



Acoustics of reverberation chambers

Associated software: ESDUpac m1511



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ACOUSTICS OF REVERBERATION CHAMBERS

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ACOUSTICS OF REVERBERATION CHAMBERS

1. NOTATION

		Units	
		<i>SI</i>	<i>British</i>
c_0	velocity of sound	m/s	in/s
d_{rms}	r.m.s deviation from averaged (ideal) frequency spacing	Hz	Hz
f	frequency	Hz	Hz
f_c	cutoff frequency	Hz	Hz
$f_{n_x n_y n_z}$	natural frequency of mode (n_x, n_y, n_z) in three-dimensional enclosure	Hz	Hz
I	r.m.s. spacing index (Sepmeyer)		
$\bar{I}$	time-averaged sound intensity	W/m ²	lbf/in s
I_{av}	average spacing index		
k	wavenumber	m ⁻¹	in ⁻¹
L	sound pressure level	dB	dB
L_{per}	sum of lengths of all edges of rectangular enclosure	m	in
L_{pi}	sound pressure level at i th measurement point	dB	dB
L_{pm}	mean of L_{pi}	dB	dB
L_w	sound power level of source	dB	dB
L_x, L_y, L_z	dimensions of edges of rectangular enclosure in x -, y - and z -directions	m	in
L_R	average band pressure level in receiver room (measurement of SRI)	dB	dB
L_S	average band pressure level in source room (measurement of SRI)	dB	dB
N	number of intervals between modes		
N_S	number of microphone measurement points		
n_x, n_y, n_z	integers defining modes in rectangular enclosure		
$\overline{p_d^2}$	mean-squared sound pressure (direct field)	Pa ²	(lbf/in ²) ²

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$\overline{p_{d,axis}^2}$	mean-squared sound pressure at distance r on designated axis	Pa ²	(lbf/in ²) ²
$\overline{p_i^2}$	mean-squared sound pressure for eigenmode i	Pa ²	(lbf/in ²) ²
$\overline{p_r^2}$	mean-squared sound pressure of reflected sound	Pa ²	(lbf/in ²) ²
Q	directivity factor		
Q_{axis}	directivity factor for particular axis		
Q_0	strength of monopole sound source	kg/s	*
q_0	spectral density	kg/s ^{1/2}	*
R	sound reduction index	dB	dB
R_c	room constant	m ²	in ²
r	distance from sound source	m	in
r_h	distance where direct and diffuse fields are equal (room radius)	m	in
r_{zx}	dimensionless room parameter ($= L_x/L_z$)		
r_{zy}	dimensionless room parameter ($= L_y/L_z$)		
S	total surface area of room	m ²	in ²
$S_1, S_2 \dots$	areas of different parts of room	m ²	in ²
S_a	area of sample in sound power absorption coefficient measurement	m ²	in ²
S_R	surface area of receiving room (sound reduction index measurement)	m ²	in ²
S_{SR}	surface area of test panel (sound reduction index measurement)	m ²	in ²
s_M	standard deviation of sound pressure level readings for six measurement points	dB	dB
s_S	standard deviation of sound pressure levels for N_S measurement points	dB	dB
T	reverberation time	s	s
T_a	reverberation time with test sample in reverberation room	s	s
T_i	reverberation time for eigenmode i	s	s
T_{60}	reverberation time for 60 dB decay	s	s

V	volume of room	m^3	in^3
V_s	volume of source room (measurement of coupling loss factors)	m^3	in^3
W	sound power	W	lbf in/s
x_0, y_0, z_0	coordinates of monopole sound source		
$\bar{\alpha}$	average sound absorption coefficient		
α_a	sound absorption coefficient of sample		
$\bar{\alpha}_R$	average sound absorption coefficient in receiving room		
$\alpha_1, \alpha_2, \dots$	absorption coefficients for different surfaces of rooms		
β	normal admittance of reverberation room surface		
δ	actual mode spacing	Hz	Hz
$\bar{\delta}$	ideal mode spacing	Hz	Hz
η_{SR}	coupling loss factor between two acoustic volumes		
λ	wavelength	m	in
ρ_0	density	kg/m^3	*
τ	sound intensity transmission coefficient		
τ_{SR}	transmission coefficient (measurement of coupling loss factors)		
ω	circular frequency	rad/s	rad/s

* In British units, for the system to be coherent the mass in lb must be divided by the constant 386.4

Abbreviations

DFAT	direct field acoustic test
IEST	Institute of Environmental Sciences and Technology
ISO	International Standards Organization
OASPL	overall sound pressure level
PWT	progressive wave tube
SPL	sound pressure level

2. INTRODUCTION

The sound that is heard in a room from a source of sound includes direct sound and sound reflected from the walls, ceiling, floor and objects in the room. In a room with a large amount of material that absorbs the sound, the direct field dominates, particularly in the region close to the source. In the case where the room boundaries are formed by hard, reflecting surfaces the reflected sound is the most significant. Rooms which are specially constructed to have dominant reflected sound are called reverberation rooms, and are the subject of this Item. Reverberation rooms are used for the measurement of the sound power of noise sources and of sound absorption coefficients. A high noise level can be achieved in reverberation rooms where the sound fields are diffuse. Reverberation rooms are also useful for testing the endurance of structures, such as satellites, in sound fields. Two reverberation chambers, connected via a panel, enable the transmission loss or the coupling loss factor of the panel to be determined. For sound fields that are considered to be coherent and in phase over the surface of the structure, the endurance testing should be carried out in a progressive wave tube (PWT).

