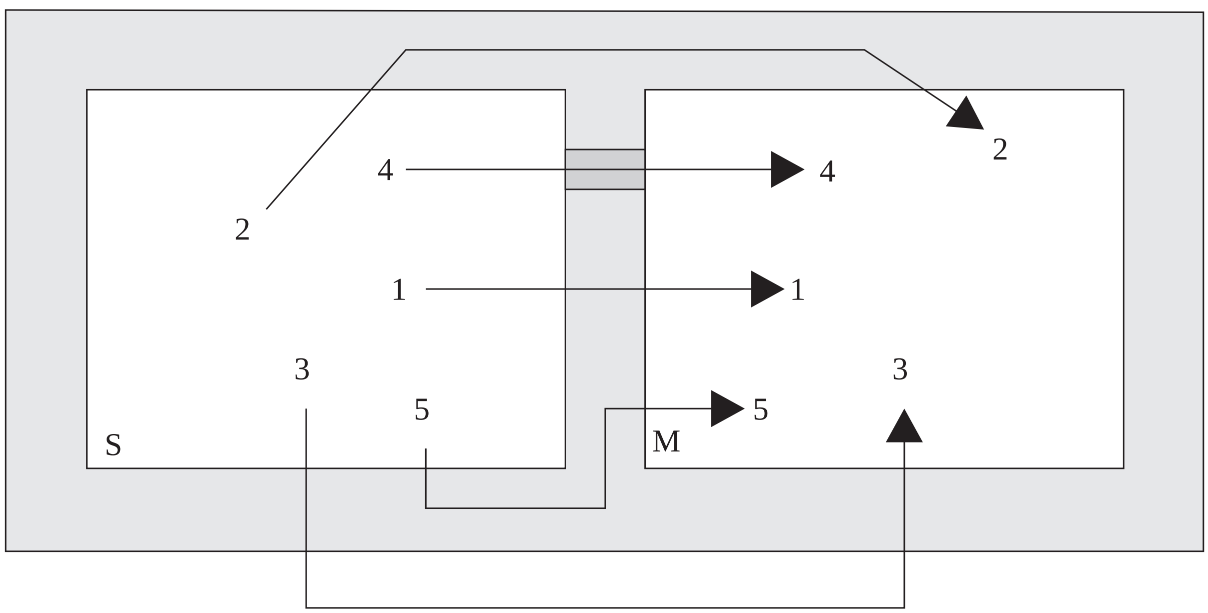
STRUCTURAL DYNAMICS, VOL. 10

Building and Room Acoustics

P.H. Kirkegaard



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Preface

This book is written related to teaching in Building and Room Acoustics for 7. semester students in Architectural Design at Aalborg University. The book gives a presentation of fundamental quantities in linear acoustics. Further, goal of the book is to present the principles for good acoustic design of rooms as well as sound isolation.

Aalborg University, November 2004
P.H. Kirkegaard

Symbolliste

A	ækvivalent absorptionsareal	m^2
B	bulkmodulet	Pa
I	lydintensitet	W/m^2
I_0	referencelydintensitet	W/m^2
L_{AeqT}	energiækvivalent, A-vægtet lydtrykniveau	dB
L_n	trinlydniveau (laboratoriemåling)	dB
L'_n	trinlydniveau (feltmåling)	dB
$L_{n,w}$	vægtet trinlydniveau (laboratoriemåling)	dB
$L'_{n,w}$	vægtet trinlydniveau (feltmåling)	dB
L_p	lydtrykniveau	dB
L_{pA}	A-vægtet lydtrykniveau	dB
L_S	lydtrykniveau i senderum	dB
L_M	lydtrykniveau i modtagerum	dB
L_u	udendøres støjniveau	dB
L_w	lydeffektniveau	dB
L_{wA}	A-vægtet lydeffektniveau	dB
P	lydeffekt	W
P_0	referencelydeffekt	W
Q	retningsfaktor	
R	reduktionstal (laboratoriemåling)	dB
R'	reduktionstal (feltmåling)	dB
R_w	vægtet reduktionstal (laboratoriemåling)	dB
R'_w	vægtet reduktionstal (feltmåling)	dB
S	areal	m^2
T_C	luftens temperatur i Celsius	^oC
T_K	luftens temperatur i Kelvin	K
T	periode	s
T_{Sab}	efterklangstid	s
V	volumen	m^3
Z_c	karakteristiske impedans	Pa s/m

c	lydhastighed	m/s
f	frekvens	Hz
f_c	kritisk frekvens	Hz
f_r	egenfrekvens	Hz
i	imaginær tal $i = \sqrt{-1}$	
$p(t)$	lydtryk	Pa
$p_t(t)$	totale lufttryk	Pa
p_s	statisk lufttryk	Pa
$\tilde{p}$	effektiv lydtryk	Pa
p_0	referencelydtryk	Pa
r	afstand	m
t	tid	s
α	absorptionskoefficient	
ε	lydenergitæthed	J/m^3
λ	bølgelængde	m
ρ_s	luftens statisk densitet	kg/m^3
$\rho(t)$	luftens densitet	kg/m^3
ΔL	trinlyddæmpning	dB
ΔL_w	Vægtet trinlyddæmpning	dB
$\Delta L_{w,c}$	Referencedæk af beton	dB
$\Delta L_{w,t}$	Referencedæk af træ	dB

CHAPTER 1
Introduction

The teaching of sound, its beginning, propagation and audibility is called acoustics, derived from the Greek expression $\alpha\kappa\upsilon\epsilon\iota\nu$ (akuein = to hear). The importance of the acoustics for design of buildings has been visible since the theaters, as *Epidauros* in the antique Greece. There is general agreement in the literature that Pythagoras (about 570-497 BC) was the first who studied acoustics. Later on in the history the Roman architects show a great understanding of the importance of the acoustics for designing of amphitheatres in order to obtain a good acoustics, which among others is documented by the Roman architect Vitruvius in his book *De Architectura*. In this book he talks about some of the acoustics parameters, which characterize the beginning and dying of the sound. In the latest century these parameters have been used as the most important design parameters.
Nowadays acoustics is an integrated part at the projecting of new building on a par with static, plumbing, electricity and geotechnical engineering. Acoustics includes among others

- building acoustics (sound insulation)
- architectural acoustics (sound control)

Sound insulation characterizes the arrangements which have the purpose to reduce the sound transmission from one room to another and from external sound sources, see figure 1–1. *Sound control* characterizes the arrangements which are made to regulate or deadened that sound which is in the same room as the sounding body. Typically, sound regulation is related to the acoustics quality in a room, eg. concert rooms, study rooms, conference rooms etc. At the decision whether a sound impression is understood as sound or *noise*, it is decisive whether the sound is wanted or unwanted. Whether a sound is wanted or unwanted might depend on the persons who are exposed to the sound. Probably, a racing driver thinks that the sound from a motor at full blast is music to his ears, whereas a neighbor to a racing course might think that the sound from a motor is disturbing. Necessarily, a sound needs not to be high to be disturbing. In certain situations the sound from a dripping tap can be just as much disturbing as a loud noising motor. The work situation also depends on such an estimate just like psychic facts like eg. pressure of time, high concentration and several simultaneous tasks do it. Sound that prevents or makes it difficult to give the information whether it is talk- or sound signals, can be called noise and demands that some kind of silencer is made. The noise in a room can either come from an external source or an internal source. Damping of external noise is a building acoustic problem (sound isolation), while damping of internal noise is an architectural acoustic problem (sound regulation).

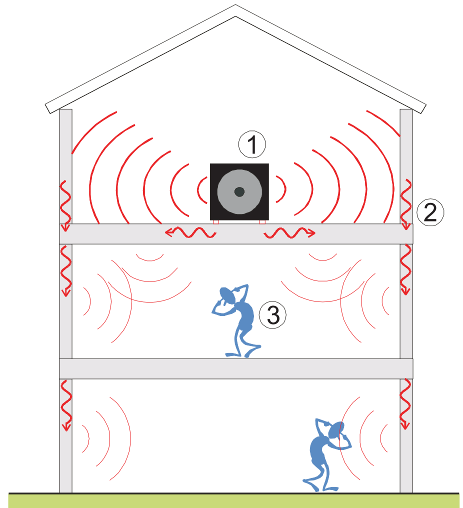

Fig. 1–1 The dispersion of sound from source (1) in and through a building's walls and floors (2) to the person receiving (3).

Sound isolation, sound regulation and noise abatement are more and more asked for as a necessary part of modern building, where a good acoustic quality is wanted. Generally, it is difficult to define ”good acoustics”, because it is a manifestation of to what extent the acoustics in a room is adapted to the room in a ”good” or ”bad” way. However, it is obvious that you do not want the same acoustics in e.g. a concert room as in a study room. Besides, evaluation also depends on a room having a ”good acoustics” and how the acoustics in a room is evaluated. To determine the acoustics in a room an acoustician will use measured objective acoustical measurements. On the contrary a musician will use not measured subjective measurements like the sound's warmth, fullness, colour, clarity etc. With that ”good acoustics” must be defined as a combination of as well objective and subjective measurements which can be divided into measures that are related to the distribution of the sound, the dispersion in the room and measures that are related to the noise level in the room.

This book has the purpose of giving an introduction to the building and room acoustic ideas, so that the student will be introduced to the theories and methods within practically applicable acoustic planning of buildings, so that a satisfactory acoustic environment is obtained. The contents of the book are organized as follows

- Chapter 1: Introduction
- Chapter 2: Definitions in Linear Acoustics
- Chapter 3: Architectural Acoustics - Sound Regulations in Buildings
- Chapter 4: Sound Isolation
- Chapter 5: Principles for Architectural Acoustics Design
- Chapter 6: Sound Measurements in Buildings

CHAPTER 2
Definitions in Linear Acoustics

Sound is supposed to mean audible vibrations in a firm elastic, liquid or gaseous medium. Sound dispersion in air is happening when a sound source through a mechanical influence starts the particles of air to vibrate round their equilibrium. As a result of the elasticity of the air this kind of vibration disperses to more distant particles of air, and in this way a progressive sound wave arises. When the particles move in the air a varying overpressure and underpressure appear - *sound pressure*, which is converted to sound impressions in the human ear. The vibrations in the air are called *air sound*, while vibrations in for example buildings are called *building sound* or *structural sound*.

When the particles of air are moving a varying overpressure and underpressure appear round the static air pressure p_s (barometric pressure), which will appear in absence of the sound waves, that is the air pressure in the undisturbed medium. The difference between the momentary total air pressure $p_t(t)$ and the static air pressure p_s gives *the sound pressure* $p(t) = p_t(t) - p_s$, is a time variable size which is stated in the unit Pascal (abbreviated Pa), see figure 2–1. As the sound pressure varies between positive and negative values the *effective sound pressure* $\tilde{p}$, is often used as sound pressure and is then stated as a ”Root mean square”-value of the time interval $t_2 - t_1$

$$\tilde{p} = \sqrt{\frac{1}{(t_2 - t_1)} \int_{t_1}^{t_2} p^2(t)dt} \tag{2–1}$$

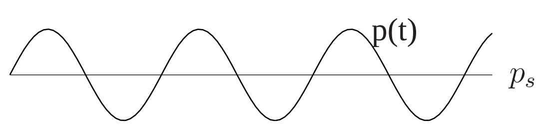

Fig. 2–1 Time variable sound pressure, $p(t) = p_t(t) - p_s$.

Note that in gases and liquids exist longitudinal waves (blast waves) where the propagation direction is parallel to the swing direction of the molecules. Contrary to this solid substances can transmit shear stresses, and in these substances there are also the so-called transversal waves, where the propagation direction is perpendicular to the swing direction of the molecules. Sound waves can be compared to waves in water in the same way as you for example throw a stone in some stagnant water. However, sound waves move faster than waves in water. The sound waves are just like the water waves that the wave propagation happens by an energy exchange and not as a transport of air or water.

The velocity of the propagation c for sound waves can be determined on the assumption of an adiabatic process, see appendix A

$$c = \sqrt{B/\rho_s} = \sqrt{\frac{\gamma R T_K}{M}} \tag{2–2}$$

where B is the compression modulus (the bulk modulus) and ρ_s is the air static density. T_K is the air temperature in Kelvin, R is the gas constant, M is the mole mass, and γ is the ratio of specific warm capacities at a constant pressure and a constant volume, respectively. From (2–2)it is seen that *velocity of sound* c is independent of the pressure, but $c \propto \sqrt{T_K}$ as R and γ are constants. By insertion of these constants (2–2) can be written as follows

$$c \approx 331,4 + 0,607 T_C \tag{2–3}$$

where T_C is the temperature in ^oC. (2–3) is an approximate equation for velocities of sound which apply to T_C about the room temperature.
The velocity of sound in different materials and a room temperature at 20 ^oC is seen in table 2-1

Material	Velocity of sound (m/s)
air	343
lead	1320
water	1500
concrete	3050
wood (soft)	3320
glass	3650
wood (hard)	4250
iron	4720
steel	5000

Table 2-1 Velocities of sound in different materials.

2.1 The Physical Description of Sound

The propagation of sound makes a *sound field*, which is often considered as being one of the following two types

- Plane sound field
- Spherical (ball-shaped) sound field

when the sound is not reflected from bounded surfaces. In chapter 3 a *diffused sound field* , which is a result of a combination of direct and reflected sound will be discussed. The expressions which in the following are stated in relation to plane and spherical waves, which disperse

unhindered, have a great practical importance. Among other things, this is due to more complicated fields of sound which can be described as a superposition of plane and/or spherical waves.
Within the audible area the propagation of sound in air with a good approximation can be described as a linear partial differential equation called, *the wave equation*

$$\nabla p = \frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{2–4}$$

where $p(x, y, z, t)$ is the sound pressure in a Cartesian (x, y, z)-system of coordinates. Similar identical wave equations could be shown for movements of particle of air $\xi(x, y, z, t)$, velocities of particle of air $u(x, y, z, t)$, air temperature $T_C(x, y, z, t)$ or the air density $\rho(x, y, z, t)$. However, the wave equation (2–4) is the most used, as the sound pressure is the most easy acoustical size to measure. Generally, the ruling equations from which the wave equation is derived are the most complicated, but on the assumption of small changes in the sound pressure great simplifications can be made, which cause that you can work linearly, see appendix A. Small changes in the sound pressure corresponding to that the sound does not give feeling of pain in the ear. When this assumption is not fulfilled it means that you have to work non-linearly, then the velocity of the sound is stated in (2–2) dependent on the pressure.

2.1.1 Plane Sound Field

A plane progressive wave is the simply possible solution to the wave equation (2–4), but has a great relevance after all, e.g. at a calculation of noise in ventilating systems. In this case the sound waves disperse in the direction of the ventilating shaft when the particles of air are swinging around their equilibrium in the same direction as the sound wave's propagation direction. It gives by turns a density and a dilution of the particles of air in the air, which results in a variation of the momentary sound pressure with the place, see figure 2–2.

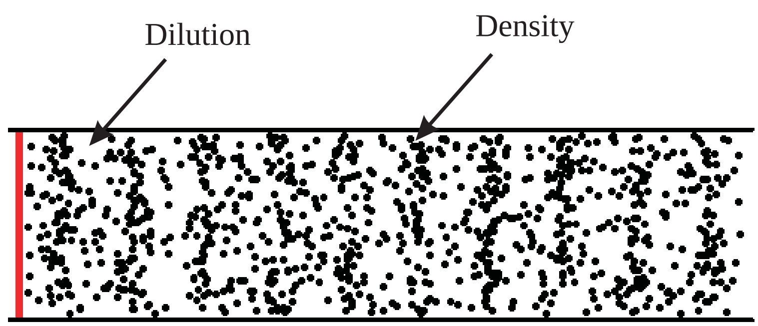

Fig. 2–2 Plane sound field in a pipe.

In this case the wave equation is stated as follows

$$\frac{\partial^2 p}{\partial x^2} = \frac{1}{c^2}\frac{\partial p}{\partial t^2} \tag{2–5}$$

where x is the coordinate in the plane propagation of the sound wave. (2–5) has the general solution to the sound pressure $p(x, t)$

$$p(x, t) = f(ct - x) + g(ct + x) \tag{2–6}$$

which can be seen by inserting (2–6) in (2–5).
The functions $f(ct-x)$ and $g(ct+x)$ are arbitrary functions, which give a general solution, as you in principle can state all kinds of plane sound waves by these two functions. $f(ct-x)$ represents a plane wave which disperses in the x-direction in a positive direction (a progressive wave) with the velocity c. $g(ct+x)$ represents a plane wave which spreads in the x-directions in a negative direction (a reflected wave), also with the velocity c. Regarding a one-dimensional sound-dispersive problem it is seen that the sound waves disperse with a constant velocity c, which is not frequency dependent, for which reason these sound waves are called *(*non-dispersive) waves. propagation of the arbitrary functions $f(ct-x)$ and $g(ct+x)$ is illustrated in figure 2–3

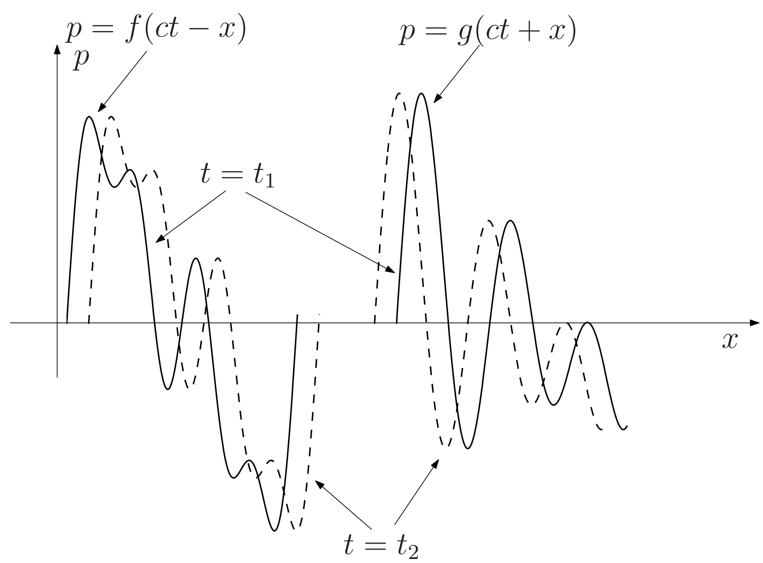

Fig. 2–3 Plane arbitrary wave.

