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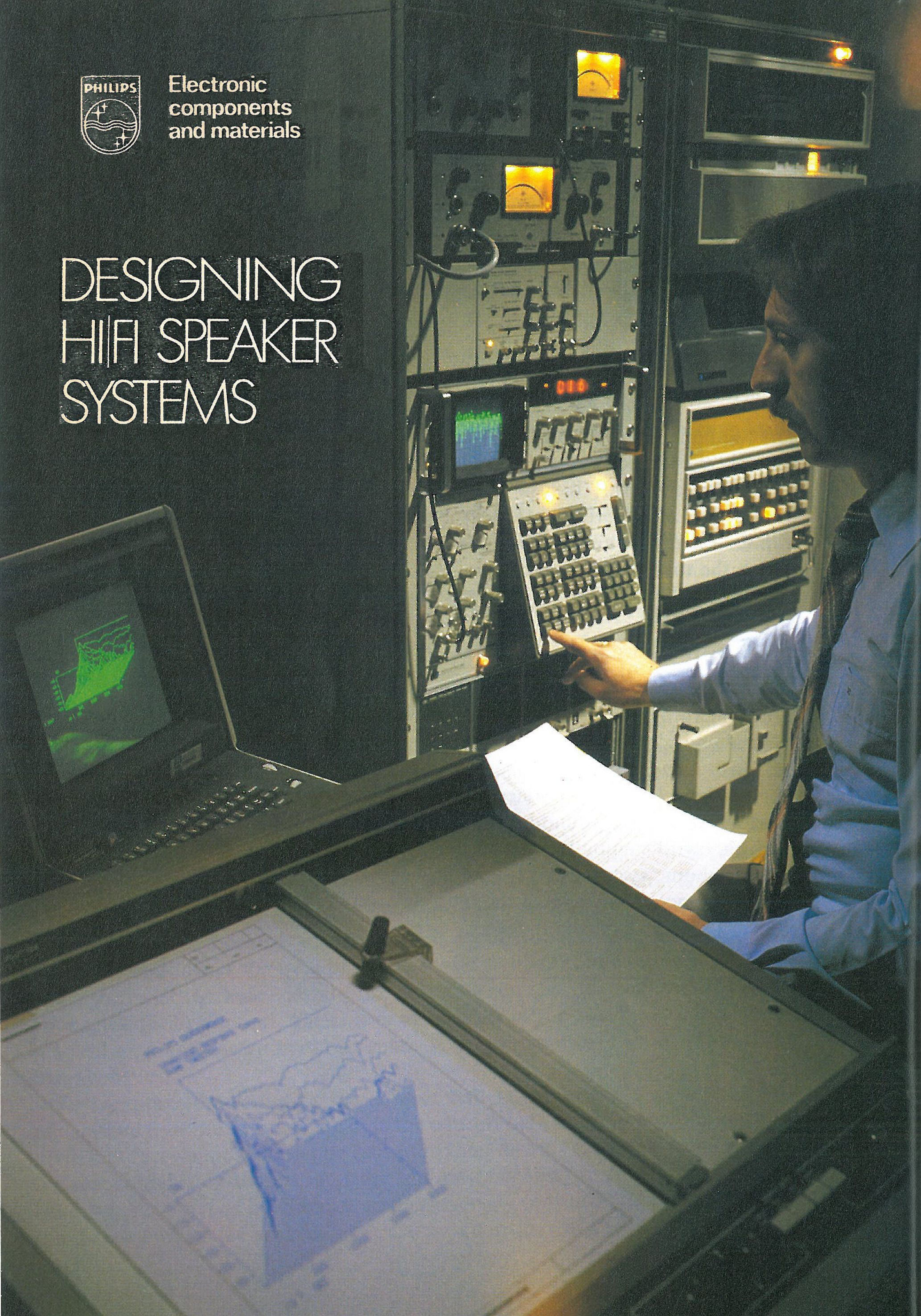
DESIGNING HI/FI SPEAKER SYSTEMS