3. ACOUSTICS OF ROOMS

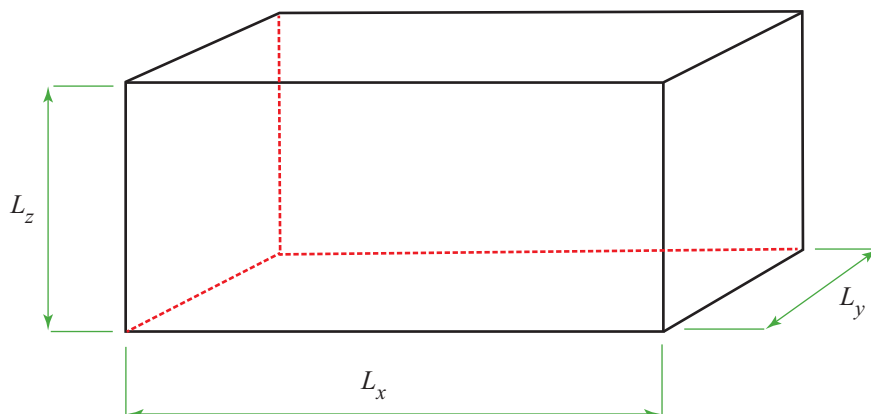
Sound is a wave motion and sound waves in enclosed spaces have to satisfy the boundary conditions at the walls and ceiling, *etc.* The general solution of the sound wave equation which fulfils the boundary conditions for a finite volume is a series of discrete functions of frequency, called eigenfunctions or normal modes of vibration. Every eigenfunction is characterized by a unique frequency of oscillations which is known as the natural frequency, eigenfrequency or normal mode frequency. The normal modes, and the natural frequencies that correspond to them, depend on the shape and size of the enclosure.

3.1 Sound in Rectangular Enclosures; Wave Motion

Thus, wave motion in an enclosed space can occur only at a certain frequency of vibration, the natural frequency. For a three-dimensional rectangular enclosure (right parallelepiped) with rigid sides of length, L_x , L_y and L_z , as shown in Sketch 3.1, the natural frequencies are given by

$$f_{n_x n_y n_z} = \frac{c_0}{2} \sqrt{\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2}, \quad (3.1)$$

where n_x , n_y and n_z are integers (0, 1, 2, 3....) and c_0 is the velocity of sound in the medium of the enclosure. Equation (3.1) is obtained from the acoustic wave equation in conjunction with the boundary conditions at the rigid walls (Kinsler *et al.*, 1982; Morse, 1948)



Sketch 3.1 Three-dimensional enclosure

The eigenfunctions, associated with a particular natural frequency, Equation (3.1), may be expressed in terms of pressure as

$$p = \cos \frac{\pi n_x x}{L_x} \cos \frac{\pi n_y y}{L_y} \cos \frac{\pi n_z z}{L_z}. \quad (3.2)$$

When the sound field in the enclosure is considered as a steady-state response to a monopole source with harmonic excitation of frequency ω , the expression for the sound pressure (associated with a particular eigenmode or natural mode i) at a point (x, y, z) , is

$$p_i = \frac{i\omega Q_0}{V} \sum_{n_x, n_y, n_z} \frac{\left(\epsilon_{n_x} \epsilon_{n_y} \epsilon_{n_z} \cos \frac{\pi n_x x_0}{L_x} \cos \frac{\pi n_y y_0}{L_y} \cos \frac{\pi n_z z_0}{L_z} \cos \frac{\pi n_x x}{L_x} \cos \frac{\pi n_y y}{L_y} \cos \frac{\pi n_z z}{L_z} \right) e^{i\omega t}}{k^2 - \left(\frac{\pi n_x}{L_x} \right)^2 - \left(\frac{\pi n_y}{L_y} \right)^2 - \left(\frac{\pi n_z}{L_z} \right)^2 - 2ik\beta \left(\frac{\epsilon_{n_x}}{L_x} + \frac{\epsilon_{n_y}}{L_y} + \frac{\epsilon_{n_z}}{L_z} \right)}. \quad (3.3)$$

The monopole sound source of strength (mass flow per second), Q_0 , is located at point (x_0, y_0, z_0) (Maling, 1967). The volume of the enclosure, V , is $L_x L_y L_z$. The wavenumber, k , is ω/c_0 and β is the normal admittance of the room surface.