Sound waves can either be transient (decreasing) or stationary where the stationary harmonic sound waves are of special interest in linear acoustics. Regarding an acoustic system you can for example let a loudspeaker affect a room at a certain *frequency* f. After this the sound pressure will everywhere vary in the room (but a different phase and amplitude according to where you are in the room). For a linear system the super position principle is also valid, which says that if the affection is compounded of several frequencies, every single frequency can then be applied and subsequently sum up the responses to find the total sound pressure. Therefore, it is convenient to consider a *harmonic sound field* , that is a sound field where the dependence of place and time of the sound pressure $p(x,t)$ can be written as follows

$$p(x,t) = \hat{p}_1 \sin(\omega t - kx + \phi_1) + \hat{p}_2 \sin(\omega t + kx + \phi_2) \qquad (2–7)$$

where $\hat{p}_1$ and $\hat{p}_2$ are the amplitudes of the sound pressure of waves, which disperse in the positive x-direction and the negative direction, respectively. ϕ_1 and ϕ_2 are the respective phases.
Given a position x a harmonic signal appears again every time the time is increased with a *period* T, which means that

$$\omega T = 2\pi \Rightarrow \omega = \frac{2\pi}{T} = 2\pi f \qquad (2–8)$$

where ω is *the angular frequency*. ϕ_1 and ϕ_2 are phase angles and k *the wave number*, which is related to the angular frequency ω by

$$k = \frac{\omega}{c} \tag{2–9}$$

Given a time t the sound pressure (2–7) appears again harmonically in the position coordinate x, which means that a repetition happens every time kx increases with 2π. Increase in x is called *the wave length* λ. Therefore, it can be applied that

$$k = \frac{2\pi}{\lambda} \tag{2–10}$$

and with this

$$\frac{\omega}{c} = \frac{2\pi f}{c} = \frac{2\pi}{\lambda} \Rightarrow \lambda = \frac{c}{f} \tag{2–11}$$

Figure 2–4 illustrates how the sound pressure changes when a plane harmonic sound wave disperses with the wave length λ

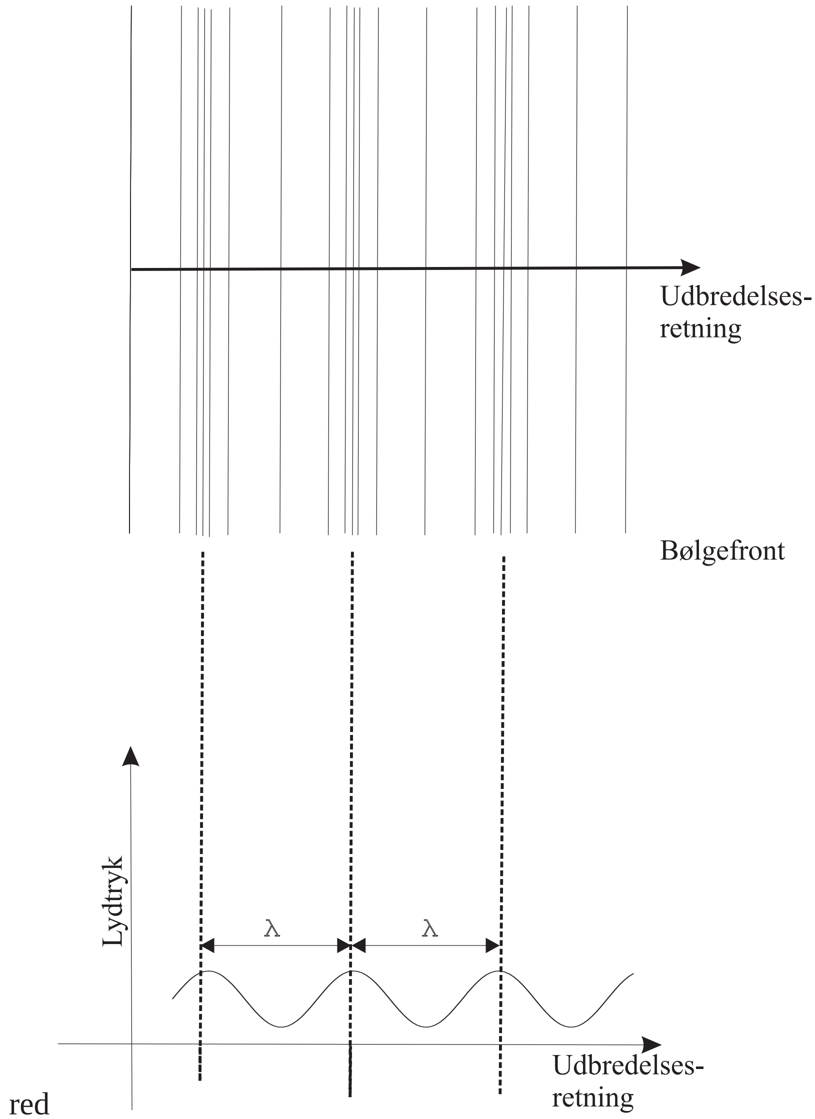

Fig. 2–4 Plane harmonic sound field.

The sound pressure (2–7) is now written with a complex plane wave representation ($i = \sqrt{-1}$)

$$p(x,t) = \hat{p}_1 \mathrm{e}^{i(\omega t - kx)} + \hat{p}_2 \mathrm{e}^{i(\omega\ t + kx)} \tag{2–12}$$

as integration and differentiation regarding time are specially simple for harmonic fields in complex shape.
The wave equation (2–4) for the *x*-direction is written as follows

$$\rho_s \frac{\partial u}{\partial t} + \frac{\partial p}{\partial x} = 0 \tag{2–13}$$

By a combination of (2–12) and (2–13) you will find a simple connection between the sound pressure and the particle velocity

$$u(x,t) = -\frac{1}{\rho_s} \int \frac{\partial p}{\partial x} dt = -\frac{1}{\rho_s} \left[\frac{-ik}{i\omega} \hat{p}_1 \mathrm{e}^{i(\omega t - kx)} + \frac{ik}{i\omega} \hat{p}_2 \mathrm{e}^{i(\omega\ t + kx)} \right] \tag{2–14}$$

which can be written as $k/\omega = 1/c$

$$u(x,t) = \frac{1}{\rho_s c}\hat{p}_1 \mathrm{e}^{i(\omega t - kx)} - \frac{1}{\rho_s c}\hat{p}_2 \mathrm{e}^{i(\omega t + kx)} \tag{2–15}$$

By a comparison of (2–12) and (2–15) it is seen that regarding a plane wave the sound pressure and the particle velocity are in phase, and the ratio of the sound pressure and the particle velocity at a point in the x-direction positive direction is related to the sound pressure as follows

$$\frac{p}{u} = \rho_s c = Z_c \tag{2–16}$$

and in the x-direction negative direction

$$\frac{p}{u} = -\rho_s c = Z_c \tag{2–17}$$

Generally, the amplitudes $\hat{p}_1$ and $\hat{p}_2$ are complex and give a surface a complex ratio between the sound pressure and the particle velocity called *specific acoustic impedance*. Regarding a plane progressive wave the ratio in a point between the sound pressure and the particle velocity is called *charactistic impedance*. Z_c is real as $\hat{p}_1$ and $\hat{p}_2$ are real. Regarding air at 20 ^oC the characteristic impedance is $Z_c = 415$ Pa s/m.

2.1.2 Spherical sound field

A *spherical sound field* arises when a punctate sound source sends out sound energy equally in all directions and is characterized as a free sound field *free sound field*.

This kind of wave propagation can be compared with the wave propagation which arises on the water level when a stone is throwing into the water. The wave propagation is the same in all directions that is to say that the vibration condition is the same in all the points which are at the same distance from the sounding body. A surface compound of points with the same vibration condition is called a *wave front*, which to a spherical sound field makes the concentric spherical surfaces and the centre in the point of the sound source, see figure 2–5. When the wave front moves away from the source, the curve of the wave front becomes smaller and a rectilinear wave front arises, and the sound waves can be regarded almost as plane waves.

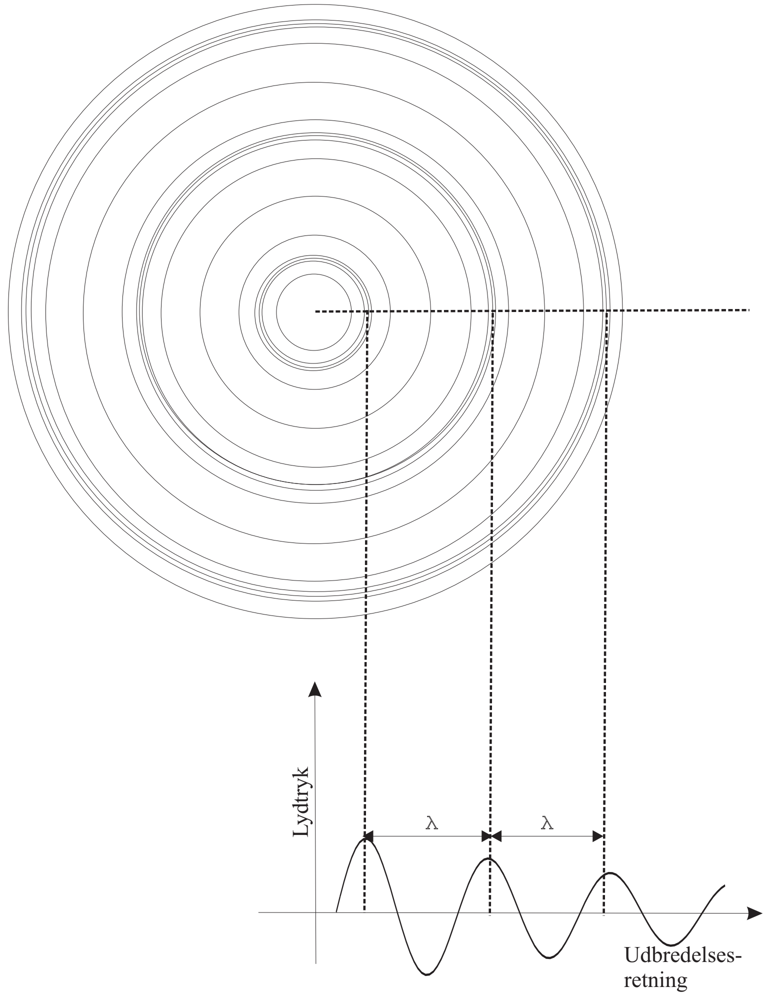

Fig. 2–5 Spherical sound field.

Regarding spherical fields of sound it is advantageous to rewrite the wave equation (2–4) to a spherical system of coordinates (r, θ, φ). If a radial field is considered that is a sound field where the vibration condition is the same all over in an arbitrary distance from the centre of the source, then the wave equation of the sound pressure becomes $p(r, t)$ as there is no dependence of the angles θ and φ

$$\frac{\partial^2(rp)}{\partial r^2} = \frac{1}{c^2}\frac{\partial^2(rp)}{\partial t^2} \tag{2–18}$$

where r is the distance from the centre of the source. (2–5) and (2–18) are identical, if p is changed to rp and x to r. Therefore, it is easy to see that the general solution of the sound pressure $p(r, t)$ becomes

$$p(r, t) = \frac{1}{r}f(ct - r) + \frac{1}{r}g(ct + r) \tag{2–19}$$

The functions $f(ct - r)$ and $g(ct + r)$ are arbitrary functions which give a general solution as you in principle can state all kind of spherical sound waves at these two functions. $f(ct - r)$ represents a spherical wave which disperses in the r-direction in a positive direction (progressive wave) and a velocity c. $g(ct + r)$ represents a spherical wave which disperses in the r-direction in a negative direction (reflected wave) and the velocity c, too. It is seen that $p(r, t) \propto 1/r$. This means that the sound pressure goes against the infinite, when the pressure goes against zero, which is due to a mathematical singularity in the solution and not because of a physical reality. In reality $r = 0$ corresponds to be in the sounding body. A harmonic solution to the spherical sound field is written as follows

$$p(r,t) = \frac{\hat{p}_1}{r}\sin(\omega t - kr + \phi_1) + \frac{\hat{p}_2}{r}\sin(\omega t + kr + \phi_2) \tag{2–20}$$

where $\hat{p}_1$ and $\hat{p}_2$ are the amplitudes of the sound pressure for waves that disperse in the r-direction in a positive and negative direction, respectively. ϕ_1 and ϕ_2 are the phases, respectively.
The sound pressure (2–20) is written in the same way as the plane wave (2–12)as a complex representation ($i = \sqrt{-1}$)

$$p(r,t) = \frac{\hat{p}_1}{r}\mathrm{e}^{i(\omega t - kr)} + \frac{\hat{p}_2}{r}\mathrm{e}^{i(\omega t + kr)} \tag{2–21}$$

The governing equation (2–18)of the r-direction becomes

$$\rho_s \frac{\partial u}{\partial t} + \frac{\partial p}{\partial r} = 0 \tag{2–22}$$

By combining (2–21) and (2–22) the following simple connection between the sound pressure and the particle velocity in spherical coordinates is found

$$u(r,t) = -\frac{1}{\rho_s}\int \frac{\partial p}{\partial r} dt = -\frac{\hat{p}_1}{\rho_s}\int \left(\frac{-1}{r^2} + \frac{-ik}{r}\right)\mathrm{e}^{i(\omega t - kxr)} dt \tag{2–23}$$

as it is only the progressive wave to be considered. In most problems from the wave propagation the reflected wave can be ignored. Now, (2–23) can be written as $k = \omega/c$

$$u(r,t) = -\frac{\hat{p}_1}{\rho_s c r}\left(1 + \frac{1}{ikr}\right)\mathrm{e}^{i(\omega t - kr)} \tag{2–24}$$

If $kr = 2\pi r/\lambda \gg 1$, it is seen from (2–24) that $1/(ikr) \ll 1$ and the sound pressure and the particle velocity will be approximated in phase. With this the ratio between the sound pressure and the particle velocity is in one point in the r-direction in positive directions

$$\frac{p}{u} \approx \rho_s c \tag{2–25}$$

which means that a spherical sound field is acting like an approximated plane sound field. It is said that the sound source in the distant field ($kr = 2\pi r/\lambda \gg 1$) acts locally as a plane wave. On the other hand, the sound field does not act globally as a plane field of wave as the sound pressure falls with the distance r. In the near field close to the source that is $kr = 2\pi r/\lambda \ll 1$ the following is stated

$$\frac{p}{u} \approx i\rho_s ckr \tag{2–26}$$

This means that the sound pressure and the particle velocity is 90^o phase shifted.

2.2 Sound Energy, Sound Intensity and Sound Effect

By solution of acoustic problems the total energy in the sound field is of no interest. The following three sizes have interest

- sound effect
- energy density
- sound intensity

as they state sound energy per unit of volume, sound energy per unit of time and the sound energy transportation through a unit of area, respectively.

2.2.1 Energy Density

The energy E in a sound field consists of *kinetic energy* E_k and *potential energy* E_p, respectively . Then the energy density (the energy per unit of volume) ε can be written

$$\varepsilon = \frac{E}{V} = \frac{E_k}{V} + \frac{E_p}{V} = \varepsilon_k + \varepsilon_p \tag{2–27}$$

where V is the volume. ε_k and ε_p are the kinetic and the potential energy density, respectively. The kinetic energy of a plane wave that disperses in the *x*-direction in a positive direction is

$$E_k(x,t) = \frac{1}{2}\rho_s V u^2 = \frac{1}{2}\frac{V_s p^2}{\rho_s c^2} \tag{2–28}$$

and correspondingly the potential energy is given by

$$E_p(x,t) = -\int p dV = \int \frac{V_s}{\rho_s c^2} dp = \frac{1}{2}\frac{V_s p^2}{\rho_s c^2} \tag{2–29}$$

as it is applied to a plane wave as follows, see figure 2–6

$$\begin{aligned} V &= V_s(1 + \frac{\partial s}{\partial x}) \\ \frac{\partial s}{\partial x} &= \frac{-p}{B} = \frac{-p}{\rho_s c^2} \\ dV &= \frac{-V_s}{\rho c^2} dp \end{aligned} \tag{2–30}$$

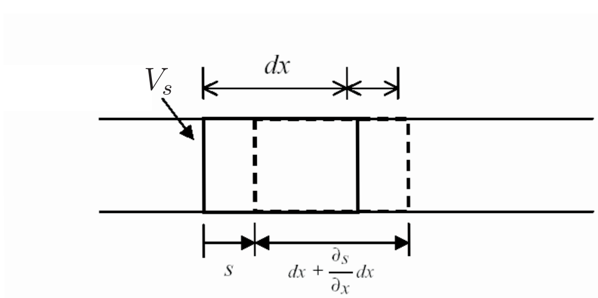

Fig. 2–6 Volume change in a plane sound field

Thus, the energy density of a plane wave becomes

$$\varepsilon = \frac{\tilde{p}^2}{\rho_s c^2} \tag{2–31}$$

as a time average value of sound pressure is used. Generally, the definition of energy density (2–27) is based on instantaneous values, but similar to common practice by other time variant signals the time average value is used.
In a free spherical sound field the energy density in the distance r from a point source with the effect P becomes

$$\varepsilon = \frac{P}{4\pi r^2 c} \tag{2–32}$$

2.2.2 Sound Intensity and Sound Effect

The acoustic intensity, I is defined by the average value of the acoustic effect P, which passes through a area element which is perpendicular to the disperse direction. Generally, the intensity is a vector size, as sound disperses in three dimensions. Only a one-dimensional sound propagation will be considered in the following.
The definition of the intensity I in a point of a plane wave that disperses in the x-direction in a positive direction is as follows

$$I(x) = \frac{1}{(t_2 - t_1)} \int_{t_1}^{t_2} p(x,t)u(x,t)dt \tag{2–33}$$

Insertion of (2–12) in (2–33) you get the intensity of the plane wave, which disperses in the x-direction in a positive direction

$$I(x) = \frac{\tilde{p}^2}{\rho c} = \frac{\tilde{p}^2}{Z_c} = \varepsilon c \tag{2–34}$$

as $p(x,t)$ and $u(x,t)$ are in a phase of a plane wave. From (2–34) it is seen that the sound intensity can be considered as propagation of energy density ε and the sound velocity c.
However, a harmonic plane wave becomes (2–34)

$$I(x) = \frac{\hat{p}_1^2}{2\rho c} \tag{2–35}$$

as $\tilde{p} = \hat{p}_1/\sqrt{2}$ for a harmonic wave.
As for a spherical wave in a distant field the intensity I becomes

$$I(r) = \frac{1}{(t_2 - t_1)} \int_{t_1}^{t_2} p(r,t)u(r,t)dt \approx \frac{\tilde{p}^2}{\rho c} = \frac{\tilde{p}^2}{Z_c} \tag{2–36}$$

as it is generally applied that a progressive spherical wave and a plane sound wave are identical.
In the close field the intensity of a spherical wave cannot be stated in the same easy way as in the distant field, which is omitted here.
The statement above shows that the intensity is related to the effective sound pressure, which can be measured with a microphone, as mentioned in chapter 6.