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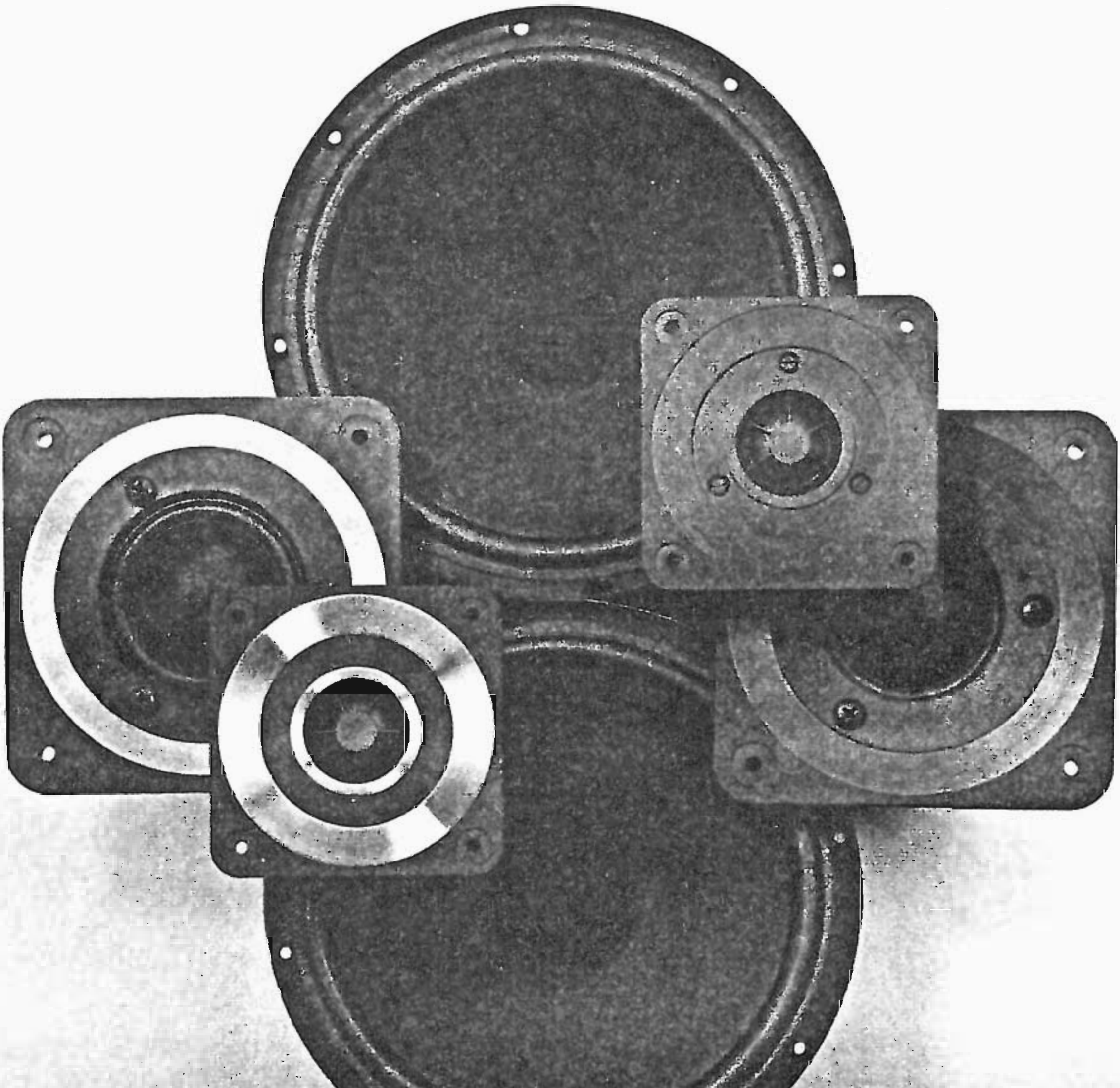

Designing Hi-Fi Speaker Systems

D. Hermans and M. D. Hull

ELECTRONIC COMPONENTS AND MATERIALS DIVISION
MARKETING COMMUNICATIONS

Contents

Designing hi-fi speaker systems - part 1

D. Hermans* and M. D. Hull**

This is the first of a series of articles on sealed loudspeaker enclosure systems. Section 1 contains the derivations of formulae used later in the three articles. Such material would normally be included as an appendix but is presented first because of the serialization of the total article. Section 2 analyses the mechanical and electrical design of the moving coil direct radiator loudspeaker. Further articles in the series will cover the design of sealed enclosure systems, cross-over filter networks, specifications, measurements and listening room acoustics.