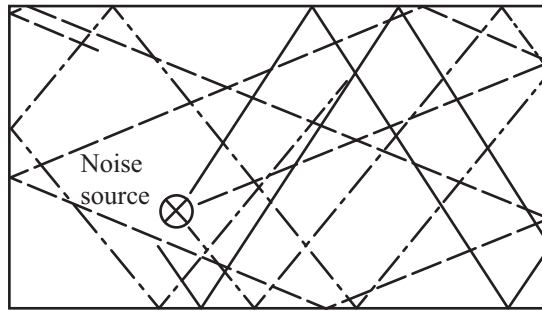
The factors ϵ_{n_x} , ϵ_{n_y} and $\epsilon_{n_z} = 1$ for n_x, n_y and $n_z = 0$
 $= 2$ for n_x, n_y and $n_z \neq 0$.

Modes in rectangular rooms may be classified as axial, tangential or oblique, as described in the ESDU Item on the Modal Density of Acoustic Spaces (ESDU, 2014). The eigenfunctions and the variations of sound pressure, particle velocity and energy density are discussed in Appendix B and a computer program is available which illustrates the functions for a particular mode.

For a large room the normal modes are so numerous in the audible range of frequencies that it is impossible in practice to use an approach based on wave motion in order to make predictions about the sound field. Also, for enclosures with a complicated geometry the wave equation cannot be solved analytically. Therefore, instead of using the wave equation the concept adopted in geometrical room acoustics is that of sound rays.

3.2 Sound in Enclosures; Geometrical Room Acoustics

In geometrical room acoustics sound, like light, propagates along rays which are lines that are always perpendicular to surfaces of constant phase. Illustrated in Sketch 3.2 is the concept of sound rays and sound reflections in a room. Paths in a plane are considered. The sound ray model is valid when (a) the amplitude of the wave and (b) the speed of sound do not change significantly with distance compared to a wavelength. Both conditions (a) and (b) should be satisfied in the ray model. Each ray of sound is independent of the others: there is no interchange of energy between rays of sound, hence phenomena such as wave diffraction cannot be taken into account.

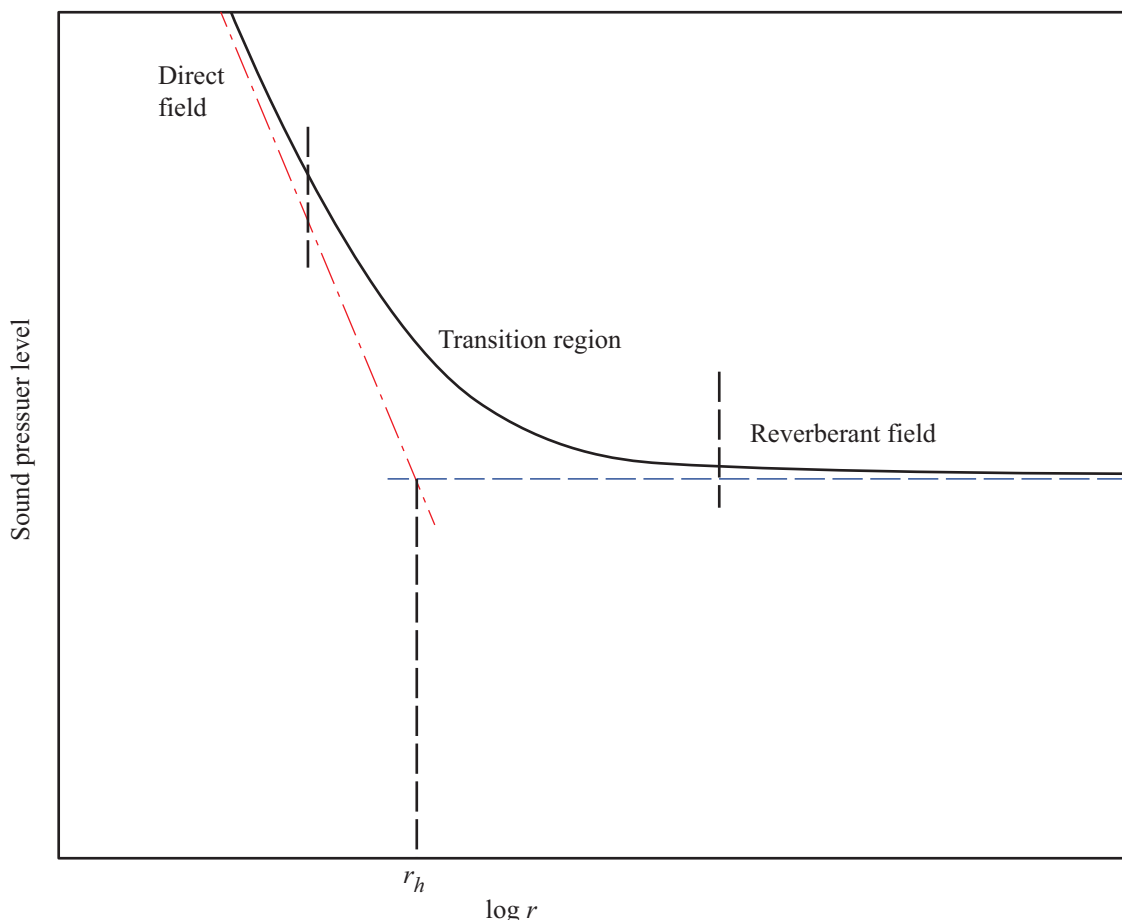


Sketch 3.2 Reflection of sound rays in a room

The sound ray approach is quite reasonable at high frequencies. High frequency sound from sources is often very directional and forms straight beams like rays of light. For the case of the reflection of sound the proper application of the sound ray model is restricted to specular reflection, as in optics. The wavelength should be small compared with the dimensions of the reflecting surfaces, but large with respect to the size of the surface roughness. The surfaces should be smooth and large for the method of geometrical ray acoustics to give good results. The method is often used, even at low frequencies, because there is no easily applicable alternative.