A simple expression for the intensity from a point source with an effect P is also applied to spherical waves in the distant field as the surface area of a ball is $4\pi r^2$

$$I(r) = \frac{P}{4\pi r^2} \tag{2–37}$$

Thus, the intensity decreases with the distance in the second power for spherical fields ($I \propto 1/r^2$). This means that the ratio between the intensities in two points with the distances r_1 and r_2 from the point source, respectively becomes

$$\frac{I_1}{I_2} = \frac{\dfrac{P}{4\pi r_1^2}}{\dfrac{P}{4\pi r_2^2}} = \frac{r_2^2}{r_1^2} \tag{2–38}$$

This connection is called *inverse square law* .

From (2–37) it is seen that the effect of a spherical source in a free field becomes

$$P = 4\pi r^2 I = 4\pi r^2 \frac{\tilde{p}^2}{\rho_s c} \tag{2–39}$$

As the effective sound pressure $\tilde{p}$ of a spherical field is in inverse ratio to the distance, see (2–19), the expression of the effect is independent of the distance r that is that in idealized circumstances no energy is left in the air (loss free medium).

Generally, the effect can be defined by

$$P = \int_S I dS \tag{2–40}$$

where S is a given surface through which the sound energy is transmitted.

2.3 Spectrum and Octave Band

A harmonic sound signal can be described in the time area and the frequency area, respectively see figure 2–7. The figure shows a time signal with a frequency of 10 Hz, which is illustrated in the frequency range with an amplitude of 10 Hz.

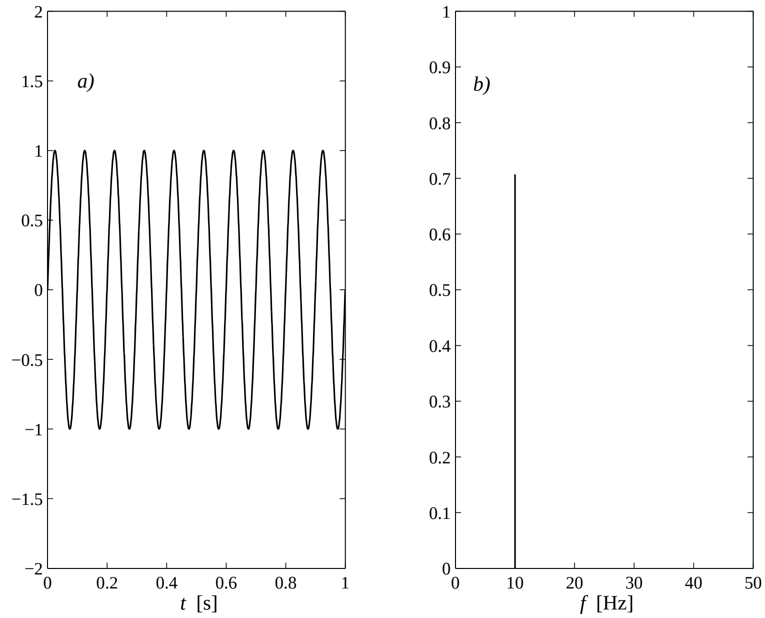

Fig. 2–7 Harmonic signal in a) the time area and b) the frequency range.

Generally, a given periodical sound signal, eg. a sound pressure as

$$p(t) = p(t + \mathrm{i}T) \quad i = 1, 2, 3, ... \tag{2–41}$$

can be written by a Fourier series which is the sum of harmonic components

$$p(t) = \sum_{i=1}^{\infty} \hat{p}_i \cos(\omega_i t + \phi_i) \tag{2–42}$$

$\hat{p}_i$ and ω_i are the amplitude and the angel frequency of the simply harmonic components in the signals with the phases ϕ_i. It is rare and almost only in connection with acoustic tests that you have sound sources which solely consist of a clear sound that is one harmonic component. As an example sound provided from an instrument does not consist of one single harmonic component, but is compounded of a number of simultaneously common harmonic vibrations. It is said that there is a *keynote* and a number of *overtones* which have a frequency which is a multiples of the frequency of the keynote. In the same way the human speech is also an example of a sound source which is compounded of many frequencies.
Figure (2–8) shows how a given periodical signal is illustrated from the time area and over into the frequency area. The description of the signal in the frequency domain is called a frequency spectrum (also called *power*- or *amplitude spectrum*), where the single ”columns” are called spectrum lines which each represents the amplitudes in the simply harmonic components in a Fourier series (2–42). A spectrum gives in this way information about how the energy in a sound signal is distributed at different frequencies and is calculated practically at a so-called FFT (Fast Fourier Transform) analysis. Sometimes it is chosen to make a visual representation (*Spectrogram*) which combines the information from the time area and the frequency area by making FFT analysis on parts of the time signal. With this you get a spectrum for each time interval which can be used to show how the frequency content in a signal varies as a function of the time.

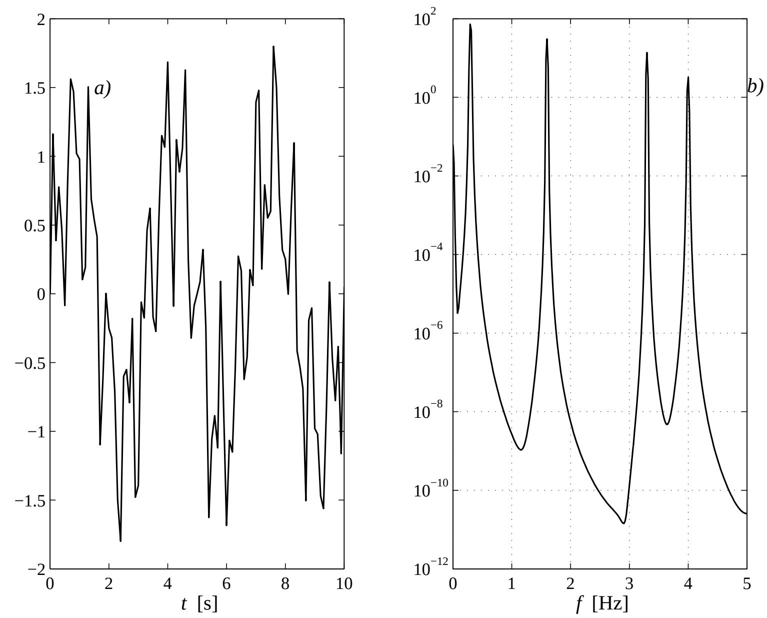

Fig. 2–8 Periodical signal in a) the time area and b) the frequency area.

By measuring sound in air a logarithmic scale of the frequency axis is often used. The logarithmic scale of frequency is in reasonable harmony with the subjective perception of the sound of a human. Similar to the art area the frequency areas are divided into octaves, where the width of the octaves are constant on the logarithmic axis. Within every single frequency area (*octave band*) , the energy of a signal can be defined and represented. An octave is formed of two frequencies with the ratio $1 : 2$ and is based on a reference frequency of 1000 Hz. However, the division into 1/1-octaves is characterized by the upper limiting frequency $f_ø$ is the double of the lower f_n. In this way the octave band with the centre frequency f_0=1000 Hz stretches from f_n= 707 Hz to $f_ø$ = 1414 Hz. This interval $\Delta = f_ø - f_n$ is called the band width. Generally, the intervals of the 1/1-octaves are stated by

$$\begin{aligned} f_0 &= \sqrt{f_n f_ø} \\ f_n &= f_0/\sqrt{2} \\ f_ø &= \sqrt{2} f_0 \\ \Delta f &= \left(\sqrt{2} - 1/\sqrt{2}\right) f_0 \end{aligned} \qquad (2–43)$$

A finer division into 1/3-octaves gives a band about 1000 Hz which stretches from 891 Hz to 1122 Hz. Generally, these 1/3-octaves give as follows

$$\begin{aligned} f_0 &= \sqrt{f_n f_ø} \\ f_n &= f_0/\sqrt[6]{2} \\ f_ø &= \sqrt[6]{2} f_0 \\ \Delta f &= \left(\sqrt[6]{2} - 1//\sqrt[6]{2}\right) f_0 \end{aligned} \qquad (2–44)$$

20	25	**31,5**
40	50	**63**
80	100	**125**
160	200	**250**
315	400	**500**
630	800	**1000**
1250	1600	**2000**
2500	3150	**4000**
5000	6300	**8000**
10000	12500	**16000**

Table 2-2 Standardized centre frequencies 1/3 and 1/1-octaves. (1/1-octaves is shown in boldface).

Table 2-2 shows standardized centre frequencies of 1/1- and 1/3-octave bands.
In building acoustics there is normally worked with 16 standard frequencies with 1/3-octave intervals with the centre frequencies 100, 125, 160, 200, ...3150 Hz, while there are normally used 6 standard frequencies with 1/1-octave intervals with the centre frequencies 125, 250, 500, 1000, 2000, 4000 Hz in the architectural acoustics. As to calculations with noise from a ventilating plant it might also be relevant to look at the octave bands 63 Hz and 31,5 Hz.

2.4 Decibel and Sound Levels

The human's perception of sound acts in such a way that a number of doubling of the sound pressure are considered as a number of equal heavy changes of the sound intensity. Therefore, a logarithmic measurement stating the sound intensity is more useful than a linear measurement. Sound intensities are stated in dB (decibel) in honour of Alexander Graham Bell. Decibel is a logarithmic relative unit of measurement of sound that corresponds to 10 times the logarithm to a sound pressure or a sound effect in the ratio of a reference value of sound pressure, sound intensity, or sound effect. These reference values correspond to the lowest level of a ear from a young human being normally can perceive at 1000 Hz. A unique statement of a strength from a sound source also complies a description of measuring conditions which will be mentioned in chapter 6 about sound ranging.
Thus, a strength from a sound source can be described by a *level of a sound pressure* L_p

$$L_p = 10 \log \left(\frac{\tilde{p}^2}{p_0^2} \right) = 20 \log \left(\frac{\tilde{p}}{p_0} \right) \tag{2–45}$$

where $\tilde{p}$ is the effective sound pressure and $p_0 = 20$ μPa is an international standardized reference sound pressure. Using the logarithmic dB scale it is seen that a sound pressure from 20 μPa to 200 Pa will be reduced to the area from 0 to 140 dB which in practice is operational to work with.
Figure 2–9 shows examples of different kinds of sound pressure levels which appear everyday.

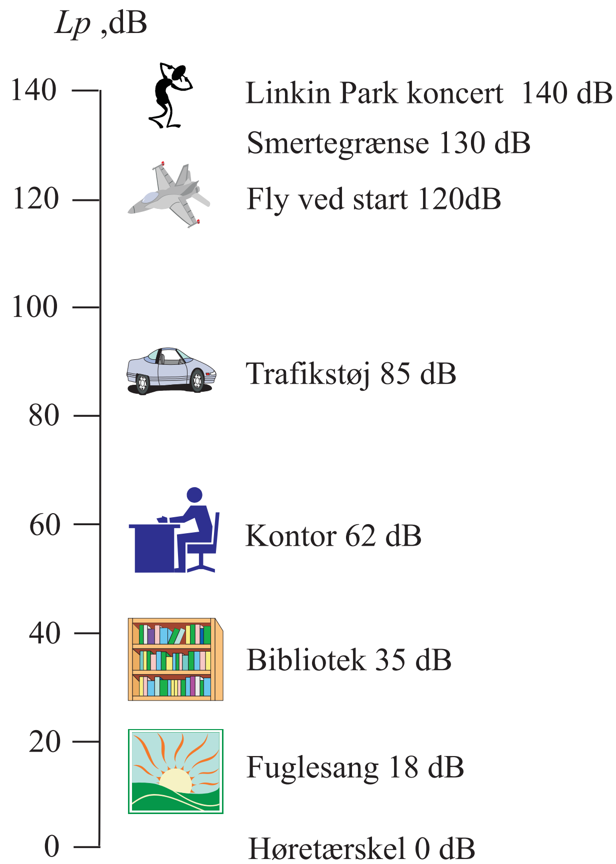

Fig. 2–9 Typical sound pressure levels L_p.

A *sound effect level*, L_w in W is stated by

$$L_w = 10\log\left(\frac{P}{P_0}\right) \tag{2–46}$$

where P is the sound effect and $P_0 = 1$ pW is the reference sound effect.
The sound intensity level L_I is stated as

$$L_I = 10\log\left(\frac{I}{I_0}\right) \tag{2–47}$$

where $I_0 = 1$ pW/m^2 is the reference value of the sound intensity level.
As the sound pressure is easier to measure than the sound intensity it can be advantageous to express L_I by L_p. Using (2–34) for a plane sound field and a spherical sound field in the distant field L_I can be defined by

$$L_I = 10\log\left(\frac{\tilde{p}^2}{\rho_s c I_0}\right) = 10\log\left(\frac{\tilde{p}^2}{p_0^2}\frac{p_0^2}{\rho_s c I_0}\right) = L_p + 10\log\left(\frac{400}{\rho_s c}\right) = L_p - 0,2\text{dB} \approx L_p \tag{2–48}$$

From this it is seen that L_I and L_p are almost equally big in a plane sound field and a spherical sound field in the distant field.

Example 2.1: Intensity and Distance to the Sound Source

A point sound source has the sound effect $P = 200$ W. How big is the sound pressure level L_p which two persons are exposed to placed 2m and 4m from the sounding body, respectively?

As the sound is propagated in the air as a ball the sound intensity will decrease as a function of the distance from the sound source which is the centre of the "sound ball", and it can be said that the sound distributes on a ball surface area. The surface area of a ball is stated as $A = 4\pi r^2$ which causes the sound intensity I to become

$$I = \frac{P}{A} = \frac{200}{4\pi 2^2} \approx 4\mathrm{W/m^2} \tag{2–49}$$

The sound pressure level which the person 2m from the sound source is exposed to becomes

$$L_p \approx 10\log\left(\frac{I}{I_0}\right) = 10\log\left(\frac{4}{10^{-12}}\right) = 126\mathrm{dB} \tag{2–50}$$

If the person was placed in a distance of 4m from the sounding body the sound pressure level becomes

$$L_p \approx 10\log\left(\frac{I}{I_0}\right) = 10\log\left(\frac{1}{10^{-12}}\right) = 120\mathrm{dB} \tag{2–51}$$

The above example shows that in a spherical sound field the sound pressure level has a decrease of 6 dB by doubling the distance, which is the distance law (2–38) for a free spherical sound field expressed in dB. It is said that there is a distant damping of 6 dB per doubling or halving of the distance. It has to be noted that a plane sound field does not have a distant damping. Connection between the sound pressure level L_p and the sound effect level L_w in a free spherical sound field is also defined by

$$L_p = L_w - 10\log\left(4\pi r^2\right) = L_w - 20\log(r) - 11 \tag{2–52}$$

2.4.1 Several Sound Sources

If the sound pressure from N simultaneous sources in a point is to be defined, the total of the sound effect P_{tot} is the sum of the single effect of the sources P_i

$$P_{tot} = \sum_{i=1}^{N} P_i \tag{2–53}$$

On the other hand it is essential to note that the sound levels from several sources cannot simply be added. It is necessary to calculate back to squared sizes like sound intensity, sound effect or the squared sound pressure. After this you can add and subsequently calculate back to the relevant sound level. It is important to distinguish between two kinds of sound sources

- uncorrelated
- correlated

If the resulting sound pressure consists of sound from several independent sound sources you can normally anticipate that the sources are *uncorrelated sound sources* , i.e. the sound from the single sound sources is of a different frequency. Machines in a production room are an example of uncorrelated sound sources. In this case the total effective sound pressure $\tilde{p}_{tot}$ defined by

$$\tilde{p}_{tot}^2 = \sum_{i=1}^{N} \tilde{p}_i^2 \tag{2–54}$$

where $\tilde{p}_i^2$ is the sound pressure of the single of the N sound sources.
According to *correlated sound sources* it is applied that the phase relations between the single signals have to be considered. Two loudspeakers that send the same signal are examples of two correlated sound sources. The resulting sound pressure is obtained by adding the sound pressures from the single sources as their mutual phase conditions have to be considered.
(2–54) shows that you can add uncorrelated sound pressure levels which are also applied to uncorrelated intensities and effects. The following example shows the result of an addition of two equal strong sound sources. The result shows that if you add two equal strong sound levels it corresponds to add 3 dB to the single sound pressure level. An increase of 6 dB corresponds to a doubling of the sound pressure.

