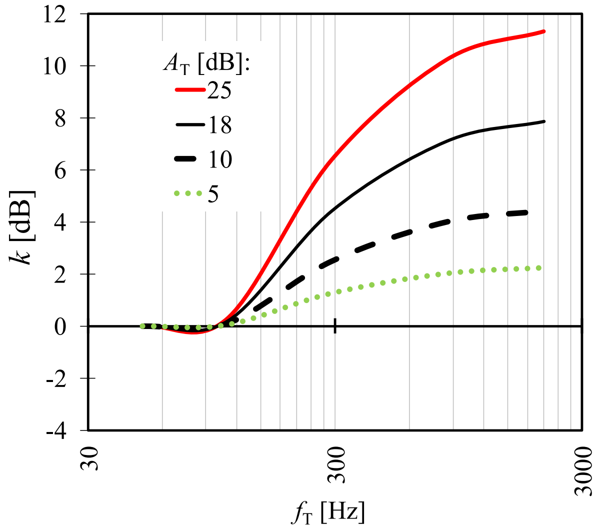


Fig. 13.3.4. The dependence of narrowband audio annoyance sanction *k* on sound frequency *fT* and sound separation *AT according* **to equation 13.3.2** (Hongisto et al., 2019).

13.4 Impulsive sound

13.4.1. Verification

In Finland, impulsiveness is usually diagnosed by one of the following methods:

- Valvira (2016): measurements in accordance with the housing health guidelines
- Nordtest (2002): measurements of environmental noise

Both methods present sanction models. However, the sanction applies only to the period during which impulsivity occurs. Since impulses do not necessarily occur continuously, applying a sanction is laborious and requires calculation.

According to Valvira's method, impulsive or impact-like noise is noise in which one or more loud sounds lasting less than one second can be distinguished. Impulsive noises are characterized by:

- A rapid and large increase in sound level at the beginning of a sound event, typically 20 dB/s (Valvira's guidelines have "ms", but this is a mistake, because the rate of increase of impulses is usually always less than 2 dB/ms).
- A fairly short standard proportion of sound level after rising; typically 0□100 ms.
- Variable length and speed of noise attenuation at the end of the signal. Typically, it takes 30 to 500 milliseconds to attenuate 20 dB. In anechoic conditions and large spaces, the attenuation time of 20 dB may be longer.
- Repetition less than 30 events (impact sounds) per second.

The correction for impulse noise varies and is 5 or 10 dB depending on the nature of the impulse noise. Impulse noise is divided into two quality classes in terms of quality:

- Highly impulsive noise, which can be caused by, for example, shots from small-calibre weapons, hammering of wooden and metal objects, impact pile ramming machines, and pneumatic rods and hammers when the sound exposure level of a single noise event (A-frequency weighted) > 55 – 60 dB and the noise is clearly distinguishable from background noise. The harmfulness correction for the AE level L of each event is 10 dB.
- (Ordinary) impulse noise, which includes impulse noise that is clearly distinguishable from background noise and which does not fall clearly into the previous category, e.g. because they have a lower LAE level. The harmfulness correction to the LAE level for each noise event is 5 dB. Typically, impulsive noise from outside belongs to class 2.