3.3 Variation of Sound Pressure Level with Distance from Source

When the distance from a sound source increases, the sound pressure level, as measured by a microphone, varies with distance as shown in an idealised way in Sketch 3.3.



Sketch 3.3 Variation of sound pressure level in a room with distance r from noise source

Close to the sound source the sound field is not influenced by reflections from room surfaces and is, hence, referred to as the free field. Additionally, because any measurement point is receiving sound directly from the source, it is also called the direct field. In the free field for a monopole sound source the sound decays at the rate of 6 dB for every doubling of distance r . In other words, the decay of the mean-squared sound pressure obeys the inverse square law. The free field is described and quantified in terms of sound wave motion.

Further away from the sound source the reflected sound is dominant and the field is referred to as reverberant. The reflected sound is said to form a diffuse field in the room when the reflections are so numerous that rays of sound are arriving in all directions and all phase shifts of the rays are equally probable. There is no preferred direction for the reflected sound at any point in the room; any direction is equally probable. To achieve a diffuse field there should be no surface areas of the room boundaries where the sound absorption is much greater or less than the average. In a diffuse field the sound pressure level and hence the averaged sound energy density are constant. Geometrical room acoustics is used to quantify the sound. Any measurements made in the diffuse field are independent of position. The direct and diffuse fields are equal at a distance r_h , known as the room radius, as indicated in Sketch 3.3.

The terms far field and near field are used to describe the sound field of a source. In order to differentiate between the far and near fields the form of the source has to be considered. For a point source which radiates spherical waves the far acoustic field exists when $kr \gg 1$, where k is the wave number (ω/c_0 , in terms of angular frequency ω and c_0 , or $2\pi/\lambda$, in terms of wavelength λ). The sound pressure and the particle velocity are in phase. In the near field $kr \ll 1$, the sound pressure and particle velocity are out of phase. In the near field there is no wave motion and the movement of fluid may be described as hydrodynamic. The near field cannot be easily defined mathematically and, furthermore, measurements may be subjected to considerable fluctuations. In practice it may be necessary to ignore the severe inequalities that define the near and far fields and choose a boundary between them. For a point or small source, the boundary could be one wavelength from the source.

3.4 Direct Field

The space where the boundaries have no effect on the propagation of sound is the direct field. The sound waves propagate from the point source as spherical, concentric waves. The sound intensity is the same on all points on the wave front and the time-averaged sound intensity, $\bar{I}$, is given by (Beranek, 1954)

$$\bar{I} = \frac{\overline{p_d^2}}{\rho_0 c_0}, \quad (3.4)$$

where $\overline{p_d^2}$ is the mean-squared sound pressure at a distance r . The sound power of the point source, W , is related to the time-averaged sound intensity by

$$W = 4\pi r^2 \bar{I}. \quad (3.5)$$

Hence,

$$\overline{p_d^2} = \frac{W \rho_0 c_0}{4\pi r^2}. \quad (3.6)$$

Equation (3.6) may be generalised to account for the position of the sound source in the room by introducing a directivity factor, Q , so that

$$\overline{p_d^2} = \frac{W\rho_0 c_0 Q}{4\pi r^2}. \quad (3.7)$$

The directivity factor Q is 1, 2, 4 or 8 for the point source located in the middle of the room, on the floor, at the intersection of two surfaces of the room and in a corner, respectively (Beranek, 1954). Thus, for a source on a hard, reflecting floor, $Q = 2$ and Equation (3.6) is modified to become

$$\overline{p_d^2} = \frac{W\rho_0 c_0}{2\pi r^2}. \quad (3.8)$$

The point source is an idealised source which radiates sound uniformly in all directions. Most sources of sound are directional, particularly at high frequencies. The directionality of sound sources is also accounted for by a directivity factor Q_{axis} which for a particular axis is defined as the ratio of the time-averaged sound intensity at a point located on a designated axis of the directional sound source to the time-averaged sound intensity at the same point from a point source of the same sound power. Hence,

$$Q_{axis} = \frac{4\pi r^2 \overline{p_{d,axis}^2}}{W\rho_0 c_0}, \quad (3.9)$$

where $\overline{p_{d,axis}^2}$ is the mean-squared sound pressure at a distance r from the source on the designated axis (Beranek, 1954).