Example 2.2: Addition of Two Sound Sources
Two engines are going full speed. The sound intensity level L_I 5m from each engine is 90 dB. What is the sound pressure level L_p for a person who is placed 5m from each of the engines?

$$L_p \approx L_I = 10\log\left(\frac{I+I}{I_0}\right) = 10\log\left(\frac{I}{I_0}\right) + 10\log(2) = 90\text{ dB} + 3\text{ dB} = 93\text{ dB} \tag{2–55}$$

General rules for addition, subtraction and multiplication of sound levels can be made. From the definition of sound pressure (2–45) you can find

$$\tilde{p}^2 = p_0^2 10^{L_p/10} \tag{2–56}$$

By combining (2–45), (2–56) and 2–54 the following addition rule for N uncorrelated sound pressure levels can be obtained.

$$L_{p_{tot}} = 10\log\left(\sum_{i=1}^{N} 10^{L_{p_i}/10}\right) \tag{2–57}$$

(2–57) is also applied for addition of sound intensity level and sound effect level.
The average value of the sound levels from a number of measuring results can be calculated by using the formula

$$L_{p_{tot}} = 10\log\left(\frac{1}{N}\sum_{i=1}^{N} 10^{L_{p_i}/10}\right) \tag{2–58}$$

If the difference between the highest and the lowest sound pressure levels not exceed 5dB, the result of the average value becomes almost correct even if it is only calculated as the arithmetic average value.

Example 2.3: Addition of Two Sound Sources
The above-mentioned example 2.2 with two engines can now be solved by (2–57)

$$L_{p_tot} = 10\log\left(10^{90/10} + 10^{90/10}\right) = 93\text{ dB} \tag{2–59}$$

However, addition can be used if you want to calculate the total noise loading of a house which is loaded with noise from several roads. If it is wanted to add the noise from 2 sources and the greatest sound pressure level is more than 10 dB higher than the lowest one, then the lowest level has in reality no influence on the total sound pressure level, which in this case will be synonymous with the highest level. This can be seen in figure (2–10) which states the size that is to be added to the highest of two dB values at an addition of two dB values in pairs. For example it can be seen that the sum of two sound sources with L_p = 0 becomes 3 dB.

Correspondingly, you can subtract the sound levels. This can be desired if the sound pressure level from a background noise L_{p_b} has to be subtracted from the total sound pressure level of a sound source $L_{p_{tot}}$ to define the sound pressure level of a sound source L_{p_k}

$$L_{p_k} = 10 \log \left(10^{L_{p_{tot}}/10} - 10^{L_{p_b}/10} \right) \tag{2–60}$$

Figure 2–11 states the size which has to be subtracted from the highest of two dB values at a subtraction in pairs of two dB values. At a difference of more than 10 dB the lowest sound level has no influence on the resulting sound level. If it is necessary to subtract the background level from the total noise level this can only be done with satisfactory accuracy if there at least is a difference of 3 dB of the two sound pressure levels. Otherwise, the result will be encumbered with too big uncertainty. This can be current if the noise at a house has been measured while it comes from two sources, for example a factory where the contribution of noise is known and a road where the contribution of noise is wanted to be calculated.

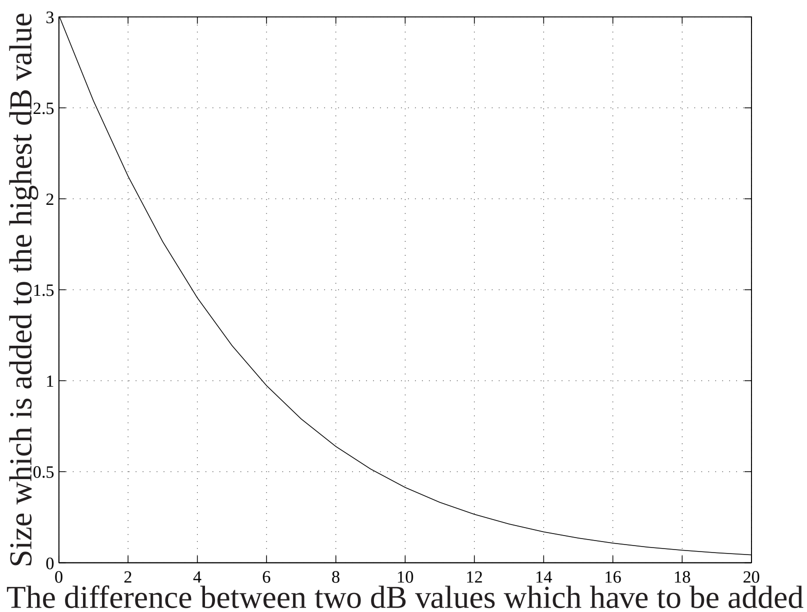

Fig. 2–10 Addition of sound levels.

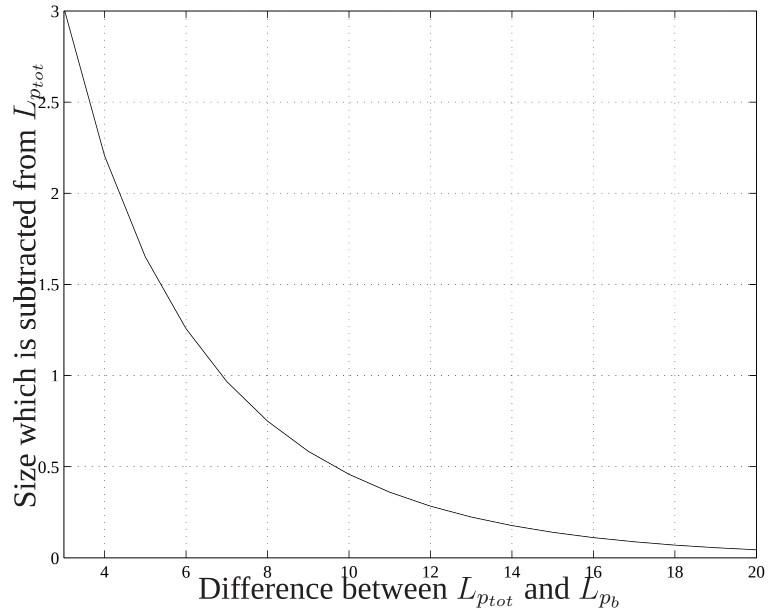

Fig. 2–11 Subtraction of sound levels.

2.5 The Perception of Sound of a Human Being

Normally, the human ear can perceive the sound in the frequency range from about 20 Hz to about 20 kHz in the sound pressure area 0 - 130 dB, see figure 2–12. The audible area varies from person to person, but especially the upper limit is dependent of a person's age and the noise loading that the ear has been exposed to throughout the life. The human hearing covers a great area of frequencies and sound pressure levels, where the lowest and the highest sound pressure levels correspond to *the threshold of hearing* and *the threshold of pain*, respectively.
Under 20 Hz you have *infrasound* and over 20 kHz *ultrasound* , which cannot be heard by the human ears but only by dogs and some other animals. Ultrasound is used for example to define the depths of water under the ships and within the health service to examine different organs. Contrary to infrasound ultrasound can be deadened by screening and hearing protector. Earlier it has been presumed that infrasound could not be heard at all, but now you know that this is not true. Infrasound can be heard if it is strong enough and the threshold of hearing is getting reasonable well determined. It is characteristic of infrasound that it is perceived as disturbing just because the sound intensity is a little bit stronger than the threshold of hearing. By common noise there is no hard-and-fast boundary from ”just audible noise” over ”distinct noise” to ”appreciably disturbing noise”. Notes which are higher than the deepest notes and are placed between these and common noise, are characterized as low-frequency noise. Here, it is also applied that when the noise intensity increases the disturbing effect becomes even stronger than if for example traffic noise was mentioned. However, the increase of the disturbing effect is not perceived to be quite as characteristic for low-frequency noise as for infrasound. When you are going to estimate low-frequency noise there is a need for a method which at the lowest notes is just as restrictive as if infrasound is going to be estimated, and which at the higher notes estimates the low-frequency noise almost like normal noise. The frequency between 1 and 8 Hz corresponds to the resonant frequencies of the body for which reason noise at these

frequencies can create resonant phenomena in the human body which have an influence on the blood circulation etc. Infrasound and low-frequency noise can be produced both from natural sources as wind and thunder and from artificial sources of noise as aeroplanes, combustion plants, ventilating plants, and vibrating machines.

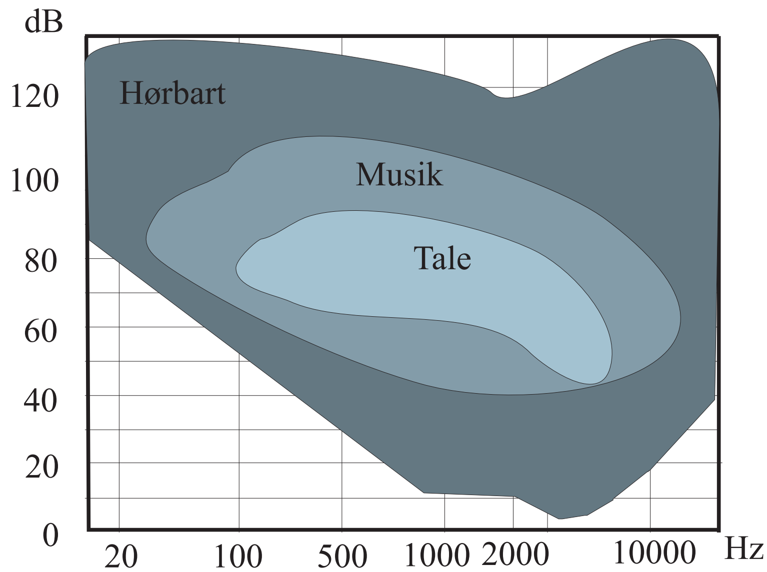

Fig. 2–12 The audible area is ranging about 10 octaves with wave lengths from about 17m to 17mm.

The way a human perceives the sound means that not all passes through to the consciousness. This is called *masking* and can be explained in this way that a sound in a given frequency range with a given strength prevents a person from hearing certain other sounds in the same frequency range with lesser strength. This means that if you are effected by a pure note at a certain frequency the threshold of hearing is considerably reduced at neighbouring frequencies. For example if a person is exposed to a pure note at 1 kHz on 80 dB the hearing of the person in the frequency range 500 Hz to 10 kHz will be effected in such a way that eg. a note with the frequency 2 kHz must have a sound pressure level at about 50 dB to be perceived. In the undisturbed case a 2 kHz will already be audible at 0 dB. The masking phenomenon is most well-defined in the high frequency range and is spreading to other frequency ranges based on the principles about overtones. Masking is made to make a ”*sound curtain*” which is used if you cannot avoid disturbing background noise. Then you can mask it with some music or something else which is nice to listen to and which is called a sound curtain.

2.5.1 Strength Level of Hearing

In figure 2–13 every single point corresponds to a note with a certain frequency and a certain sound pressure. The curves link together the points which human beings on average perceive as equally strong. These curves are called *hearing strength curves* or *ear sensitivity curves*. The lowest curve shows a *treshold of hearing*. The strength level of hearing which is the subjectively perceived sound intensity is stated in X *phon* which is equal to X dB at 1000 Hz. It is seen in the figure that a pure note at eg. 100 Hz with the *physical strength* corresponding to a sound pressure level at 50 dB is perceived by the human ear as a pure note at 1000 Hz with a sound pressure level at 40 dB. The sensitivity of the ear and with that *the physiological strength* of the sound is not only dependent on the frequency but also on the physical strength. Concerning weak sounds the sensitivity of the ear is considerably lesser at low frequencies than at high frequencies. This difference in sensitivity becomes lower the stronger the sound is.

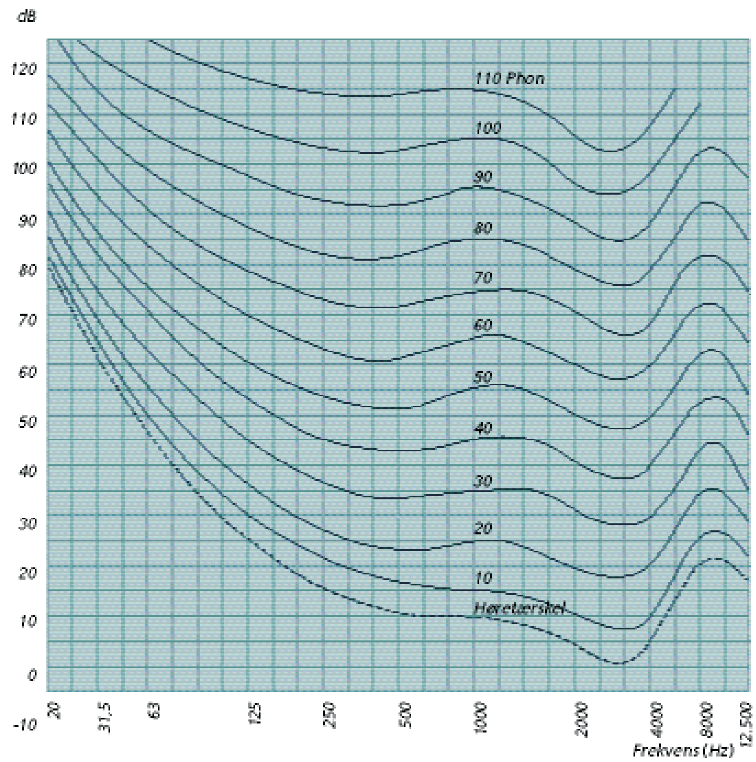

Fig. 2–13 Hearing strength curves. The curves link together the sound pressures at different frequencies which human beings on average perceive as equally strong. Den The subjectively perceived sound intensity is stated in the unit of ”phon”.

Hearing strength curves are based on a work by Fletcher and Munson in Bell’s laboratory in the thirties (Fletcher and Munson, 1933) and since then refined by Robinson and Dadson in the fifties (Robinson and Dadson, 1956) which have resulted in ISO standard curves. The curves were made by test persons who were going to judge whether pure sound notes at two different frequencies had the same sound pressure level. The curves show that the ear in principle acts as a filter which favours sound with a frequency contents in the area from 1 to 5 kHz and in particular about 4 kHz. This area also corresponds to the frequency contents mentioned.
The following general lines for the human perception of changes in the sound pressure level L_p can be stated as follows.

2.5.2 Influence of Sound on Humans

All sound which is disturbing, unwanted, noisy or irritating at work and when you rest etc. is noise. Whether a sound is noise or not for a person depends on many factors as for example

- contents of frequency

Change in L_p	Experienced a change i L_p
3 dB	Just audible
5 dB	Clearly audible
10 dB	Doubling/halving
20 dB	4 times or 1/4

Table 2-3 A human's subjective evaluation of changes in L_p.

- time variation
- duration
- sound source
- background noise
- spirits of the person

The permanent noise loading in our everyday life can result in reduction of hearing at an early time. There is a risk of a permanent damage of hearing if you more than 40 hours per week are exposed to a sound pressure of 85 db or more. If you are only in a short period exposed to a loud noise level you can get a short reduction of hearing, but the fine hair cells in the ear are quickly improved and the hearing is normalized.

On the other hand, if you are exposed to a continued heavy noise loading the hearing cells will be destroyed and they cannot be regenerated. Particularly dangerous to the hearing, the doctors mention loud music, eg. at discotheques or by using receivers, where the sound intensity is very loud. In these cases the sound intensity easily attains a level of 110-120 db.

In many workplaces you are also exposed to a high noise level. It is naturally that loss of hearing owing to noise has the leading place on the list of occupational diseases. It is not only affected to workers in the building trade and in noisy industrial halls but also to musicians in pop groups and orchestras.

The size of a hearing damage depends on factors as the capability of the person for responsiveness and the general state of health. Beyond the arose loss of hearing in the internal ear because of noise which can result in deafness, also nervous or organic damages can arise: If you for several years have been exposed to a constant loading of noise it might result in cardiovascular diseases, disturbances in the stomach and bowel system, balance disturbances, and psychical diseases.

Noise can also cause tinnitus (Latin: to ring). This is noise in one or both ears which can be experienced in a very different way and with different intensity, but the noise is not made from a sound coming from outside. The cause can be internal sounds in your own body (objective tinnitus) which also the doctor can hear eg. by means of a stethoscope. In case of ”subjective tinnitus” it is only the person itself who can hear the noise.

A sudden loud report, a detonation, a short or something like that can in a split second have the same effect as agelong loading of noise: A progressive loss of hearing and a special capability for further noise damages.

Table 2-4 gives some instructions for connections with defective hearing and different sound pressure levels.

Noise level	Damage
150 dB	permanent loss of hearing
130 dB	painful and should be avoided
110 dB	shorter intervals cause defective hearing
90 dB	intervals for some time give temporary defective hearing
65 dB	intervals for some time give tiredness

Table 2-4 Consequencies of noise.

Figure 2–14 shows that if you are exposed to high permanent noise you can get a hearing damage which shows that the threshold of hearing is dislocated upwards at some (or all) the frequencies.