The application of Valvira's method is outlined in Fig. **13.4.1.**

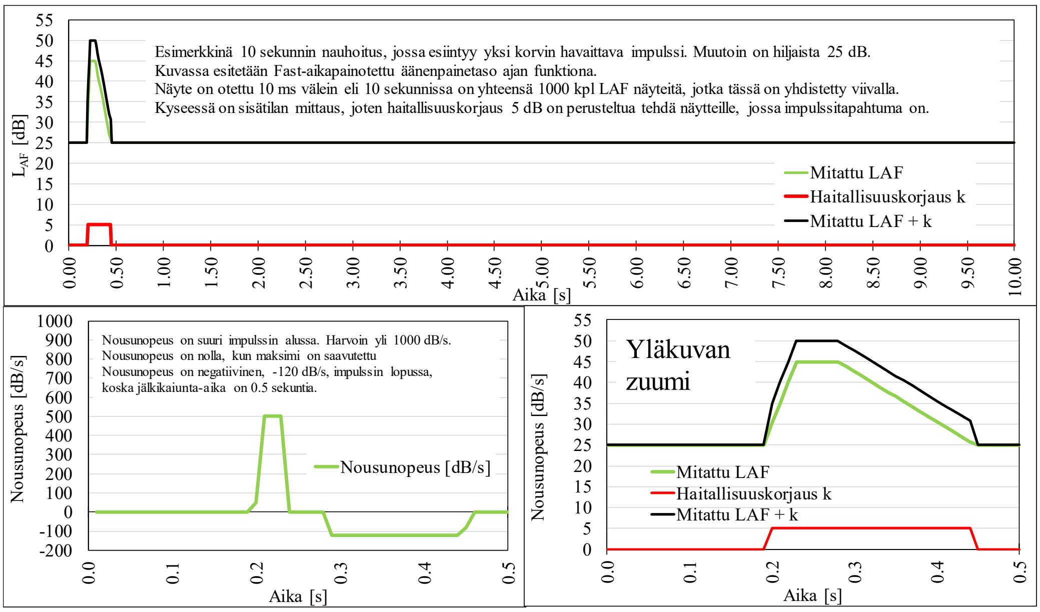
In Nordtest method, each impulse is described by two parameters: the rate of rise *Ron* [dB/s], which describes the rate of increase of the impulse, and the level difference DL [dB], which indicates how much the maximum point of the impulse differs from the background. The main characteristics of impulse are as follows (Fig. **13.4.2**):

1. The sound signal is A-weighted. From the A-weighted audio signal, the fast time constant is used to determine the L_{AF} time profile with a time resolution of 10 ms.
2. The ascent begins ($L_{AF,start}$) when the steepness of a straight line fitted between two consecutive L_{AF}*'s* exceeds 10 dB/s.
3. The ascent ends ($L_{AF,end}$) when the steepness of two consecutive lines fitted between L_{AF}*'s* falls below 10 dB/s.
4. If a new ascent starts less than 50 ms after the end of the previous ascent, the ascents can be combined.
5. The rate of ascent R_{on} is the slope of the first-order fitting line matched to the ascent.
6. The impulse level difference is recorded as $DL=L_{AF,end}-L_{AF,start}$.
7. The significance of the impulse is P (*prominence*) by equation **13.4.1.**
8. The penalty k is calculated by the equation **13.4.2.**

(13.4.1) $$P=3\cdot\log_{10}\left(R_{on}\right)+2\cdot\log_{10}\left(D_L\right)$$

(13.4.2) $$k=\begin{cases}1.8\cdot(P-5), & P>5\\ 0, & P\le 5\end{cases}$$

It is proposed that the sanction be applied to the 30-minute average $L_{Aeq,30min}$ and the sanction will be determined based on the maximum significance observed during the 30-min period. There are very loose psychoacoustic grounds for determining sanctions (Pedersen, 2000), but it seems to work reasonably well according to the psychoacoustic study presented below (Rajala &; Hongisto, 2019&2020).



$$\boxed{L_{AE}=L_{Aeq,T}+10\lg\left(\frac{T}{t_0}\right)}$$

	Vain impulssiosuus		Koko ääninäyte	
	LAeq,0.26s [dB]	LAE,0.26s [dB]	LAeq,10s [dB]	LAE,10s [dB]
T [s]	0.26	0.26	10	10
Mitattu LAF	40.7	34.8	27.9	37.9
Mitattu LAF + k	45.7	39.8	31	41
Edellisten ero	**5.0**	**5.0**	**3.1**	**3.1**

Fig. 13.4.1. Application of Valvira's method to a single impulse when the measurement duration is 10 seconds (top image). Without sanction, the average sound level is $L_{Aeq,10s}$ = 27.9 dB. The criterion of climb rate of 20 dB/s is exceeded (bottom

left image), there are less than 30 impulses per second, the impulse lasts less than a second, and the maximum lasts less than 100 ms, so it is an impulse. The measurement has been carried out indoors, in which case the sanction (harmfulness correction) is $k=5$ dB. Since the correction according to the Housing Health Decree (sanction k) is given only for the time of the impulse (bottom right image), and the sanction is not applied to impulse-free time, the effect on the average sound level is not a full 5 dB but 3.1 dB. The harm-corrected value is $L_{Aeq,10s}$ = 31.0 dB. If, on the other hand, the measurement results are compared with the guideline values for environmental noise or the sound environment setting, the penalty of 5 dB is applied for the entire period and the corrected value is 27.9 + 5.0 = 32.9 dB.