3.5 Diffuse Field

As the measurement or observation point moves further away from the sound source, it reaches the diffuse or reverberant field where the reflected sound is dominant. An acoustic field is considered to be perfectly diffuse if the energy density is the same at all points in the volume (Pearce, 1989; Nelisse and Nicolas, 1997). Although a diffuse field is often called reverberant, a reverberant field is not diffuse if one surface of the room has reflective properties which are very much different from the rest of the room. A diffuse field may not exist in the transient stage just after a source has been switched on, but the field may become diffuse once steady-state conditions have been achieved.

The mean-squared pressure $\overline{p_r^2}$ of the reflected sound is constant in space for a diffuse field and depends upon the acoustic power of the source, the properties of the medium in which the waves propagate and the properties of the surfaces of the room, as

$$\overline{p_r^2} = \frac{4W\rho_0 c_0}{R_c}, \quad (3.10)$$

where R_c is the room constant (Beranek, 1954; Pierce, 1989). The room constant is defined in terms of the average sound absorption coefficient $\bar{\alpha}$ of the surfaces of the room for the room as a whole:

$$R_c = \frac{S\bar{\alpha}}{1 - \bar{\alpha}}, \quad (3.11)$$

where S is the total surface area of the room.

3.5.1 Average sound absorption coefficient

The average sound absorption coefficient takes into account the absorption of all room surfaces. In many rooms the ceiling, for example, is made from one material and its entire surface S_1 is associated with a particular sound absorption α_1 . Similarly, the walls of area S_2 may be made of a uniform material and characterised by a sound absorption coefficient α_2 , and so on. Thus, $\bar{\alpha}$, the average sound absorption coefficient includes the effects of the different surfaces, as

$$S\bar{\alpha} = S_1\alpha_1 + S_2\alpha_2 + \dots \quad (3.12)$$

The products $S_1\alpha_1$, $S_2\alpha_2$, etc., are sometimes called the absorption areas and the product $S\alpha$ the total equivalent absorption area. In the case of objects where the surface area is difficult to define, such as chairs, tables or machines, the absorption area is used to describe the absorption. For small value of α , the product $S\bar{\alpha}$ is approximately equal to the room constant, R_c .

3.5.2 Reverberation time

The acoustic properties of a room are often characterised by the reverberation time. This is the time required for a sound in the room to decay in intensity to almost zero. More precisely, the reverberation time is defined as the time required for the mean-squared sound pressure to decay to one millionth of its initial value. Or the time for the sound pressure level to decrease by 60 dB. An expression for the reverberation time in seconds was determined experimentally by Sabine as (Sabine, 1964)

$$T = \frac{60V}{1.086c_0S\bar{\alpha}}, \quad (3.13)$$

where V is the volume of the room. Note that an anomaly occurs for $\bar{\alpha} = 1$, the condition for complete absorption, in Equation (3.13). A finite value of reverberation time is predicted for $\bar{\alpha} = 1$, whereas a zero reverberation would be expected for the case of no reflections. The Eyring formula for reverberation time provides an accurate result for this condition and is (Eyring, 1930):

$$T = \frac{60V}{-1.086c_0S\ln(1-\bar{\alpha})}. \quad (3.14)$$

Equation (3.13), the Sabine equation, is used for low absorption coefficients, whereas the Eyring equation, Equation (3.14), can be used for average absorption coefficients greater than 0.2.

A reverberation time based on the decrease of sound pressure level of 60 dB is indicated by T_{60} . The method, which is most usually used for the measurement of reverberation time, is the interrupted noise method (ISO 354, 2003). In this approach a broad-band random sound is established in the space and is suddenly discontinued. The reverberation time is established from the decay of the sound pressure level. An omni-directional sound source, usually based on a loudspeaker system, is used together with an omni-directional microphone. The averaging of several measurements is required for each source and measurement position. Ensemble averaging is often used and in this form of averaging the sound pressure level at each time (where $t = 0$ corresponds to the instant when the source is interrupted) is added and divided by the number of records. A different approach is adopted in the integrated impulse response method, where an impulsive sound source, such as a pistol shot or a bursting balloon, is used and the impulse response is integrated backwards (Schroeder, 1965). The result is theoretically equivalent to an infinite number of averaged decays obtained by the interrupted noise method.

3.6 Combined Direct and Diffuse Fields

At a point in the room both the direct and diffuse fields exist and contribute to the value of the mean-squared sound pressure at that point. The mean-squared sound pressures of the direct and diffuse fields are scalar quantities and can be added, thus,

$$\overline{p^2} = \overline{p_d^2} + \overline{p_r^2} \quad (3.15)$$

$$= \frac{W \rho_0 c_0 Q_{axis}}{4\pi r^2} + \frac{4W \rho_0 c_0}{R_c} \quad (3.16)$$