Introduction

Prior to 1925, the power output of most radio sets was only a few milliwatts. The sets used valves which were very inefficient and, since the power was supplied from batteries, the current drain had to be kept to a minimum.

Early in the 1920s, C. W. Rice and E. W. Kellogg of the General Electric Company, U.S.A., designed what was then something quite unheard of – a one watt, mains driven power amplifier. With this 'powerful' tool at hand, they set to work on a new type of loudspeaker; one that did not rely on resonances to give sufficient output. In 1925, they described the results of their work and went on the market with the first moving coil loudspeaker as we know it today, complete with its power amplifier.

The Rice-Kellogg loudspeaker had a 6-inch diameter cone and a rubber surround. The magnetic field was provided by a direct current flowing in a coil.

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It was soon realized that the same kind of direct current used to supply the loudspeaker field winding could also be applied to power the radio set and the all-mains radio receiver was born.

Later the Marconi Company patented the idea of using the field coil as the smoothing choke in the receiver HT supply line but, when efficient permanent magnets became available, the days of the mains-energized loudspeaker were numbered.

Despite 50 years of scientific and technical progress in all aspects of sound reproduction, the moving coil loudspeaker remains substantially the same today as it was in 1925 when Rice and Kellogg disclosed the results of their work. Materials and methods of production have changed, and so has the performance. There have been many improvements with different sizes, different impedances, extended frequency ranges and less distortion. But, after half-a-century of research and development, no satisfactory alternative has yet been devised. For all the other refinements in audio engineering, the last remaining stumbling block to the ultimate goal of perfect sound reproduction is the moving coil loudspeaker.

There can be few homes in the civilized world that cannot boast at least one loudspeaker. Some have three or four. Even cars have stereo. There are literally millions and millions of loudspeakers. It would therefore seem surprising that there is anything more to be said about them. However, circumstances change; over-population and the preference for smaller, easy-to-run households has brought with it a size reduction in housing accommodation. Smaller furniture and, consequently, more compact domestic sound installations are a necessity. The modern requirement is for small boxes; the smaller, the better. But, at the same time, a more affluent society demands higher quality reproduction, and is prepared to pay for it. So, even with a 50-year-old component, there are still new developments.

Reducing the size of a speaker enclosure means reducing its efficiency at low frequencies. More power input is needed to obtain the same loudness and, as perfection is the goal, there must be higher power amplifiers to drive the speakers. Thanks to semiconductor developments, amplifiers with output powers undreamed of a few years ago can now be accommodated in small attractive units in keeping with modern decor. The simplest way to make a good loudspeaker enclosure is to make a closed box. Its cost is reasonable and, in general, this method of mounting has proved very acceptable to both manufacturer and customer alike. What is most important technologically is that the performance can be carefully controlled.

1 The wave equation and acoustic elements

1.1 Sound propagation

Consider a plane surface vibrating in air. As it moves forward from rest, the surface will accelerate adjacent particles and compress the air just in front of it. The accelerated particles collide with their neighbours and in that way transfer momentum so that the compression is propagated outwards. When the vibrating surface reverses its motion a rarefaction occurs in front of it. Air particles move in to fill the void and are followed in turn by more remote particles so that the rarefaction, like the compression, is also propagated outwards from the surface. The outward moving alternation of compression and rarefaction is a sound wave. The wave is propagated at a certain speed and each particle in the medium moves with the same frequency as the vibrating surface. The wavelength λ, frequency f, and speed of propagation c are related by

$$\lambda = c/f.$$

At sound frequencies there is no time for heat exchange, so the pressure alterations are essentially adiabatic and the medium behaves according to the equation,

$$PV^{\gamma} = \text{a constant} \qquad (1.1)$$

where,

P = pressure

V = volume

and γ is the ratio of specific heat at constant pressure to specific heat at constant volume. For air, $\gamma \approx 1{,}4$.

1.2 Wave equation

We shall now derive the wave equation for sound propagation in tubes or horns. Although the derivation is based on the assumption of small diameter compared to wavelength, the resulting equation also holds for plane or spherical free progressive waves (a spherical wave may be regarded as being propagated in a number of conical horns radiating from a common centre).