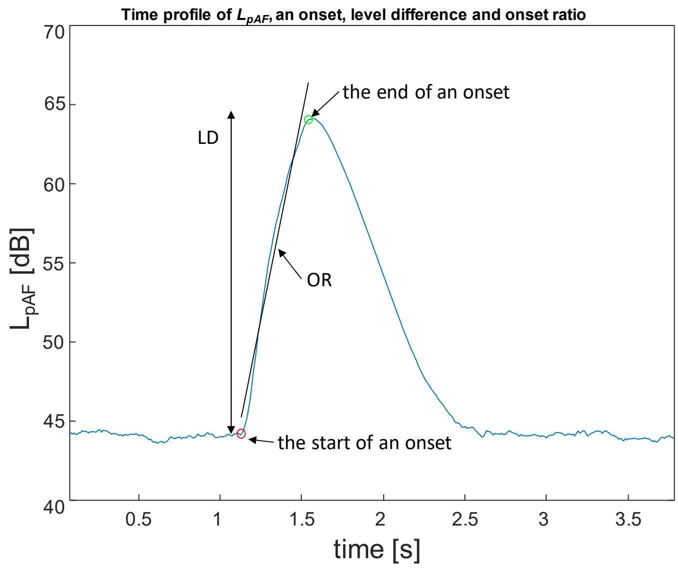


Fig. 13.4.2. Impulse search in a time profile. For the impulse of the image, the level difference (LD) is DL=20 dB and the rate of rise *(OR, onset rate*) is Ron=66 dB/s.

13.4.2. Annoyance

Below is a summary of the most extensive psychoacoustic laboratory study to date, which dealt with the disturbing sanction of impulse sound (Rajala &; Hongisto, 2019&2020). Impulses were regularly performed every 3 seconds. The impulse sanction ranged from 0 to 8 dB. The value depended heavily on the values of the objective variables R_{on} *(*onset rate) and D_L (level difference) according to the Nordtest method. The experiment used short sounds, so the sanctions obtained may not be the same as in a habitat where exposure can occur continuously, but the results are a good indication of the mutual significance of the objective variables R_{on} and D_L for sanction.

Need. Finnish noise legislation includes standard sanctions for impulse noise of 3, 5 and 10 dB. According to ISO 1996-1 (2016), the sanction is either 5 or 12 dB. The values are based on extensive studies of gunshot noise from the 80s and 90s (Rice, 1996). Nordtest (2002) presents a consistent method for detecting impulsivity in a sound signal, as well as a sanction model. However, the scientific basis for the sanction model is lax.

Target. The aim was to determine how the variables R_{on} and D_L of the Nordtest method affect the annoyance sanction of impulse noise when the sound level is close to the guideline value for outdoor noise.

Methods. 32 subjects were recruited for the psychoacoustic experiment. There were 33 impulse sounds studied on spectrum S1 (wideband sound similar to ventilation) and 33 on spectrum S2 (higher frequency broadband sound). Impulse rise rates R_{on} were between 5 and 800 dB/s. The differences in impulse levels DL were between 5 and 40 dB. The level of all impulsive sounds was L_{Aeq} = 55 dB. In addition, the experiment included 8 broadband reference tones on spectrum S1 at levels 49– to 70 dB L_{Aeq}, which made it possible to determine the sanction of impulsive sounds (see **Fig. 13.3.3).** The subjects rated the annoyance of each sound on a scale of 0–to 10.

Results. The dependence of the sanction on R_{on} and D_L is shown in Fig. **13.4.3.** At its highest, the sanction was as high as 8 dB. This means that a certain type of impulsive sound with a sound level of 55 dB L_{Aeq} was perceived as disturbing as a wideband sound with a sound level of 63 dB L_{Aeq} that is not impulsive. The highest sanctions were observed with high R_{on} and D_L values. With low R_{on} values, no sanction was observed, although the D_L was high. Nordtest's model gave clearly higher sanction values than observed when R_{on} ≥ 200 dB/s.