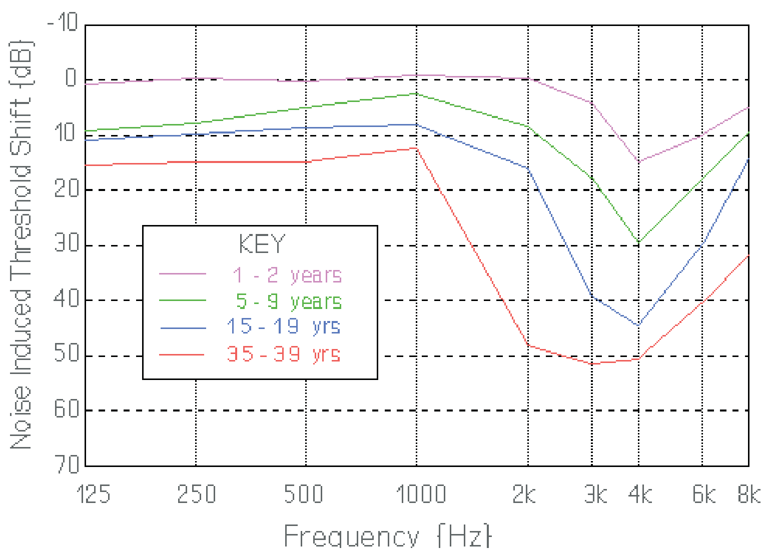

Fig. 2–14 Defective hearing at permanent intervals in noise.

Even if noise is unwanted and can be deleterious, it can in some situations also gives advantages to the perception of sound of a human being. A reasonable level of background noise gives among others a masking which makes it possible to reduce other kinds of sound/noise. Besides, a human is not feeling well in a total sound-dead area for some time.

2.5.3 Sound Weight Curves

As there is a difference between the physical sound intensity and the physiological strength, the hearing strength cannot be defined objectively correct by means of measuring instruments. Therefore, you have to use a measuring instrument which has built in a filter adapted to the frequency sensitivity of the ear. With this you add the sound intensity in the frequency range where the sensitivity of the ear is least, a relatively lesser importance. To describe this sensitivity to different frequencies the sensitivity curves A, B, and C are introduced. The curves correspond approximated to the frequency dependent sensitivity of the ear at 40, 70, and 100 dB sound pressure at 1000 Hz that is to say they correspond to 40, 70, and 100 phon. References to a sound pressure measured after one of these curves, dB(A), dB(B), or dB(C) can often be seen. Referring to acoustics of buildings the A-curve is often used. Please also notice that the hearing strength level curves become more flatten with a growing phon-number. Therefore, several weight curves B and C are made which were going to consider this fact which is seldom used. The different curves are usually used in the following case.

- A-weight can be used at sound levels less than 55 phon. However, A-weight is also often used at sound levels higher than 55 phon.
- B-weight is used at sound levels between 55 and 85 phon.
- C-weight is used at sound levels of more than 85 phon.
- Estimating aircraft noise (from a jet plane) you should use the D-weight which is specially developed for this purpose.
- Referring to the very lowest notes as infrasound you use another scale, dB(G), which can describe the sensitivity of the ear to these very low notes.

The A- and C-weight curves can be defined by the following expression

$$A_{weight} = 20\log\left(\frac{F_{A0}F_4^2 f^2}{(f^2+F_1^2)(f^2+F_2^2)}\right) \tag{2–61}$$

$$C_{weight} = 20\log\left(\frac{F_{C0} f^2}{(f^2+F_2^2)(f^2+F_3^2)}\right) \tag{2–62}$$

$$\begin{array}{ll} F_{A0} = 1,249936 & F_{AC} = 1,007152 \\ F_1 = 20,598997 & F_2 = 107,65265 \\ F_3 = 737,86223 & F_4 = 12194,127 \end{array} \tag{2–63}$$

where F_i are the constants in the filters and f is the frequency. Weight factors defined by these expressions are stated in table 2-5 and shown graphically in figure 2–15.
If the A-weighted sound pressure level L_{pA} is wanted to be calculated for a sound source, for which a frequency analysis as 1/3-octave or 1/1-octave is given, the values from table 2-5 are used to correct the values in the few frequency bands. The A-weighted sound pressure level L_{pA} is calculated by summation of the corrected sound pressure levels $L_{pkoor,i} = L_{p,i} + A_{weight,i}$ from the i'th frequency band of the few 1/3- or 1/1-octave bands which are divided into N_b frequency bands

$$L_{pA} = 10\log\left(\sum_{i=1}^{N_b} 10^{L_{pkorr,i}/10}\right) \tag{2–64}$$

Example 2.4: Determination of A-weighted sound pressure

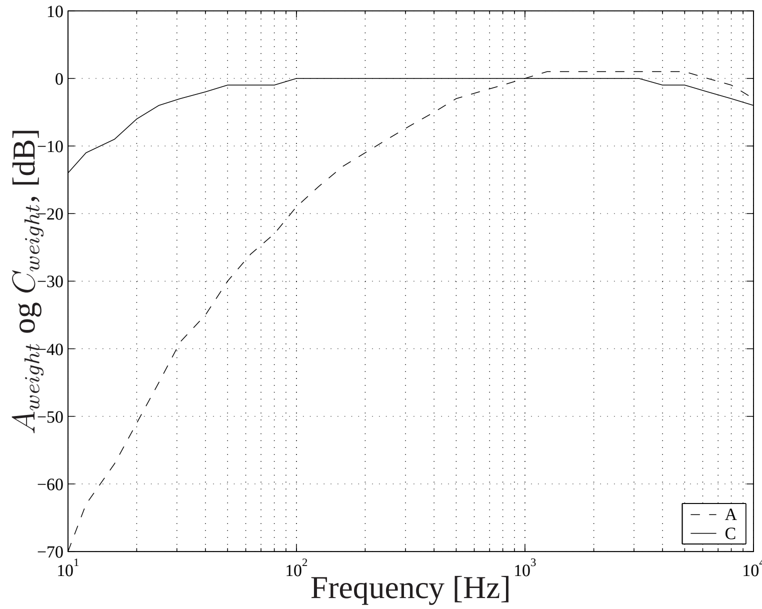

Fig. 2–15 A- and C-weights curves.

From the octave values in table 2-6 the A-weighted sound pressure level becomes

$$L_{pA} = 10\log\big(10^{(90-16)/10} + 10^{(94-9)/10} + 10^{(91-3)/10} + 10^{(89)/10} + 10^{(85+1)/10} + 10^{(85+1)/10} + 10^{(81-1)/10}\big) = 94\ \mathrm{dB(A)} \tag{2–65}$$

2.5.4 The Equivalent Sound Pressure Level

If variable working conditions or noise sources are found which do not load the surroundings with the same noise level all the time, the noise has to be reduced for over a certain period - named ”the reference period”. The noise level is stated as the equivalent sound pressure level and this means that it is the ”average” sound pressure level which is stated. Regarding the variable sound for the time T the energy equivalent A-weighted sound pressure is used which can be mentioned $L_{Aeq,T}$. The energy equivalent A-weighted sound pressure level corresponds to the sound pressure level from a constant sound which contains the same A-weighted sound energy as the considered variable sound. The equivalent sound pressure level can be defined by

$$L_{Aeq,T} = 10\log\left(\frac{1}{T}\sum_{i=1}\Delta T_i 10^{L_{pA,i}/10}\right) \tag{2–66}$$

where T is the reference period and T_i is the working time of the single working situation having a sound pressure level at $L_{pA,i}$.
Normally, the same reference period is not used the whole day and night, mainly because great variations in the noise level at buildings in the night are lesser acceptable than in the day. As an example the reference period is often much lesser in the night than in the day. As an example the reference periods can for eg. productive activity be

- The most noisy period of 8 hours in the day period from 07.00 to 18.00

Frequency [Hz]	A [dB]	C [dB]
10	-70	-14
12	-63	-11
16	**-57**	**-9**
20	-51	-6
25	-45	-4
31	**-39**	**-3**
40	-35	-2
50	-30	-1
63	**-26**	**-1**
80	-23	-1
100	-19	0
125	**-16**	**0**
160	-13	0
200	-11	0
250	**-9**	**0**
315	-7	0
400	-5	0
500	**-3**	**0**
630	-2	0
800	-1	0
1000	**-0**	**0**
1250	1	0
1600	1	0
2000	**1**	**0**
2500	1	0
3150	1	0
4000	**1**	**-1**
5000	1	-1
6300	0	-2
8000	**-1**	**-3**
10000	-3	-4

Table 2-5 Standardized A- and C-filters to a decision of sound intensity for 1/3- and 1/1-octaves division, (1/1-octaves are shown in boldface).

- The most noisy period of 1 hour in the evening period from 18.00 to 22.00
- The most noisy period of half an hour in the night period from 22.00 to 07.00

The reference period can be defined in an other way in some situations, as an example on

f [Hz]	125	250	500	1000	2000	4000	8000
$L_{p,i}$ [dB]	90	94	91	89	85	85	81

Table 2-6 Octave values.

Saturdays, Sundays, and holidays. Thus, as an example there will only be worked with one reference period = 1 hour concerning motoring tracks. Calculating $L_{Aeq,T}$ a short loud noise will give a great contribution to $L_{Aeq,T}$. This means that a constant noise over 15 minutes at 100 dB(A) corresponds to a energy equivalent A-weighted sound pressure level at 85 dB(A) over 8 hours. Table 2-7 shows the connection between retention time and sound pressure level corresponding to 85 dB(A) over 8 hours. You will see that an increase of 3 dB(A) in the sound pressure level corresponds to a halving of the retention time.

L_{pA}	Retention time before damage
85 dB(A)	8 hours
88 dB(A)	4 hours
91 dB(A)	2 hours
94 dB(A)	1 hour
100 dB(A)	15 minutes
103 dB(A)	8 minutes
106 dB(A)	4 minutes
109 dB(A)	2 minutes
112 dB(A)	1 minute

Table 2-7 Connection between L_{pA} and retention time.

CHAPTER 3

Architectural Acoustics - Sound Regulation in Buildings

The subject architectural acoustics is about how sound disperses and dies in closed rooms. It is possible by a correct choice of the room's size, shape, placing of sound reflecting/sound absorbing surfaces to optimize the acoustics to the use of the room, so that a good acoustic quality can be obtained, which as mentioned in chapter 1 can be characterized at objective and subjective targets related to the distribution and dispersion of sound in the room together with targets related to the noise level in the room.

The acoustic conditions which define the use of a room and with that if there is a good acoustics, are

- the direct and reflecting dispersion of sound from a sounding body to a receiver,
- the beginning and the dying of the sound, the echo,
- acoustic reflections (echo, flutter echo),
- sound proofing and background noise from for example ventilating plant

An estimate of the acoustic conditions of a room is usually done in relation to theories of sound dispersion in a closed room, in which it is presumed that the dimensions of the room are great compared to the wavelengths of the sounds which are of interest. Fields of sound can then be described at one of the following three methods

- wave theoretic architectural acoustics
- geometrical architectural acoustics
- statistical architectural acoustics

3.1 Wave Theoretic Architectural Acoustics

One of the main causes why sounding bodies sound different, partly in different rooms and partly in different positions, can be explained by means of the wave theory of a closed three-dimensional room. Use of the wave theory which is called *wave theoretic architectural acoustics*, is based on the wave equation and its harmonic solutions for closed rooms, where dissipation of energy (damping) is ignored. This means that a solution to the wave equation is

$$p(x,y,z,t) = X(x)Y(y)Z(z)\mathrm{e}^{(i2\pi ft)} \tag{3–1}$$

which by insertion in (2–4) and separation of variables give

$$\begin{aligned}\left(\frac{\partial^2 X}{\partial x^2} + k_x^2 X\right) &= 0\\ \left(\frac{\partial^2 Y}{\partial y^2} + k_y^2 Y\right) &= 0\\ \left(\frac{\partial^2 Z}{\partial z^2} + k_z^2 Z\right) &= 0\end{aligned} \tag{3–2}$$

where it must be applied that

$$k^2 = k_x^2 + k_y^2 + k_z^2 \tag{3–3}$$

In this case the wave equation is called *the Helmholtz equation*. This method gives a physical correct description of sound in closed rooms. Though the practical use of wave theory is limited in relation to an arbitrary shaped room an understanding of the basic principles in the wave dispersion in closed rooms is of great importance when a room with a good acoustics is elaborated.

When the wave theory is introduced, a room is considered as a complex resonator which has several kinds of natural vibrations (constant waves), each with its characteristic frequency (*natural frequency*) . As to the sound of a constant wave the human ear will perceive it as an irritating sound. The eigenfrequency $f_r(n_x, n_y, n_z)$ of a room is the frequency with which the room will produce vibrations after removal of the outer load which has brought the room out of the position of equilibrium. As to an empty rectangular room with plane, parallel, and completely reflecting surfaces $f_r(n_x, n_y, n_z)$ is given at the following expression

$$f_r = \frac{c}{2}\sqrt{\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2} \tag{3–4}$$

The sizes n_x, n_y, n_z are related to the wave number at

$$\begin{aligned}k_x &= \frac{\pi}{L_x}n_x \qquad n_x = 0,1,2,..\\ k_y &= \frac{\pi}{L_y}n_y \qquad n_y = 0,1,2,..\\ k_z &= \frac{\pi}{L_z}n_z \qquad n_z = 0,1,2,..\end{aligned} \tag{3–5}$$

where L_x, L_y, L_z are the edge lengths in a rectangular room, see figure 3–1. Therefore, there is a threefold infinity of eigenfrequencies in a rectangular room. The three lowest eigenfrequencies correspond to wave lengths, which are the double of the length, the width, and the height of the room. As to a given choice of n_x, n_y, n_z corresponds different kinds of natural vibrations which can be divided up into the following three types

- one-dimensional *axial natural vibrations* when two of the sizes n_x, n_y, n_z are zero. This means that the sound pressure only changes along a co-ordinate axis and the movement of the particle of air is parallel to this axis, see figure 3–2a.

- two-dimensional *tangential natural vibrations* when one of the sizes n_x, n_y, n_z is zero. Here the particles of air are making movements along two co-ordinate axes that is reflections from four walls arise, see figures 3–2b and 3–3.

- three-dimensional *oblique natural vibrations* when none of the sizes n_x, n_y, n_z are zero. For these natural vibrations the particles of air are moving along all three co-ordinate axes, which cause reflections from all six walls, see figure 3–2c.

If a sounding body is placed in a room the acoustic energy disturbed by the sound source will excite the natural vibrations in the room, so that standing waves are built up by a number of the eigenfrequencies of the room. The shape of these waves will depend on the damping (absorption) of the room.

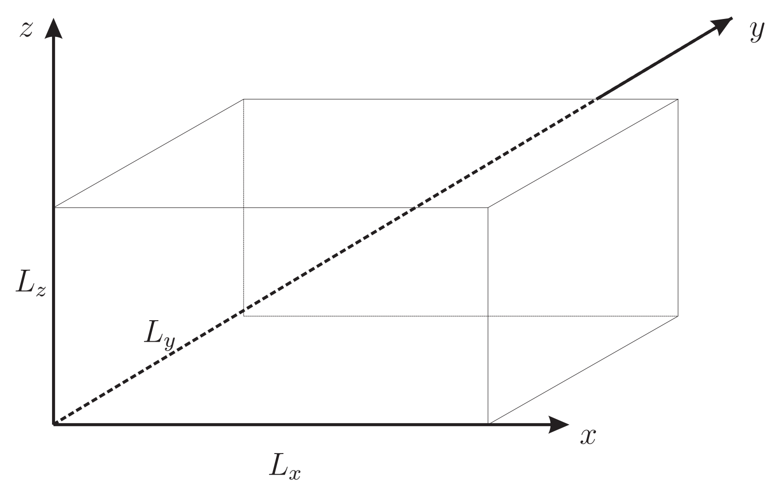

Fig. 3–1 Cartesian system of co-ordinates in a rectangular room.

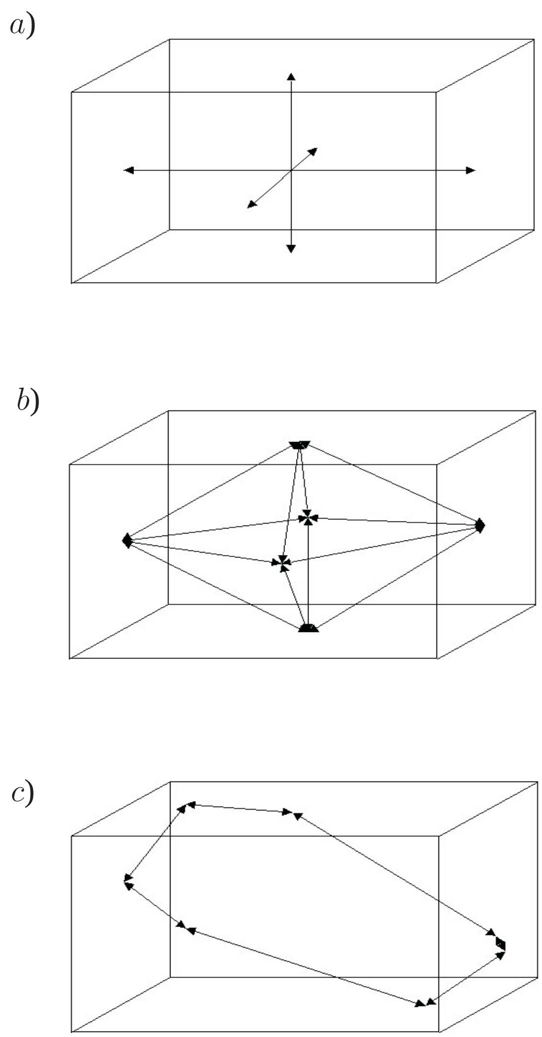

Fig. 3–2 Natural vibrations in a rectangular room.

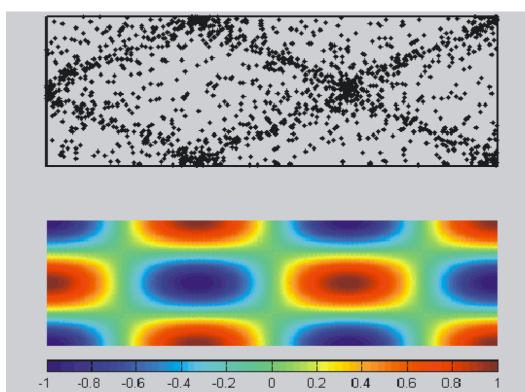

Fig. 3–3 Two-dimensional natural vibrations in a rectangular room for n_x=3,n_y=2,n_z=0.