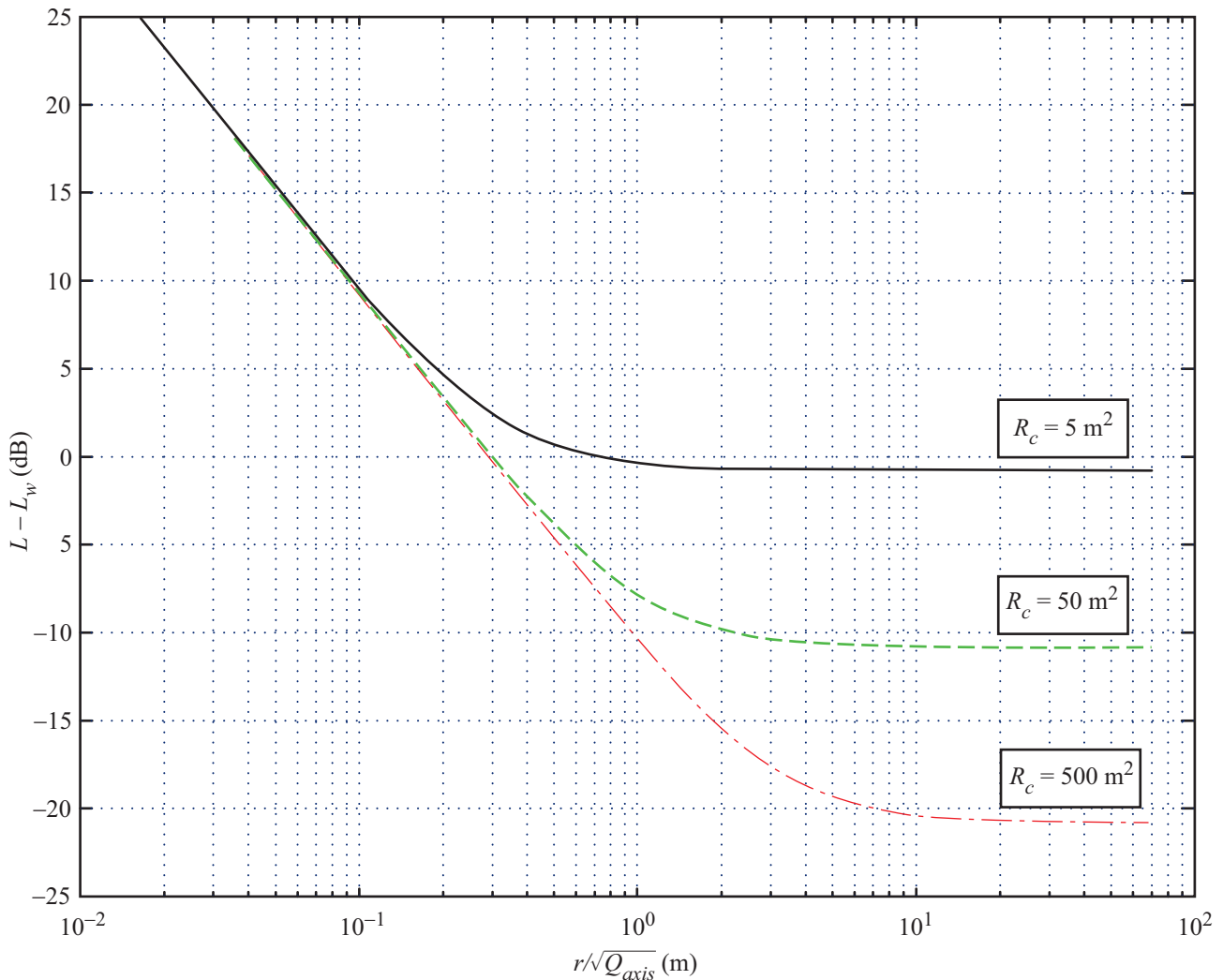
Equation (3.16) may be expressed in logarithmic form in terms of the sound power level, L_w , of the source and the sound pressure level, L , of the combined field, as

$$L = L_w + 10 \log \left(\frac{Q_{axis}}{4\pi r^2} + \frac{4}{R_c} \right) + 10 \log \left(\frac{\rho_0 c_0}{400} \right) \quad (3.17)$$

The sound power level in decibels is relative to 10^{-12} W and the sound pressure level is relative to 20 μ Pa. The value of the specific acoustic resistance $\rho_0 c_0$ for air in a room at 20° C and normal atmospheric pressure is 415 kg m⁻² s⁻¹, hence Equation (3.17) becomes:

$$L = L_w + 10 \log \left(\frac{Q_{axis}}{4\pi r^2} + \frac{4}{R_c} \right) + 0.2 \text{ dB} \quad (3.18)$$

The difference between the sound pressure level and the sound power level as a function of distance $r/\sqrt{Q_{axis}}$ is shown in Sketch 3.4 for different room constants. The direct and diffuse sound fields are equal when $r = 0.25 \sqrt{R_c/\pi}$ for $Q_{axis} = 1$. The distance where the field are equal is sometimes called the reverberation distance r_h (Pierce, 1989; Kuttruff, 1991).