Conclusions. Impulsive sound can increase the annoyance of sound, but the penalty *k* [dB] describing the annoyance increase depends on R_{on} and D_L. Impulse noise at high ascent rates would be a greater risk factor for annoyance than impulse noise at low rise rates. However, in a residential environment, noise disturbance is also affected by factors other than the characteristics measured by sound. For example, if the sound is varied, comes from an unnecessary source, is clearly distinguishable from other background sound, or is completely new and unprecedented, it may be perceived as disturbing in different ways than found in this study.

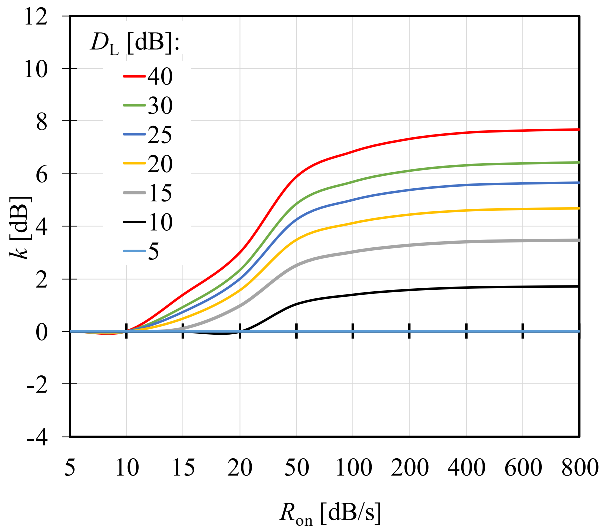


Fig. 13.4.3. The annoyance sanction for impulsive noise *k as a function of the impulse parameters* Ron (*rate of rise) and* DL *(level difference) determined according to Nordtest (2002*). The curves are derived from data (Rajala &; Hongisto, 2019&2020).

13.5 Amplitude modulated sound

13.5.1. Verification

Amplitude modulation, or periodic pulsation, became a topical special feature only in the 2010s, when wind turbines began to become more common. Wind turbines cause periodic pulsation at a frequency that depends on the rotational speed and the number of blades. Amplitude modulation is described by the modulation frequency f_m [Hz] and modulation depth D_m [dB], as defined in Fig. **13.5.1.** Modulation can apply to the entire frequency band of sound or a specific part of it.

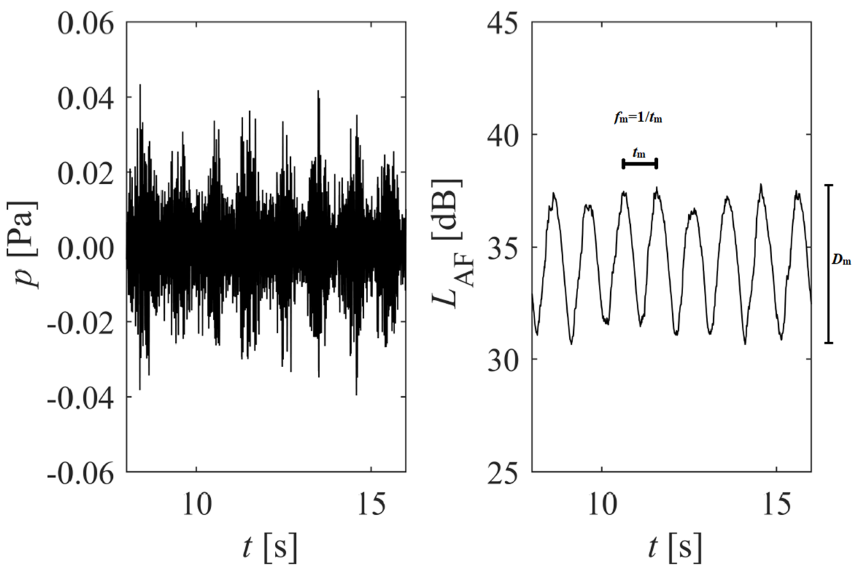


Fig. 13.5.1. Amplitude-modulated sound means that the volume is periodically varied. The Figure shows sinusoidal amplitude modulation and its graphs f_m and D_m (Virjonen et al., 2019).