Knowledge to the eigenfrequencies of a room is essential to understand the acoustic qualities of a room. To make a room with a good acoustics it is important to have a very even distribution of the eigenfrequencies as possible. If you choose to make a room with a simple proportion between the sides as e.g. 1:1:1 or 1:2:3, many of the eigenfrequencies will be identical and will give a possibility of many reflections which are mentioned in the next chapter. Many corresponding eigenfrequencies should be avoided and are especially important by dimensioning of small rooms, where the number of eigenfrequencies at low frequencies already are very small.

A recommended edge proportion in a room is often $1{:}\sqrt[3]{2}{:}\sqrt[3]{4}$, which however, will give big floor-to-ceiling height in larger rooms. In (Rindel, 1990) good edge proportion is stated for medium-sized rectangular rooms, see figure 3–4. For large rooms it is applied that the eigenfrequencies are so compact that a distribution is very even and does not give any problems.

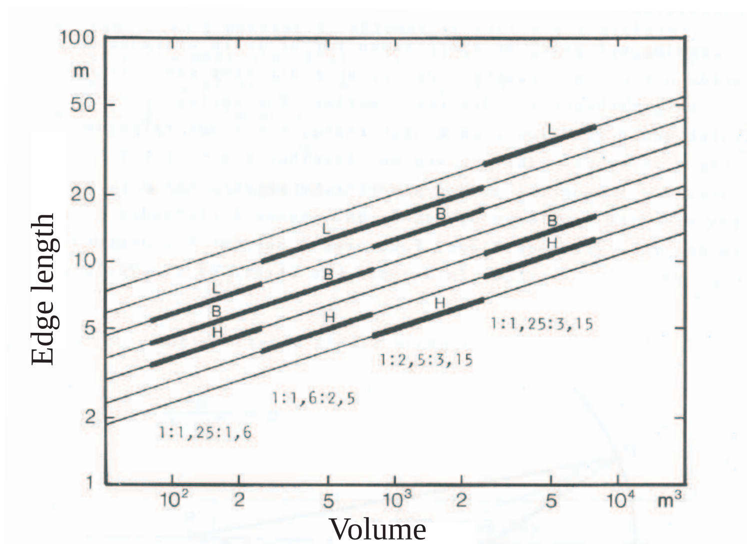

Fig. 3–4 Favourable dimensions for rectangular radio- and television studies. L : length, B : width, H : height.

3.2 Geometrical Room Acoustics

If you assume that the dimensions of a room are big in relation to the wavelength of the sound waves, these can be considered in the same way as light is considered in the optics. In the same way as light is reflected, the sound waves are reflected from hard surfaces according to the rules for reflection of light that means the angle of incidence is equal to the angle of reflection. That is to say that the wave fronts from a plane sound wave which hits a plane surface under the angle of incidence θ, will after the reflection still make a plane wave and move away from the surface under the angle of reflection θ, se figure 3–5.

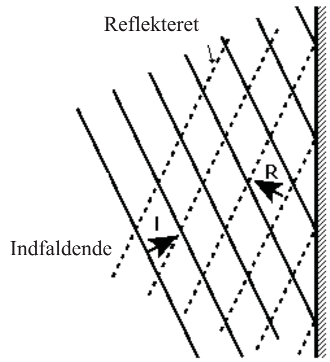

Fig. 3–5 Reflection of a plane wave from a plane surface.

Wave fronts from a spherical sound wave which hits a plane surface will after the reflection still make a spherical wave, which moves away from the reflecting surface, as if the front came from a fictive sound source placed in the reflection from the sounding body in the reflecting surface, see figure 3–6.

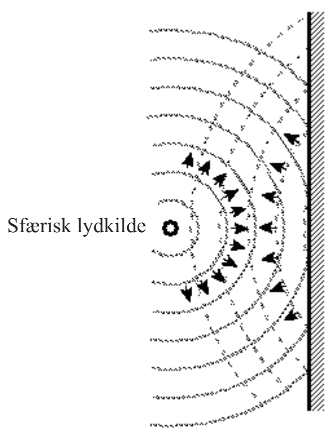

Fig. 3–6 Reflection of a spherical wave from a plane surface.

Furthermore, as it is for light it is applied that sound waves sent against a curved surface, will either be focused on curved surfaces or spread to convex surfaces, see figure 3–7.

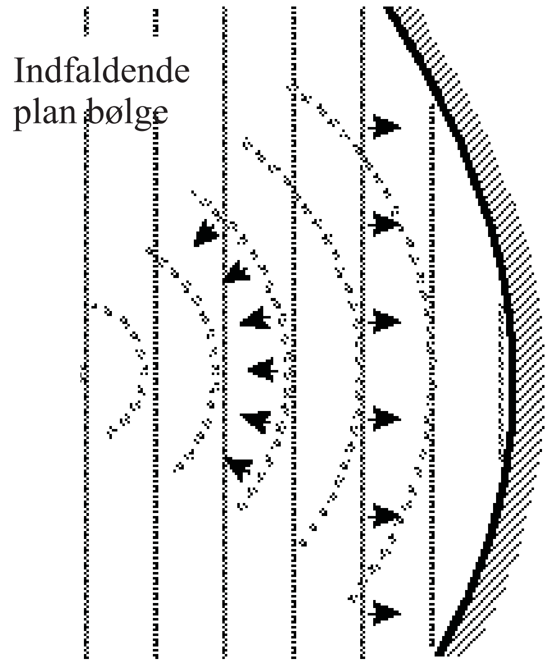

Fig. 3–7 Reflection of plane sound waves from a curved surface.

This method to analyse the acoustics of a room is called *geometric acoustics* and is a pure graphic method which informs of the dispersion direction of the reflecting sound, but does not inform anything about the strength of the total field of sound, which is compounded of the direct and reflected sound.
Information of the dying of the field of sound at the end of a sounding body is not given as well. Even if the geometrical acoustics is limited to detection of primary and secondary reflections, before the field of sound dies, it is possible based on this information to choose how acoustic defects can be repaired by a reasonable placing of the sound absorptions and reflectors, see chapter 5.
Geometrical acoustics is a valuable method when you make big rooms and lecture rooms, as acoustic defects as

- echo (audible reflections)
- flutter echo (audible series of reflections)
- ”dead” area (shadow formation)

can be discovered already when a room is made.
An *echo* is heard by a listener as a strong reflection of the direct heard signal from a sounding body, see figure 3–8. The echo will disturb the clarity of the direct sound and affect the subjective impression of the acoustics of the room. If the reflecting sound is just as heavy as the direct sound, you can hear an echo if the lag is greater than 50 ms. The ear does not hear a time lag lesser than 50 ms as a separate echo, but on the contrary as a strengthening of the direct sound. A lag of 50 ms corresponds to a difference in the disperse direction about 17 at a sound speed of 340 m/s. Therefore, by modelling a room you have to be aware of the distance. That the ear

does not register an echo after 50 ms is the so-called *Haas effect*, which refers to the human brain somehow ”deadens” the importance of sound in a period at about 50 ms after the ear has registered a sound signal the first time.

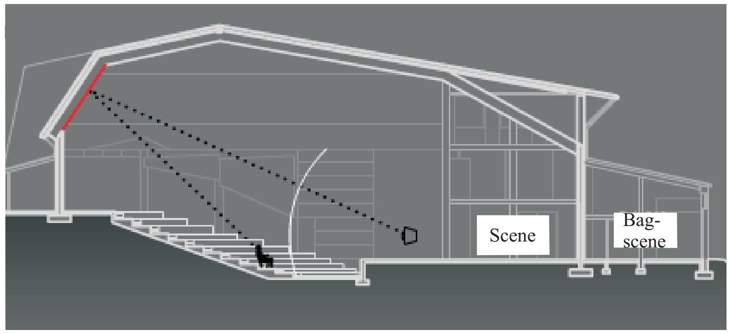

Fig. 3–8 Echo in a room.

When you have a room with parallel hard surfaces and when other surfaces near by are very little absorbing, then a *flutter echo*, might turn up, as the sound several times can be reflected backwards and forwards without an essential reduction of the sound intensity, see figure 3–9. A well-known example of flutter echo is acoustic development in a bathroom with parallel hard walls. Such a room has the lowest eigenfrequency in a frequency range, where the human voice has its greatest energy. Therefore, even a modest voice development might cause a significant sound which is strengthened when the natural vibrations are given. Flutter echo can be reduced by taking the distribution of the eigenfrequencies into consideration as mentioned in the chapter about wave theoretical architectural acoustics and use materials on wall surfaces which absorb the sound energy, see chapter 5.

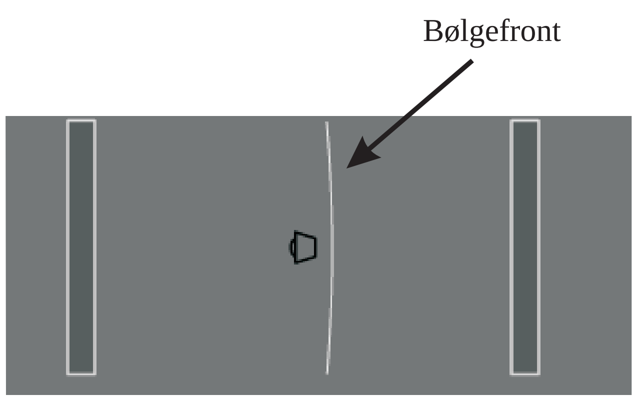

Fig. 3–9 Flutter echo in a room.

A *dead area* might arise a given place in a room where the sound neither can come as direct sound nor reflected sound, see figure 3–10

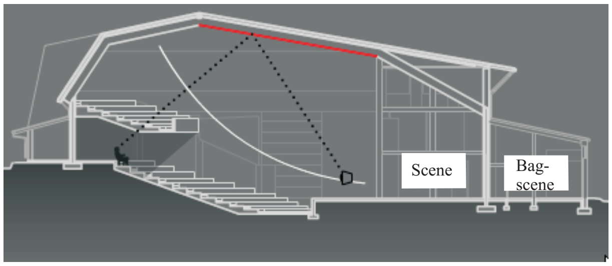

Fig. 3–10 Dead area (shadow formation).

3.3 Statistical Architectural Acoustics

Regarding too many practical purposes it is enough to calculate a field of sound in a closed room with the accuracy that makes it possible to define and estimate some typical characteristic sizes as the stationary sound energy and the time of the echo. Expressions for these sizes can be arranged based on a average value of the energy density in the room and not out from a more detailed description of the energy of the individual particles of air. From this the idea *statistical architectural acoustics* is coming.

3.3.1 Sound Absorption

When a sound wave, which disperses in air, hits a material as for example a wall, then a part of the sound effect will be reflected I_r and a part will be absorbed I_{abs}. From the absorbed quantity of energy some of it is changed to heat and a part of it will be transmitted I_t, so that a sound effect on the other side of the material is sent out. The ability of a material to absorbing the sound is characterized by the *Absorption coefficient* α, which states how great the energy of the incident sound wave is which is absorbed by the surface in question, see figure 3–11.

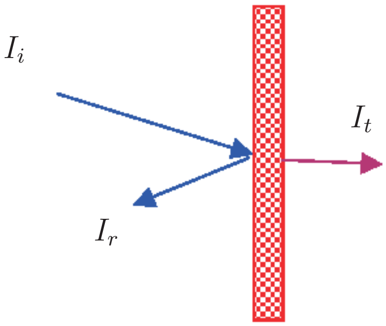

Fig. 3–11 Field of sound that hits absorbing material.

The absorption coefficient of a given material is stated by

$$\alpha = \frac{I_{abs}}{I_i} = \frac{I_i - I_r}{I_i} \tag{3–6}$$

where I_i is the incident sound energy of an absorbing surface. Referring to α which depends on the material, frequency and angel of incident, the following is applied

- $\alpha = 1$ means that all the incident sound energy is absorbed
- $\alpha = 0$ means that all the sound energy is reflected

In connection with the mention of the absorption coefficient α it is natural also to mention the sizes the loss coefficient δ, the reflection coefficient r, and the transmission coefficient τ. These are stated as

$$\begin{aligned} \delta &= \frac{I_i - I_r - I_t}{I_i} \\ r &= \frac{I_r}{I_i} \\ \tau &= \frac{I_t}{I_i} \end{aligned} \tag{3–7}$$

α, δ and τ are related in the following way

$$\begin{aligned} 1 &= \delta + r + \tau \\ \alpha &= \delta + \tau \\ \alpha &= 1 - r \end{aligned} \tag{3–8}$$

Porous materials like carpets, curtains, and acoustics ceilings have a high sound absorption, i.e. more than 0.5, while hard surfaces like linoleum, tile, concrete, and glass have a little absorption about 0.01 to 0.05. Most of the materials do not absorb equally effective at low and high frequencies and therefore, are often stated in tabular form at frequencies 125, 250, 500, 1000, 2000, and 4000 Hz, as shown in table 3.3.1, which gives an example of absorption coefficients for typical materials. More examples are given in appendix B.

Material	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Opening to the free	1	1	1	1	1	1
Wooden floor	0.15	0.11	0.10	0.07	0.06	0.07
Linoleum	0.02	0.02	0.03	0.04	0.04	0.05
Curtains 90mm from a wall	0.05	0.06	0.39	0.63	0.7	0.73
Smooth plaster on a hard wall	0.01	0.01	0.02	0.02	0.02	0.04
Unplastered tiled wall	0.02	0.03	0.03	0.04	0.05	0.07

Table 3-1 Examples of absorption coefficients.

As the absorption coefficient is dependent on the frequency the sound technical estimates will therefore be treated as a function of the frequency, e.g. low frequencies (low notes) separately and high frequencies (loud notes) separately, see figure 3–12, which shows that sound absorbing materials can be classified after an international standardized system, where class A is the most sound absorbing while class E is the least sound absorbing.

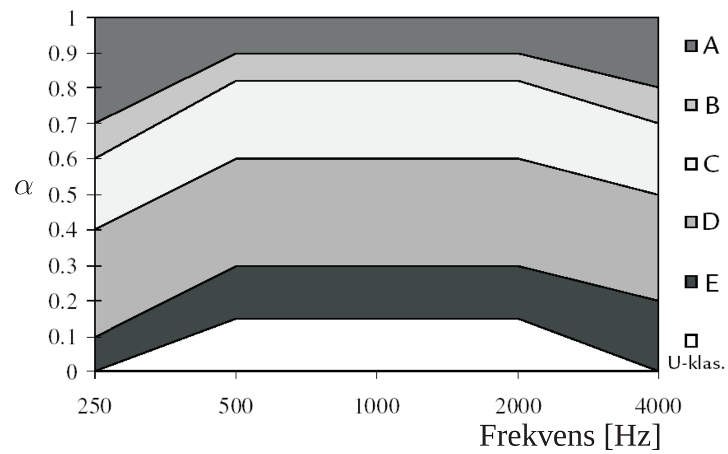

Fig. 3–12 Sound absorption class according to ISO 11654.

The standard ISO 11654 is developed with the wish to simplify the description of the sound regulation. It should be noted that you cannot conclude that a class A absorber sound technically is better than for example a class E absorber. This depends on what the room is going to be used for.

In practice you distinguish between 3 types of *passive* absorbers

- Porous absorbers
- Resonance absorbers
- Membrane absorbers

The porous absorber are porous, air penetrating materials like Rockwool, glass wool, carpets etc, which have a good absorption ability within a wide frequency range dependent on the thickness and the placement of the material

The absorption ability increases with the frequency and therefore porosity absorber finds first of all its use where absorption is wanted at average height and high frequencies.

Porosity absorbers work at the energy consumption that is produced when the particles of air move inside the porous material. Hereby the kinetic energy of particles of air is converted into heat energy by friction. To obtain a large friction the material must have a large number of small openings or pores so that a large friction for the sound pressure waves can be made. Regarding porosity absorbers it is characteristic that an additional absorption at low frequencies can be obtained by pulling our the absorber quite a bit out from an underlying surface. This is the principle you use at hanging ceilings, which are lowered about 0.2-0.3m compared to a firm ceiling.

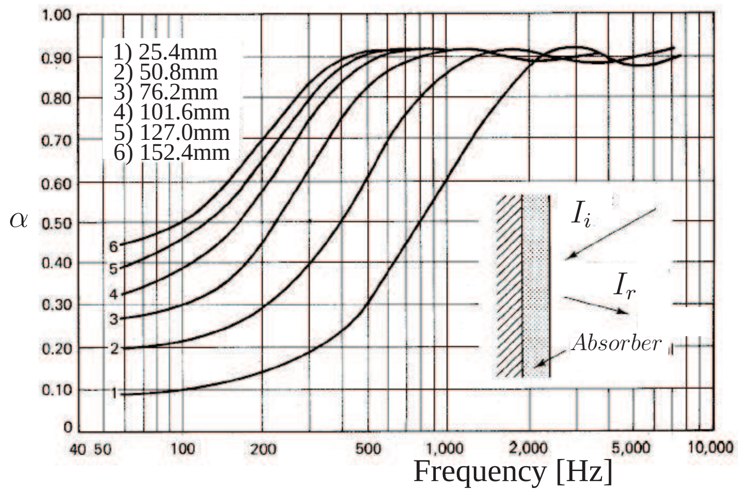

Fig. 3–13 Change in α as a function of frequency. The material is felt.

From figure 3–13 it is seen that a porous absorber is most effective to absorb at high frequencies. The efficiency depends on the thickness of the material relative to the wave length of the sound. In order to a porous absorber is effective at a given frequency the material thickness has to be at least $1/4$ of the wave length of the sound.