1.2.1 EQUATION OF MOTION

In Fig 1.1 we have a volume of air in a conical horn. Sections S_1 and S_2 are perpendicular to the direction of sound propagation, x. When dx tends to 0, $S_1 = S_2 = S$; the enclosed air mass is then given by

$$M = \varrho S dx$$

where ϱ is density. The force on the left side of S is PS and the force on the right side is

$$F_{RHS} = \{P + (\partial P/\partial x)dx\}S,$$

so the resulting force acting on the mass M is

$$F = -\frac{\partial P}{\partial x} dx\, S.$$

Then, from $F = Ma = M\, dv/dt$, we can derive:

$$-\frac{\partial P}{\partial x} dx\, S = \varrho S\, dx \frac{dv}{dt}$$

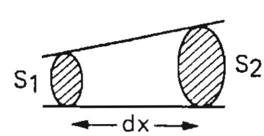


Fig. 1.1

whence

$$\frac{\partial P}{\partial x} + \varrho \frac{dv}{dt} = 0. \qquad (1.2)$$

This also holds if sections S_1 and S_2 are unequal for then the greater force on the larger section is balanced by the horizontal component of the force on the conical sides (see Fig. 1.2).

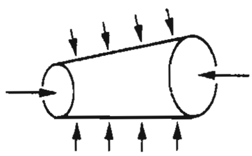


Fig. 1.2

1.2.2 CONTINUITY EQUATION

During time increment dt the mass displacement on the left side is vS, and on the right side $\{v + (\partial v/\partial x)dx\}\ \{S + (\partial S/\partial x)dx\}$. Since no mass can be lost between S_1 and S_2, the volume $V(= S dx)$ must change at a rate

$$\frac{\partial V}{\partial t} = \left(S \frac{\partial v}{\partial x} + v \frac{\partial S}{\partial x} \right) dx. \qquad (1.3)$$

1.2.3 GAS LAWS

By differentiating eq. (1.1) with respect to time we get

$$\gamma P V^{\gamma - 1} \frac{\partial V}{\partial t} + V^{\gamma} \frac{\partial P}{\partial t} = 0$$

in which $P = P_0 + p$, where P_0 is static pressure and p is alternating (sound) pressure. Thus we can write

$$\frac{\partial P}{\partial t} = \frac{\partial p}{\partial t} \quad \text{and} \quad P \approx P_0$$

whence

$$\frac{\partial V}{\partial t} = - \frac{V}{\gamma P_0} \frac{\partial p}{\partial t} \qquad (1.4)$$

Combining eq. (1.3) and eq. (1.4) gives,

$$- \frac{V}{\gamma P_0} \frac{\partial p}{\partial t} = - \frac{S\,dx}{\gamma P_0} \frac{\partial p}{\partial t} = \left(S \frac{\partial v}{\partial x} + v \frac{\partial S}{\partial x} \right) dx$$

whence

$$\frac{1}{\gamma P_0} \frac{\partial p}{\partial t} + \frac{\partial v}{\partial x} + \frac{v}{S} \frac{\partial S}{\partial x} = 0$$

and

$$\frac{1}{\gamma P_0} \frac{\partial p}{\partial t} + \frac{\partial v}{\partial x} + v \frac{\partial (\ln S)}{\partial x} = 0. \qquad (1.5)$$

Differentiating eq. (1.5) with respect to time gives,

$$\frac{1}{\gamma P_0} \frac{\partial^2 p}{\partial t^2} + \frac{\partial^2 v}{\partial x \partial t} + \frac{\partial v}{\partial t} \cdot \frac{\partial (\ln S)}{\partial x} = 0.$$

Differentiating eq. (1.2) with respect to x, and making the substitution $\partial P = \partial p$, gives,

$$\frac{\partial^2 p}{\partial x^2} + \varrho \frac{\partial^2 v}{\partial x \partial t} = 0.$$

Combining these two expressions then yields the wave equation,

$$\frac{\varrho}{\gamma P_0} \frac{\partial^2 p}{\partial t^2} = \frac{\partial^2 p}{\partial x^2} + \frac{\partial p}{\partial x} \cdot \frac{\partial (\ln S)}{\partial x}$$

in which the factor $\varrho / \gamma P_0 = 1/c^2$, where c is the propagation speed. Thus,

$$\frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} = \frac{\partial^2 p}{\partial x^2} + \frac{\partial p}{\partial x} \cdot \frac{\partial (\ln S)}{\partial x}. \qquad (1.6)$$