Sketch 3.4 Difference between sound pressure level and sound power level as a function of $r/\sqrt{Q_{axis}}$

4. SPECIAL ACOUSTIC ROOMS

It can be seen from Sketch 3.4 that there are two extreme situations which are of interest. Firstly, when R_c is large, the inverse square law is obeyed for large distances from the source, *i.e.*, absorption of the walls is high. As indicated in Sketch 3.3, this region indicates the direct field. Secondly, when R_c is small, reflections are dominant and a constant reverberant sound level persists for most parts of the room. For the first condition an anechoic room is achieved, whereas the second condition leads to a reverberation room (Bies, 1976).

4.1 Anechoic Rooms

Anechoic rooms are special rooms for acoustical measurements in which the walls of the room are perfectly absorbing, *i.e.*, the surfaces of the room have an average sound absorption coefficient which is ideally equal to unity. The name anechoic derives from the fact that no echoes exist in the room. In an anechoic room a direct or free field is simulated, so there are no reflections and the situation is similar to that achieved in the open air. In a semi-anechoic room (sometimes called a hemi-anechoic room) the floor has no sound absorption and is replaced by a hard, reflecting surface, which makes the room useful for tests on machinery,

such as trucks and aero-engines (Cooper, 1994; Moreland, 1989). Anechoic rooms are used to measure the sound power of sources and to investigate their directional characteristics. Anechoic rooms are sometimes used for direct field acoustic tests (DFAT). In DFAT testing loudspeakers are arranged around the test article so that it is subjected to a direct sound field. Overall sound pressure levels up to 145 dB can be achieved. In theory a special room is not required, but it is useful to have the test carried out in a confined space. A small chamber with noise-absorbent walls has been designed and used for testing small satellites at sound levels up to 142 dB (Saggini *et al.*, 2012). Further details of anechoic rooms are found in Appendix A.

5. REVERBERATION ROOMS

Reverberation rooms are specially constructed chambers with highly reflecting walls. The interior surfaces are generally concrete or steel. Reverberation rooms are used for:

- (a) Measurement of the sound power of sources
- (b) Measurement of sound absorption coefficients
- (c) The high intensity testing of structures and electronic components for accreditation in relation to acoustic fatigue or flight validation
- (d) Measurement of sound reduction indices and coupling loss factors (two reverberation rooms are required)

In a reverberation room the diffuse field predominates, apart from a small region around the noise source. Thus, in Equation (3.16) the term representing the diffuse field is greater than that for the direct field, *i.e.*,

$$\frac{4}{R_c} > \frac{1}{2\pi r^2},$$

for the case of a sound source on a hard, reflecting surface. If the effect of the direct field is negligible, Equation (3.18) becomes

$$L = L_w + 10 \log \frac{4}{R_c} + 0.2 \text{ dB.} \quad (5.1)$$

Equation (5.1) may be used to estimate L_w , the sound power level of the source, from one measurement of L , the sound pressure level. The room constant, R_c , must be known and is estimated from the reverberation time, T , and the volume of the chamber, V . In a reverberation room the average sound absorption coefficient, $\bar{\alpha}$, is very small and the room constant may be approximated to $S\bar{\alpha}$. Thus, from Equation (3.11)

$$R_c \approx S\bar{\alpha} = 0.161 V/T, \quad (5.2)$$

for a temperature of 20° C and standard pressure.

5.1 Acoustic Design of Reverberation Rooms

In many cases the internal space of a reverberation room is a rectangular parallelepiped (right cuboid). There are many normal modes of vibration in such a space. The values of the natural frequencies for the higher modes are very close together. Additionally, the modes are distributed in many different directions in space. Thus, a diffuse field is easily achieved at high frequencies. More specifically, it has been shown that, where there are at least 20 to 30 modes in a measurement band, such as a one-third octave band, a diffuse field is achieved (Nelisse and Nicolas, 1997). At low frequencies, however, the natural frequencies are relatively widely separated. This can cause a problem in the case of a sound source which radiates at discrete frequencies, because the frequency of the tones from the source may fall between two natural frequencies of the chamber, and the room modes cannot be excited by the source. For each mode there is

a maximum in the response, the sharpness of which depends upon the absorption of the acoustic waves at the boundaries of the room. Thus, for a large value of the reverberation time the modes have the appearance of a sharp peak. As the frequency increases, the number of modes and natural frequencies within a frequency band become more numerous. The maxima begin to merge together and the statistical parameters of the room depend upon only the volume and the reverberation time. The frequency at which the maxima merge together has been referred to as either a critical frequency (Schroeder and Kuttruff, 1962) or a cutoff frequency (Pearce, 1989). An approximate expression for this frequency, f_c , at room temperature has been given as (Schroeder and Kuttruff, 1962; Pearce, 1989)

$$f_c = 2000(T_{60}/V)^{1/2}, \quad (5.3)$$

where V is the volume of the room in m^3 . The cutoff frequency depends upon c_0 , which is assumed to be 344 m/s in the derivation of Equation (5.3).