Example 3.1: Necessary thickness of porous absorber

A material with a thickness of 0.15m will at a room temperature at 20 ^oC absorb the sound with a frequency greater than

$$f = \frac{c}{\lambda} = \frac{343}{4 \cdot 0,15} = 572 \text{ Hz} \tag{3–9}$$

Absorption at low average height frequencies or in a delimited frequency range can be produced by means of *resonance*- and *membrane absorber*. These only give absorption within a narrow frequency interval which, however, in connection with the resonance- or membrane absorber can be enlarged by using a porosity absorber. Resonance absorbers are typical perforated sheets separated from an underlying surface. Thus, in this space is often used a porosity absorber. On the other hand, membrane absorbers are sheets without perforations, which also are placed separately from an underlying surface. Regarding the resonance arsorbers and the porosity absorber it is applied that they together with the underlying hollow space make a mechanical system which absorbs the energy by the resonance frequency of the system. Therefore, they are only effective for absorption of sound energy in a little frequency interval around these resonance frequencies. In chapter 5 there are examples of practical use of these different types of arsorbers. When the separate arsorbers are known for a room, then the *equivalent absorption area*, A with the unit m^2-Sabine, which states the number of absorption units of an object, a surface or a room, can be calculated by

$$A = \sum_{i=1}^{n} \alpha_i S_i \tag{3–10}$$

where α_i is the absorption coefficient of part areas S_i of the i'th part of for example the n surfaces of a room, which contributes to the absorption of the sound. From

Table 3-2 shows an example of the absorption coefficients for a wooden chair and a person, respectively, which can be included in an equivalent absorption area by

$$A = \sum_{i=1}^{n} \alpha_i S_i + \sum_{i=1}^{n_a} A_j n_j \tag{3–11}$$

n_j is the number of each unit (for example persons, chairs), which each has the absorption area A_j. n_a is the number of units.

	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
wooden chair	0,15	0,11	0,10	0,07	0,06	0,07
person	0,02	0,02	0,03	0,04	0,04	0,05

Table 3-2 Equivalent absorption area of persons and chairs .

In normal offices, class rooms etc., where the volume is less than 300 - 500 m^3, it is not necessary to show consideration for *air absorption*. In larger rooms the absorption of the air plays a certain part at higher frequencies, where it contributes to the total equivalent absorption area. This contribution S_{air} can be calculated as an equivalent absorption area

$$S_{air} = 4mV \tag{3–12}$$

where V is the volume and m is a frequency-dependent factor which varies according to the relative humidity, see table 3-3

	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
4*m*	0	0	0.0016	0.0040	0.0096	0.0240

Table 3-3 The contribution of air to the equivalent absorption area at 50% humidity.

When the absorption of the air is considered then the equivalent absorption area becomes

$$A = \sum_{i=1}^{n} \alpha_i S_i + A_j n_j + 4mV \tag{3–13}$$

When the equivalent absorption area A is known you can define the average absorption coefficient of a room α_m as

$$\alpha_m = \frac{A}{S} = \frac{\sum_{i=1}^{n} \alpha_i S_i}{\sum_{i=1}^{n} S_i} \tag{3–14}$$

where S is the total area of absorbed surfaces in a room.

3.3.2 Fields of Sound in Closed Room

The sound pressure level in a spherical field of sound is defined by the sound effect and the distance from the sound source, see (2–52). If a great reflective surface in a field of sound is placed nearby the sound source, then there will be both direct sound and reflective sound in a point in the field of sound. The reflective sound will be delayed compared to the direct sound because of the longer way and it is weaker, as a part of the sound energy is absorbed by the reflective surface. I.e. when the sound source is interrupted then first the direct sound disappears and then later on the reflective sound.

If there are more reflective surfaces around a sound source, as there is in a closed room, there will be more reflections to come from each side together with a direct sound.

The reflective sound will consist of sound which is reflected once and reflective sound which is reflected several times from the surrounding surfaces, see figure 3–14. Regarding the individual surrounding surfaces, if it is accepted that $\alpha_i \approx 0$, i.e. they do not absorb any important sound energy, then the energy density will be almost constant in the whole room, illustrated in figure 3–16 at the homogeneous distribution of the wave fronts. You can talk about a perfect diffuse field of sound *diffuse sound field*, if the energy density in a given point in a closed room is homogeneous distributed in all directions, i.e. you presume to have the same energy density all over the room and with equal probability in all directions. It happens practically with a fairly good approximation only in acoustic laboratories (hard sound rooms) which are used by acoustic test with more sound sources. This intuitive and approximate explanation of a diffuse sound of field is based on the theories which were made in the section about geometrical architectural acoustics. That is to say that you assume that sound radiates and reflects as ”light beams”. Approximately, you can assume to have a diffuse sound of field in a big room with very reflective walls, floors, and ceilings and several fields of sound.

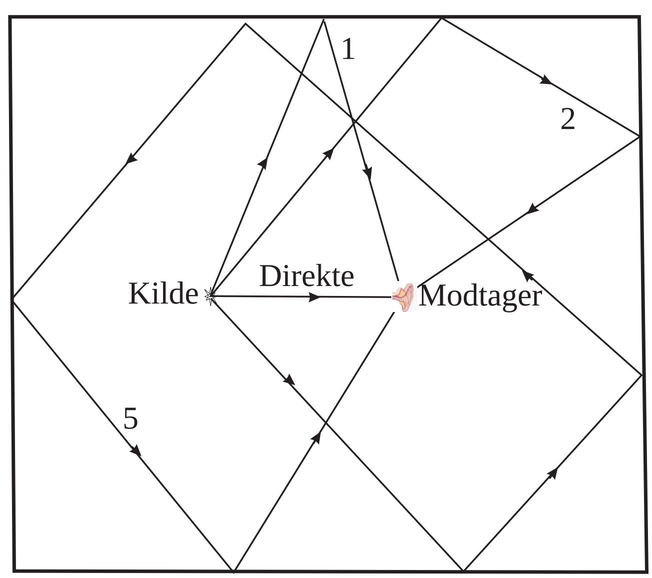

Fig. 3–14 Direct and reflective sound in a closed room.

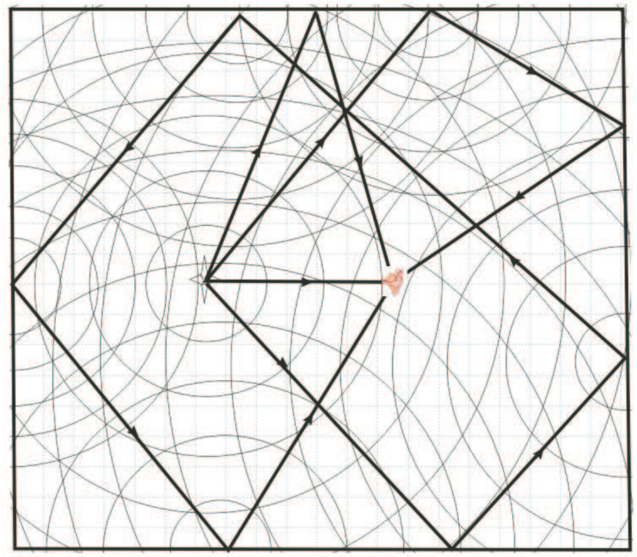

Fig. 3–15 Field of sound with direct and a great number of reflection in a closed room.

3.3.3 Stationary and Transient Energy Density in a Closed Room

When a field of sound occurs in a room it is understood to take place momentarily, as you first hear the direct sound, after which the sound level gradually will grow in strength because of the sound from the reflections. If a sounding body with the effect P is lit in a closed room with the volume V, there will be a growth of the energy density $\frac{d\varepsilon}{dt}$ which will be equal to the added effect minus that part of incident sound energy I_i, which is absorbed by surrounding walls. That equilibrium can be written if you assume a diffuse field of sound

$$V\frac{d\varepsilon}{dt} = P(t) - I_i\, A \tag{3–15}$$

In a room with a diffuse field of sound the total incident sound energy per unit of time of a surface with the area dS can be defined by projecting ($\cos(\theta)dS$) the intensity $\varepsilon dV/4\pi r^2$ into the surface dS. This gives

$$I_i\, dS = \int_V \frac{\varepsilon dV}{4\pi r^2}\cos(\theta)dS \tag{3–16}$$

where it is assumed that the sound radiates equally in all directions from the volume element dV with the energy density ε. The volume element is placed in the distance r from the wall, see figure 3–16, which means that dV can be described

$$dV = dr\; rd\theta\; r\sin(\theta)d\psi = r^2\sin(\theta)d\psi d\theta dr \tag{3–17}$$

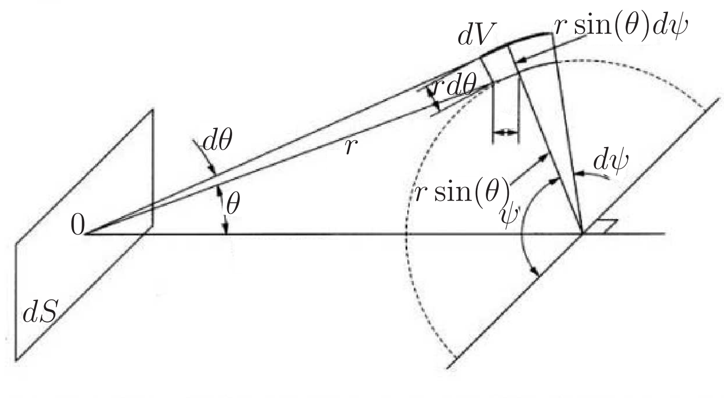

Fig. 3–16 Figure with angles.

From (3–15) and (3–17) the total energy from all the volume elements in the distance r can now be defined by integration of the angles ψ and θ

$$I_i = \int_0^{\pi/2} \int_0^{\pi} \frac{\varepsilon}{4\pi r^2} \cos(\theta) r^2 \sin(\theta) dr d\psi d\theta = \frac{\varepsilon c}{4} \tag{3–18}$$

as $dt = dr/c$ indicates the time lag it takes to the total energy in dV to run through dS. (3–18) shows that the intensity in a diffuse field of sound is a fourth of the intensity in a plane wave (2–34).

As a diffuse field of sound can be understood as being compound of a lot of uncorrelated plane fields of sound, then the energy density becomes, see (2–31)

$$\varepsilon = \frac{\tilde{p}^2}{\rho_s c^2} \tag{3–19}$$

With this (3–18) can be written as

$$I_i = \frac{\tilde{p}^2}{4\rho_s c} \tag{3–20}$$

From (3–15) it can be seen that for the stationary state ($t \to \infty$) the stationary energy density is ε_s in a diffuse field of sound

$$\varepsilon_s = \frac{4P}{cA} \tag{3–21}$$

(3–19) and (3–21) give with this the following simple expressions to define the effective sound pressure in a point, when the effect of the sounding body and the equivalent absorption area of the room is known.

$$\tilde{p} = \sqrt{\frac{4\rho_s c P}{A}} \tag{3–22}$$

By using (2–45) together with (3–22) the sound pressure level L_p in a stationary diffuse room can be written

$$L_p \approx L_w + 10\log\left(\frac{4}{A}\right) \tag{3–23}$$

As the direct sound will be dominant just around the sounding body the relation is applied when the sound pressure is considered in a great distance r from the sounding body.

In the same way as it is possible to make a relation between L_p and L_I in a free spherical field of sound (2–48), it can also be made to a compound field of sound in a room. By using (**??**) L_I is defined by

$$L_I = 10\log\left(\frac{\tilde{p}^2}{4\rho_s c I_0}\right) = 10\log\left(\frac{\tilde{p}^2}{p_0^2}\frac{p_0^2}{4\rho_s c I_0}\right) = L_p + 10\log\left(\frac{100}{\rho_s c}\right) = L_p - 6\text{ dB} \quad (3\text{–}24)$$

From this it can be seen that for a diffuse field of sound L_I and L_p are not identical, as it is for a plane field of sound and a spherical field of sound in the distant field, see (2–48).

Generally, a field of sound in a closed room is not diffuse. Therefore, to calculate the level of the sound pressure in a closed room you can assume that the field of sound is a combination of a free spherical field of sound and a diffuse field of sound. This means that the effective sound pressure is given by a sum of the effective sound pressure from the spherical field of sound (2–39) and the diffuse field of sound (3–22), respectively

$$\tilde{p}^2 = \left(\frac{P\rho_s c}{4\pi r^2} + \frac{4\rho_s c P}{A}\right) = P\rho_s c\left(\frac{1}{4\pi r^2} + \frac{4}{A}\right) \quad (3\text{–}25)$$

However, by using (2–45) the sound pressure in a *combined sound field* in a closed room will then be

$$L_p \approx L_w + 10\log\left(\frac{1}{4\pi r^2} + \frac{4}{A}\right) \quad (3\text{–}26)$$

In a stationary diffuse field of sound the sound is reflected from surrounding surfaces minimum once, which gives the reduced sound effect $P(1-\alpha_m)$. This means that (3–22) can be written

$$\tilde{p}^2 = \frac{4\rho_s c P(1-\alpha_m)}{A} = \frac{4\rho_s c P}{R} \qquad R = \frac{A}{1-\alpha_m} \quad (3\text{–}27)$$

where R is mentioned as the room constant. With this (3–26) will be changed to

$$L_p \approx L_w + 10\log\left(\frac{1}{4\pi r^2} + \frac{4}{R}\right) \quad (3\text{–}28)$$

Figure 3–17 shows how *room damping* $(L_p - L_w)$ varies as a function of the room constant and the distance to the sounding body. The figure shows that in a short distance from the sounding body in a room with a big R, i.e. a great equivalent absorption area A $(L_p - L_w)$ is reduced when the distance r is increased. Opposite to this, with a too little R $(L_p - L_w)$ is on the whole constant. It should be noticed that (3–28) under- and overestimate the sound level close to and far from the sounding body, respectively.The figure also shows that there is a limit to how much $(L_p - L_w)$ can be reduced by increasing R. This limit corresponds to 6dB by a doubling of r, which corresponds to the result from the law of range (2–38).

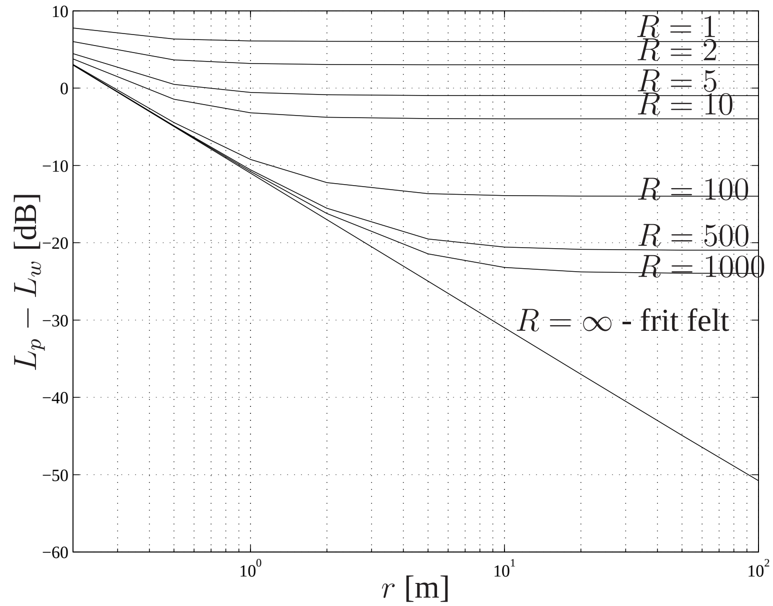

Fig. 3–17 The sound pressure level as a function of the room constant R and the distance r to the sounding body in a closed room .

Table 3-4 shows typical values of room damping at different types of rooms, see (Valbjørn et al., 2000).

Type of room	Room damping	Description
Normal office	4 dB(A)	Wall-to-wall carpeting
Open-plan office	12 dB(A)	Carpet and sound absorbing ceiling
Conference room	10 dB(A)	Carpet and sound absorbing ceiling
Classroom	7 dB(A)	Sound absorbing ceiling and 2 walls with notice boards
Bedroom	4 dB(A)	

Table 3-4 Typical values of room damping $(L_p - L_w)$.

If P is assumed to be constant in (3–15), a stationary situation will arise at a time, where the absorbed sound energy is equal to the sent out sound energy from the sounding body. The stationary complete solution to (3–15) is given by

$$\varepsilon = \frac{4P}{Ac}\left(1 - \mathrm{e}^{-\frac{cA}{4V}t}\right) = \varepsilon_s\left(1 - \mathrm{e}^{-\frac{t}{t_k}}\right) \qquad t_k = \frac{4V}{Ac} \tag{3–29}$$

where t_k is designated as a time constant. If the sounding body is cut off after that the energy density has attained its stationary condition, then the transient solution to (3–15) will be

$$\varepsilon = \varepsilon_s \mathrm{e}^{-\frac{t}{t_k}} \tag{3–30}$$

(3–29) and (3–30) show that the growth and the dying from the energy density of a field of sound happen at the same exponential function.

3.3.4 Reverberation Time

The figures 3–18 and 3–21 show the relative energy density's $\varepsilon/\varepsilon_s$ growth and dying in a closed room in a linear and logarithmic correspondence, respectively. The time constant in (3–29) decides how quick the energy density will grow in a closed room. If the equivalent absorption area A is small then it will take relatively long time before the energy density in a room reaches its final stationary value. The figures show as mentioned earlier that the human ear perceives that the growth from a field of sound happens momentarily, as you first hear the direct sound, after which the sound level gradually will grow in strength on account of the sound from the reflections. When the sounding body in a room is cut off then the feeling will be as if the sound disappears very slow. This is due to the fact that after the sounding body is cut off then there will be lots of reflections from the sent sound from the surrounding surfaces, see figure 3–15. Not before these reflections cease the field of sound will die away. This gradually fall in the sound energy is mentioned reverberation. This reverberation process is quite clear in some rooms and is among others important for intelligible speech in a room. For example, if the temporal distance between some syllables and words from a speaker in an auditorium is less than the time it takes for this sent sound to die away, then a straight pronounced syllable can still be heard when the next syllable or word is said. This will give an acoustics where it is difficult to distinguish the single sounds from each other. On the other hand, regarding music it is just an advantage that there is a superposition between the single notes, so that the sound is perceived as melting together. A room with a hard sound is a room where the sound is hanging in the room for a long time - long reverberation, while a room with a short reverberation will be considered as a dead room.