Sanctions have been proposed for amplitude-modulated wind turbine sound, as several psychoacoustic experiments have found amplitude-modulated wind turbine noise to be more disturbing than flat wind turbine noise with the same L_{Aeq}. For the time being, sanctions are not yet in the legislation of any country, because there is no standardized method for measuring amplitude modulation. The annoyance surcharge has been taken into account so that the guideline values for wind turbine noise are 5–10 dB stricter than for other noise types. The Ministry of the Environment (2014) guidelines on measuring wind turbine noise include a mention of amplitude modulation, i.e. a sanction for significant pulsation, although Decree 1107/2017 does not require it. Taking into account the precautionary principle, it can be estimated that when the modulation depth of the sound produced by the wind turbine exceeds 3 dB and the modulation frequency is 0.5–30 Hz, the noise is meaningfully pulsating (amplitude modulated) in nature and it is appropriate to take the sanction into account in the measurement result. However, the regulations do not include sanctions, so no sanctions will be issued for pulsation for the time being.

13.5.2. ANNOYANCE

Below is a summary of the most extensive psychoacoustic laboratory study to date on the distraction sanction of amplitude-modulated sound (Virjonen et al., 2019). According to it, the sanction strongly depends on both f_m and D_m values, ranging from 0 to 12 dB. Short sounds were used in the experiment, so the sanctions obtained may not be the same as in a habitat where exposure can occur continuously, but the results are a good indication of the mutual significance of the objective variables f_m and D_m for sanction.

Need. Wind turbine noise is often amplitude modulated (AM). AM means that the sound has periodic variations in intensity, which makes the sound stand out more easily from the background and this can increase distraction. The main characteristics of AM sound are modulation frequency f_m [dB] (speed of variation) and modulation depth D_m [dB] (intensity of variation). The sanction is difficult to justify because the scientific evidence is limited and there is no standardised method for measuring.

Target. The aim was to determine how f_m and D_m affect the annoyance sanction of AM's sound when the sound level is low, corresponding to the sound levels found in residential environments.

Methods. 40 subjects were recruited for the psychoacoustic experiment. There were 35 AM sounds studied on spectrum S1 (spectrum resembling wind turbine sound) and 35 on spectrum S2 (spectrum resembling road traffic sound). The modulation frequencies studied were 0.25, 0.50, 1, 2, 4, 8 and 16 Hz. The modulation depths studied were 1, 2, 4, 8 and 14 dB. The level of all AM votes was L_{Aeq} = 35 dB. In addition, the experiment had 11 broadband reference tones for each spectrum at levels 19 – 49 dB L_{Aeq}, which made it possible to determine the sanction as shown in Fig. **13.3.3.** The subjects rated the annoyance of each sound on a scale of 0–to 10.

Results. The dependency of the sanction on different *fm* and *Dm* values is shown in Fig. **13.5.2.** The maximum sanction was 12 dB. This means that a certain type of amplitude modulated sound with a sound level of 35 dB L_{Aeq} was perceived as disturbing as wideband audio with a sound level of 47 dB L_{Aeq} that is not amplitude modulated. It is difficult to justify the standard sanction, because with low f_m values no sanction was detected, even though the D_m was high. Spectrum S1 and S2 yielded similar sanction values.

Conclusions. Amplitude-modulated sound can increase the annoyance of the sound, but the penalty k [dB] describing the annoyance increase depends on the values of f_m and D_m. The results can also be used to estimate the sanction for wind power noise when f_m and D_m are known. The results suggest that high-frequency AM would be a greater risk factor for annoyance than low-frequency AM. However, in a residential environment, noise disturbance is also affected by factors other than the characteristics measured by sound. For example, if the sound is varied, comes from an unnecessary source, is clearly distinguishable from other background sound, or is completely new and unprecedented, it may be perceived as disturbing in different ways than has been observed in this study.

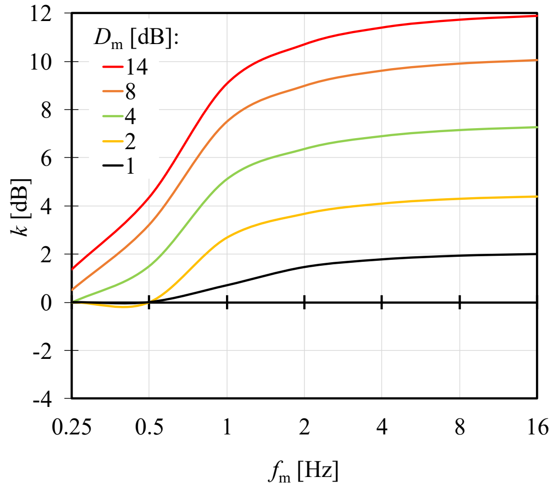


Fig. 13.5.2. Amplitude-modulated sound distraction sanction *k as* a function of parameters *fm* and *Dm* (Virjonen et al., 2019).