We now introduce the velocity potential Φ such that

$$-\bar{v} = \frac{\partial \Phi}{\partial x} \bar{i} + \frac{\partial \Phi}{\partial y} \bar{j} + \frac{\partial \Phi}{\partial z} \bar{k}$$

where $\bar{v}$ is the velocity vector and $\bar{i}$, $\bar{j}$ and $\bar{k}$ are unit vectors in the x, y and z directions. Thus, for the one-dimensional case,

$$v = - \frac{\partial \Phi}{\partial x} \bar{i}$$

and eq. (1.6) can be written in the form

$$\frac{1}{c^2}\frac{\partial^2 \Phi}{\partial t^2} = \frac{\partial^2 \Phi}{\partial x^2} + \frac{\partial \Phi}{\partial x}\frac{\partial (\ln S)}{\partial x}.$$

If the horn is perfectly rigid, S is not a function of time and we can make the substitution $\partial(\ln S)/\partial x = \mathrm{d}(\ln S)/\mathrm{d}x$; thus,

$$\frac{1}{c^2}\frac{\partial^2 \Phi}{\partial t^2} = \frac{\partial^2 \Phi}{\partial x^2} + \frac{\partial \Phi}{\partial x}\frac{\mathrm{d}(\ln S)}{\mathrm{d}x}. \tag{1.7}$$

Furthermore, from eq. (1.2)

$$\frac{\partial P}{\partial x} = -\varrho\frac{\partial v}{\partial t}$$

which, for the one-dimensional case, we can write in the form

$$\frac{\partial P}{\partial x} = -\varrho\left(-\frac{\partial^2 \Phi}{\partial x \partial t}\right)$$

whence

$$P = \varrho\frac{\partial \Phi}{\partial t}. \tag{1.8}$$

1.3 Mechanical and acoustic elements

1.3.1 ACOUSTIC IMPEDANCE

Mechanical impedance Z_M is defined by the relation

$$Z_M = \frac{F}{v}.$$

This can be represented either by an impedance circuit (Fig. 1.3) or a mobility circuit (Fig. 1.4).

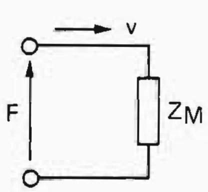


Fig. 1.3

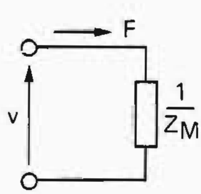


Fig. 1.4

In acoustics we are more interested in pressures and volume velocities than in forces and particle velocities. Acoustic impedance Z_A is therefore defined by the relation

$$Z_A = \frac{p}{U}$$

where p is sound pressure and U is volume velocity. Thus the acoustic impedance presented by a force F acting on an area S moving at a velocity v is given by,

$$Z_A = \frac{F/S}{vS} = \frac{Z_M}{S^2}. \tag{1.9}$$

Acoustic impedance can also be represented by either an impedance circuit (Fig. 1.5) or a mobility circuit (Fig. 1.6).

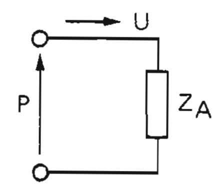


Fig. 1.5

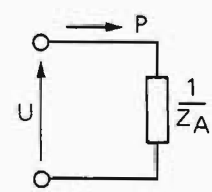


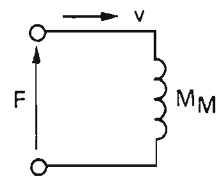
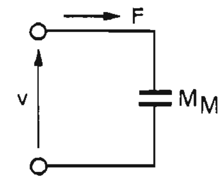
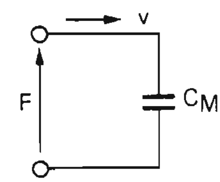
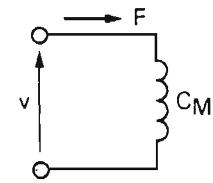
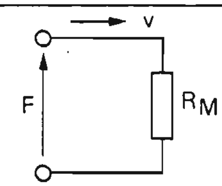
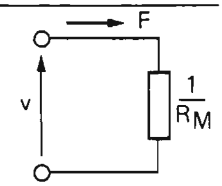
Fig. 1.6