Most of the guidelines on the design of reverberation rooms refer to the measurement of sound power and absorption coefficients. A typical volume for a reverberation room for sound power measurement is 200 m^3 . Chambers for testing satellites are often larger. A chamber with a volume of 200 m^3 can be used for frequencies of 100 Hz and above. For rooms with a smaller volume, the lowest operational frequency is greater than 100 Hz. Small rooms can be used at higher frequencies. Over the frequency range of interest, the surfaces in the reverberation room within one wavelength of the noise source should have an absorption coefficient less than 0.06 (ISO 3741, 2010). The surfaces of the room should have absorptive properties with the room empty such that the inequality

$$T_{60} > V/S$$

is satisfied, where V is the volume of the room and S is the total surface area.

Table 5.1 shows the lowest frequency of interest for different reverberation room volumes (ISO, 2010).

TABLE 5.1 Recommended minimum volume of reverberation room as a function of the lowest frequency band of interest (ISO 3741)

Lowest one-third octave band frequency of interest	Minimum volume of reverberation room
Hz	m^3
100	200
125	150
160	100
>200	70

5.1.1 Frequency spacing

Modal density of acoustic spaces has been considered in an ESDU Item (ESDU, 2014), where the model of Blevins has been applied to obtain the cumulative number of modes below a frequency f (Blevins, 2006). The results obtained by the model were compared with the averaged modal density derived by Maa (Maa, 1939). The concept of the averaged modal density given by Maa, combined with the definition of modal density as the inverse of the averaged distance between neighbouring natural frequencies, leads to the averaged frequency spacing:

$$\bar{\delta} = \left(\frac{4\pi f^2 V}{c_0^3} + \frac{\pi f S}{2c_0^2} + \frac{L_{per}}{8c_0} \right)^{-1}, \quad (5.4)$$

where V is the volume of the room, S is the surface area, L_{per} is the perimeter and c_0 is the velocity of sound. Certain chamber shapes can be selected to ensure that the natural frequencies are distributed as uniformly as possible especially at low frequencies. The rms deviation from the averaged frequency spacing is a useful measure of the merit of a reverberation room. The quantity, d_{rms} , is defined (Louden, 1971) as:

$$d_{rms} = \left(\frac{1}{N} \sum (\delta - \bar{\delta})^2 \right)^{1/2} \quad (5.5)$$

where δ is the actual spacing between frequencies and N denotes the number of spacings in the considered band.

The achieve improvements in frequency spacing several other measures have been introduced (Bolt, 1946; Maling, 1971; Maling, 1985). Sepmeyer devised the r.m.s. spacing index (Sepmeyer, 1965):

$$I = \left[\frac{1}{N} \sum \left(\frac{\delta}{\bar{\delta}} \right)^2 \right]^{1/2} \quad (5.6)$$

The average spacing index is defined as (Maling, 1971)

$$I_{av} = \frac{1}{N} \sum \frac{\delta}{\bar{\delta}} \quad (5.7)$$

A recommended set of proportions is $4^{1/3} : 2^{1/3} : 1$ (ISO, 2010). For rooms with a volume of about 200 m^3 the standard (ISO 3741, 2010) provides five other sets of recommended ratios, which are given in Table 5.2. The ratio provided by the standard have been modified to be consistent with the notation used in this Item. Included in the table are the corresponding values of d_{rms} and I , which have been specially calculated for this Item. Some other room proportions are also given.

TABLE 5.2 Recommended room dimension ratios for right cuboid rooms

	L_x/L_z	L_y/L_z	d_{rms}	I
ISO 3741	2.1277 (1.00/0.47)	1.7660 (0.83/0.47)	1.7536	1.2866
ISO 3741	1.5385 (1.00/0.65)	1.2769 (0.83/0.65)	1.3626	1.0804
ISO 3741	1.5873 (1.00/0.63)	1.2540 (0.79/0.63)	1.4520	1.1407
ISO 3741	2.3810 (1.00/0.42)	1.6190 (0.68/0.42)	1.2470	1.1767
ISO 3741	1.6949 (1.00/0.59)	1.1864 (0.70/0.59)	1.5596	1.1940
Louden, 1971	1.9	1.4	1.1459	1.1578
Milner, 1989	1.439	1.186	1.0314	1.1106
ESDU	2.1375	1.475	0.9637	1.1138

Louden has shown that the ratio which gives the least irregularity in modal spacing is 1.9: 1.4: 1 (L_x/L_y , L_x/L_z , 1) (Louden, 1971). His investigation was based on 36 modes and he used a criterion to minimise d_{rms} . Milner and Bernhard similarly considered 36 modes and proposed a slightly improved ratio of room dimensions, see Table 5.2. To achieve a uniform distribution of frequency spacing at low frequencies no two dimensions of the room should be in the ratio of integers (ISO 354, 2003). It is suggested that the length of the longest diagonal which fits within the bounds of the room should be less than $1.9 V^{1/3}$ (for a rectangular room it is the major diagonal) (ISO 354).

Figures 5.1 and 5.2 refer to the space formed by the chamber ratios ($r_{zx} = L_x/L_z$, $r_{zy} = L_y/L_z$). The contours of r.m.s. deviation, d_{rms} , and rms spacing index, I , are shown, respectively, in the diagrams. Both measures are applied in this Item to 36 modes.