13.6. Sound frequency distribution

The frequency distributions of sounds vary greatly, even if they are not narrowband. It is not possible to create unambiguous measurements or combinations of a few dimensions for the frequency distribution of sound, because the same values can be obtained with very different spectra. The best-known measurements are the A-weighted SPL L_{pA} and the loudness L_N (*loudness*). However, the same value can be achieved with sounds that are very different in spectrum (and distraction). For this reason, surprisingly little spectrum-induced interference research has been conducted on a generic level, even though spectrum is the most common measurement of sound quality after the average sound level.

Hongisto et al. et al. (2015) studied the annoyance of 11 smooth broadband sounds with different spectral but similar A sound levels (42 dB L_{Aeq}) in the laboratory using an exposure time of 90 seconds. They found that sounds that were predominantly high-frequency were significantly more distracting than sounds that were predominantly low-frequency (Fig. **13.6.1**). Low-frequency sounds were preferably described with the adjective "rumbling" and high-frequency sounds preferably with the adjective "hissing". Hissing has also been found to be a factor that increases annoyance in other studies. When the overall level differs significantly from that used in the study (42 dB L_{Aeq}), a different result can be obtained between the sounds, partly due to the fact that the shape of the standard loudness curves depends on the loudness level.

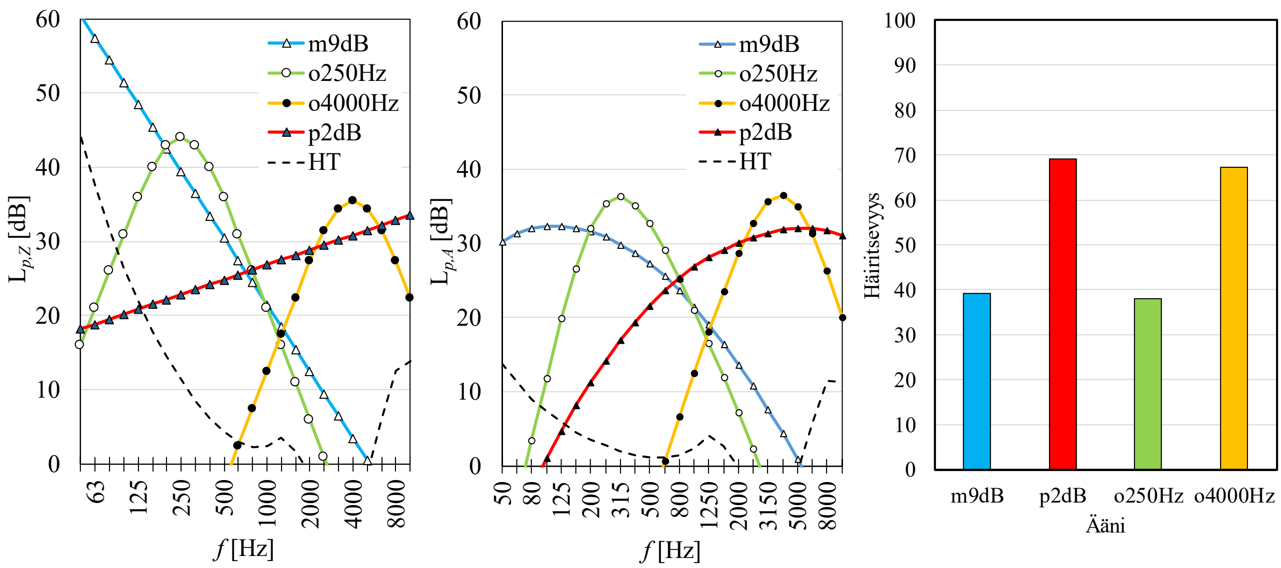


Fig. 13.6.1. According to the study, high-frequency broadband audio (p2dB) was more distracting than low-frequency broadband audio (m9dB). Similarly, high-frequency octave noise (o4000Hz) was more disturbing than low-frequency octave noise (o250Hz). HT is the hearing threshold according to ISO 226.

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ANNEX 1. FINNISH REGULATIONS AND GUIDELINES

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