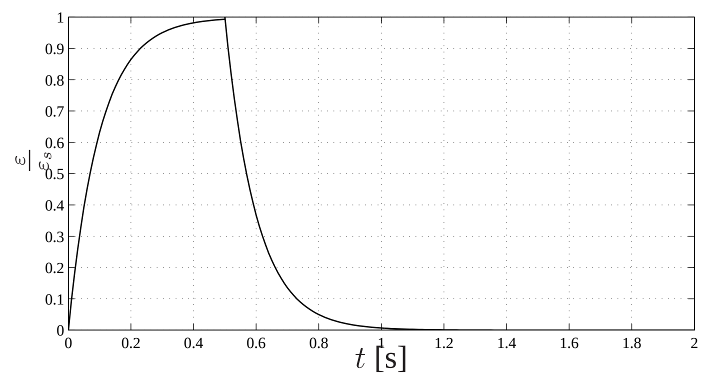

Fig. 3–18 Relative energy density as a function of the time in a linear function.

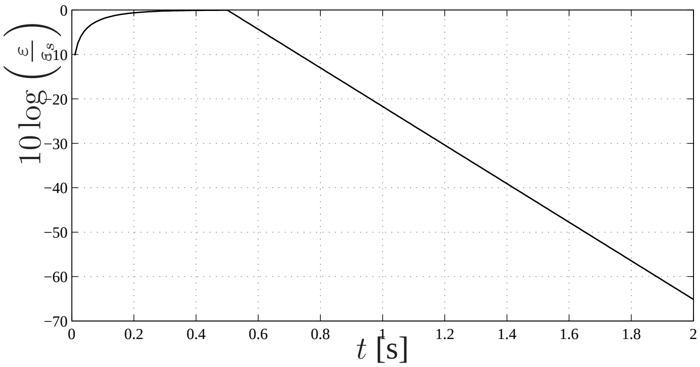

Fig. 3–19 Relative energy density as a function of the time in a logarithmic function.

Subjectively, a room with a long reverberation is considered as being ”resounding”, while a short reverberation is considered as if the room is ”sound dead”. The most objective target for a room’s acoustic quality is the reverberation time which is the time that passes before a momentarily cut off sound pressure level from a source of sound or noise is fallen 60dB. In small rooms with parallel walls it happens that the dying of the field of sound often differs from the expected rectilinear course as shown in figure 3–21. Therefore, it is difficult to define a real reverberation, also because that the conditions of the static architectural acoustics are not fulfilled. On the other hand, for greater room it is possible to make an objective target for the reverberation. Using (2–45), (3–30), and (3–19) you get Sabine´ s formula of reverberation

$$-60 = 10\log\left(\mathrm{e}^{-\frac{t}{t_k}}\right) = -\frac{10}{2,303}\frac{t}{t_k} \tag{3–31}$$

If (3–31) is solved as regards to $t = T_{Sab}$, you get

$$T_{Sab} = 55,3\frac{V}{cA} = 0,161\frac{V}{A} \tag{3–32}$$

where T_{Sab} is the reverberation is in seconds and c=343 m/s. That you can call it Sabine´ s reverberation formula is due to the fact that W.C. Sabine in the period from 1895 to 1898 came to the same formula after some tests. In 1895 Sabine studied physics at Harvard, where you had some troubles with the sound in a new-built auditorium (Fogg Art Museum). Sabine was set to solve the problem and the investigations and tests he made the next three years brought to the formula of reverberation. As the reverberation depends on the geometry of the room and the sound absorbing qualities of the surrounding surfaces, then the reverberation will be frequency dependent and therefore, it is important by calculations to consider the frequency dependence of the reverberation. Normally, 6 standard frequencies with 1/1 octavo intervals and the center frequencies 125, 250, 500, 1000, 2000, 4000 Hz are used in the architectural acoustics, which is the frequency range which is interesting for speech and music. Normally, if only a reverberation is stated, it is done at 500Hz. Many people think that the reverberation should be as homogeneous as possible in all octavo bands so that all the frequencies are treated in the same way. As it is seen from the absorption coefficients in appendix B, it is often difficult to get the reverberation down at the low frequencies. But as a rule-of-thumb a placing of a great porous absorber in the corners of the room could help a lot on this problem, as there is a pressure maximum for all frequencies in the corners of a rectangular room. However, this will

also lower the reverberation at the other frequencies. With this it can be seen that it can be a complicated matter to optimize a reverberation of a room.
Already in the project phase you should consider the conditions that ensure an optimal reverberation in a given room. Often the size of the room and the floor and wall surfaces are determined beforehand which means that the possibility of the reverberation often is limited to the ceiling construction. Here a short reverberation causes a large quantity of absorbing material whereas the opposite results in a long reverberation.
Regarding rooms with noisy activities, workshops and so on the reverberation should be as low as possible. In other types of rooms, for example classrooms or auditoria which are used for speech, the optimal reverberation is 0,5 $-$ 1,2 seconds, dependent of the size. The reverberation must not be too long as the speech will then be resounding and difficult to understand. On the other hand, the reverberation must not be too short either, as there will then be a risk of the speech dying out before it reaches the audience. Rooms for practising music demand a longer reverberation 1 $-$ 2,5 seconds, so that the music does not sound dead. Here the optimal value depends very much on what kind of music is being discussed. Besides, generally it is applied that the reverberation of a room is strongly dependent on whether the room is crowded with people or not. That you do not want the same reverberation in a room which is used for speech and a room which is used for music as in a church give incompatible demands to the reverberation. Figure 3–20 shows an example of the connection between the volume of a room and the instructive reverberation, (**?**).

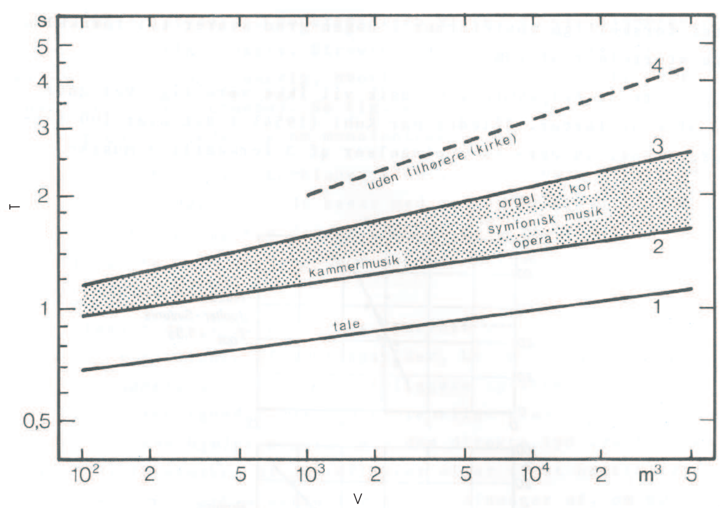

Fig. 3–20 The connection between the volume of the room and the reverberation.

In (Stephens and Bate, 1950) the following simple formula is recommended to determination of an instructive optimal reverberation at 500 Hz

$$T_{Sab} = K(0,0118V^{1/3} + 0,1070) \tag{3–33}$$

where K is put to be

- K=4, for room to speech
- K=5, for room to music from an orchestra

- K=6, for room to choir-singing

One thing is to define the reverberation in a room, but as a room in reality consists of two or more connected rooms, for example in a theater where the stage room is coupled to the room with the audience, then the reverberation has to be calculated for each of the rooms, as no absorption between the rooms $\alpha = 1$ is assumed. (Rindel, 1990) has made an offer for how you then should act. Generally, it can be concluded that in acoustic coupled rooms the inconvenience are greatest in that case that you are in the room with a shorter reverberation than in the coupled room.
In chapter 5 there will be given some examples of how the reverberations can be regulated by using different materials and designs of rooms, so that you get reverberations which can live op to the claims in (og Boligstyrelsen, 1995).

3.3.5 The Validity of The Sabine Reverberation Formula

(3–32) is good for a quick estimation of T_{Sab} in a given room and gives good results at the most applications. However, (3–32) has many limitations in its use, among others because it does not consider the shape of the room together with the place of the absorbing material. (3–32) is based on the following theories

- diffuse field of sound
- absorbing material placed homogeneously on all surfaces
- no big openings
- the field of sound is diffuse
- the sound disperses with the same probability in all directions
- no focusing

These theories are difficult to fulfil in many cases. For example in an auditorium the sound will be sent out in one direction, and you have a great difference in absorbing areas. The theory about a diffuse field of sound is also difficult to fulfil. The field of sound in a given room can be considered as diffuse for frequencies over Schröder's limiting frequency f_{sch}

$$f_{sch} = 1900\sqrt{\frac{T_{sab}}{V}} \qquad (3\text{–}34)$$

When you have a homogeneous place of absorbing material in a room Sabine´ s reverberation formula gives good results, as one of the fundamental assumptions of the formula is fulfilled. However, by increasing the quantity of the absorbing material the results become even more worse. Among others it is seen that (3–32) is not applied in the line where $\alpha_m = 1$, corresponding to a free range.
The formula should never be used by calculating the reverberation in strong damping rooms where the sound intensity varies very much with the distance to the sounding body. With a reasonable accuracy it can be used when it comes to the single absorption coefficients that

$\alpha \leq 0,2$ separately and the average absorption coefficient $\alpha_m \leq 0,2 - 0,3$ which in most cases can be fulfilled. If the average absorption coefficient exceeds 0,3 the reverberation can be calculated according to Eyring´ s formula

$$T_{Sab} = 0,161 \frac{V}{-S\ln(1-\alpha_m)} \tag{3–35}$$

This expression can be derived in the same way as (3–32) by assuming that the energy density in the room is based on a theory about reflections from plane waves. At every reflection the energy density is reduced with a factor ($(1-\alpha_m)$). Eyring's formula gives good results, if the room has a homogeneous distribution of absorbing material. If you have a room with a great variation in absorbing material it is better to use Millington−Sette's formula

$$T_{Sab} = 0,161 \frac{V}{\sum_{i=1}^{n} -S_i \ln(1-\alpha_i)} \tag{3–36}$$

where you just have substituted an average value for the logarithm of the absorption coefficients of the individual surfaces. This formula gives in principle more weight to absorbing material than it has importance for the reverberation.
In table (3-5) are shown the results from calculated reverberations which are obtained by different formulas not for calculation of the reverberation for a big room of 308 m^3 with a moderate number of absorbing material.
The formulas are not mentioned above as they are only modification of Eyring's formula, where you suggest different ways of considering variation in the absorption coefficients. Table (3-6) shows the reverberation for the same room, in which two different kinds of measuring positions are used, in the middle of the room and alongside the room, respectively. When the results in the tables are compared a great variation in the calculated results are seen, dependent on which fundamental assumptions you base your calculations. Moreover, a great variation between calculated and measured values is seen. From this it can be concluded that care has to be devoted every time you have to choose one of the formulas for calculating the reverberation in a room. Especially, when it is a room with no straight shape.

Formula	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz
Sabine	1.42	0.90	0.68	0.78	0.72	0.68
Eyring	1.34	0.82	0.60	0.71	0.64	0.60
Arau	1.63	1.00	0.66	0.80	0.69	0.64
Fitzroy-Sabine	2.07	1.34	0.83	1.01	0.83	0.79
Fitzroy-Eyring	1.99	1.26	0.75	0.94	0.75	0.71

Table 3-5 Comparison of calculated reverberation.

3.4 Objective Acoustic Measure

For many years after that W.C Sabine recommended his formula (3–32) for determination of the reverberation in a closed room, this was the only acoustic measure to determine the acoustics

	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz
Position 1	1.77	1.54	1.52	1.62	1.67	1.70
Position 2	2.04	1.54	1.53	1.62	1.67	1.70
Average	1.91	1.54	1.53	1.62	1.67	1.70

Table 3-6 Comparison of measured reverberations.

of a room. To many engineers and architects this is still the only measure they use to-day. With the passage of time many different objective architectural acoustic measures are recommended to define the acoustic conditions of a room and are only mentioned briefly here, see for example (Rindel, 1990) and (**?**). Among these the following measures are the most used

- EDT (Early Decay Time)
- RASTI-index (Rapid Speech Transmission Index)
- STI (Speech Transmission Index)
- Clarity
- Deutlichkeit

Most of these items only show separately consideration to the individual factors which are important for the acoustics of a room.
EDT is a measure of the reverberation based on a fall of the sound pressure level of $10 - 15$ dB instead of a fall of 60 dB, because it is this course of the reverberation that the human being can perceive. Objects of good listening conditions which are good speech intelligibility which is dependent on sufficient hearing strength and clearness is also made. In a room the speech intelligibility expresses popularly spoken how great deal of the spoken message that can be understood a given place in the room. The speech intelligibility depends strongly on the signal/noise circumstances and the reverberation. The speech intelligibility is measured as for example by a RASTI-index (RApid Speech Transmission Index) on the scale of 0 to 1. The speech intelligibility should be at least 0.6 in ordinary classrooms and should preferably be more than 0.8 before you talk about having a good speech intelligibility in a room. Normally, the value will vary from seat to seat in a room and in certain cases ”dead areas” will occur, where the speech intelligibility is considerably worse than in the rest of the room. To a great extent the speech intelligibility depends on the acoustic conditions mentioned above whether a room has a good acoustics.
Clarity and Deutlichkeit are related to the first 50 ms of a sound’s arrival to the human ear. The measures give a ratio of the quantity of early and late arrived sound energy.

3.5 Internal Noise

Generally, it is applied that a halving of the reverberation causes a reduction in the sound level of 3 dB, which subjectively will be heard as a considerably audible improvement in a noisy room.

Therefore, demands on a reverberation are used as for example stated in (og Boligstyrelsen, 1995), among others to lower the sound pressure level in a room.

To examine how much absorption material which has to be provided a room to lower the noise level the formula (3–28) can be used, as it gives a connection between the room constant and the sound pressure level in a room. However, to choose the type of absorption material, you first have to make a frequency analysis of the sound signal which gives the noise in a room. In chapter 5 it will be stated how you in practice can solve the noise problems in closed rooms by using different absorbers.

Sometimes you use a total measures for the noise in the whole frequency range. Several methods exist for this purpose, of which NC, PNC- and NR-curves are most used. NC-curves (Noise-Criterion) were developed in 1957 in USA for an estimate of internal noise from for example a ventilating plant. For a spectrum of the noise an NC estimate can be made by plotting the sound pressure levels from each octave band into an NC curve. Then a noise have an NC estimate which corresponds to the lowest NC-curve that is not exceeded by the plotted values. As an example the results in table 3-7 give an NC estimate of 46, which is seen in the figure 3–21, where the values from the table 3-7 exceed NC-45 with 1 dB at 500 Hz.

Center frekvens	62.5	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8kHz
Sound pressure level	41	45	48	50	46	42	40	38

Table 3-7 Results from a noise measurement.

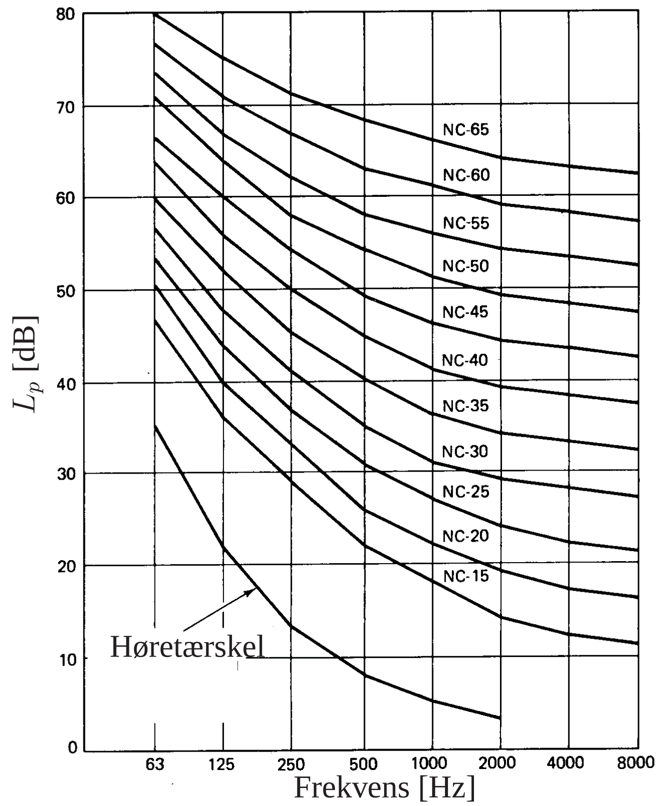

Fig. 3–21 NC kurver.

PNC-curves (Preferred Noise-Criteria) were introduced in 1971 as a modification of NC-curves as offices projected out from NC-curves were too noisy regarding air-conditioning. The figures in table 3-7 give a PNC judgment of 47.

Generally speaking, NR-curves (Noise-Rating)correspond to NC-curves, but are mainly used in Europe. NR-curves are mentioned in relation to their use and demands in chapter 5.

CHAPTER 4
Sound Isolation

CHAPTER 5
Principles for Architectural Acoustics Design

CHAPTER 6
Sound Measurements in Buildings

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The wave Equation

APPENDIX B
Absorptions Coefficients

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