It is sometimes convenient to consider the mechanical or acoustic impedance per unit area. This is called specific impedance, Z_s.

$$Z_s = \frac{p}{v} = \frac{Z_M}{S} = Z_A S. \tag{1.10}$$

1.3.2 MECHANICAL/ELECTRICAL ANALOGY

The table on p.5 illustrates the analogy between the elements that contribute to mechanical or acoustic impedance and their electrical counterparts.

element	equation of motion: general	equation of motion: harmonic	symbol: impedance	symbol: mobility
mass, M	$F = Ma = M\frac{dv}{dt}$	$F = Mj\omega v$		
compliance, C_M ($C_M = 1/k$, where k = spring constant)	$F = kx = \frac{x}{C_M} = \frac{1}{C_M}\int v\,dt$	$F = \frac{v}{j\omega C_M}$		
resistance, R_M (viscous friction)	$F = R_M v$	$F = R_M v$		

1.3.3 ACOUSTIC MASS

The air in a small diameter open-ended tube (Fig. 1.7) behaves as a mass; it is very stiff compared to the air outside and can only be accelerated, not compressed. The equation of motion is therefore,

$$F = Mj\omega v$$

where

$$F = pS, \quad M = lS\varrho, \quad v = \frac{U}{S},$$

whence

$$p = \frac{l\varrho}{S} j\omega U. \qquad (1.11)$$

In this expression the term $l\varrho/S$ corresponds to the acoustic mass M_A; thus,

$$p = M_A j\omega U.$$

(Acoustic mass is proportional to the mass M of air undergoing acceleration and is, ideally, given by the equation $M_A = M/S^2$. In practice, corrections must be applied to take account of how the tube is terminated. If the tube ends in free air, the term l in eq. (1.11) becomes $l + 0{,}33\sqrt{S}$; if it ends in an infinite baffle, l becomes $l + 0{,}45\sqrt{S}$.)

1.3.4 ACOUSTIC COMPLIANCE

If the tube ends in a closed volume (Fig. 1.8), the air in the volume will behave as a compliance; it can only be compressed, not accelerated. Therefore, eq. (1.1) applies:

$$PV^\gamma = \text{a constant}$$

Fig. 1.7

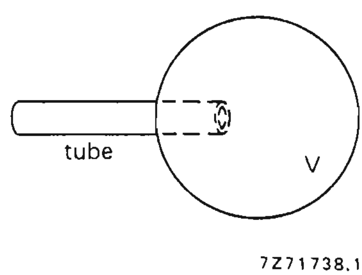


Fig. 1.8

whence

$$dP = -\frac{\gamma P_0}{V}\,dV$$

where

$$dP = \text{sound pressure} = p$$

$$dV = \text{volume displacement} = \frac{U}{j\omega}.$$

Thus

$$p = \frac{\gamma P_0 U}{j\omega V}$$

and the acoustic impedance

$$Z_A = \frac{p}{U} = \frac{\gamma P_0}{j\omega V}$$

in which the factor $\gamma P_0/V$ corresponds to a spring constant. Thus the acoustic compliance is

$$C_A = \frac{V}{\gamma P_0}$$

or, since $\gamma P_0 = \varrho c^2$,

$$C_A = \frac{V}{\varrho c^2} \tag{1.12}$$

1.3.5 ACOUSTIC RESISTANCE

Any device in which the flow of air is in phase with, or directly proportional to, the applied pressure may be represented as an acoustic resistance. Examples are fine-mesh screens, small-bore tubes, narrow slits, porous materials.

1.4 Radiation impedance

1.4.1 QUALITATIVE DESCRIPTION

Consider a small, harmonically pulsating sphere. When the radius is maximum, the velocity of the surface and the air close to it is zero and the alternating pressure is maximum. When the radius is midway between the maximum and minimum, the surface velocity is maximum and the alternating pressure is zero. Thus, the sound pressure p leads the velocity by 90° (see Fig. 1.9). The radiation impedance in this case is purely reactive and can be regarded as a pure mass that has to be moved to and fro.

Now consider a pulsating sphere, or a vibrating plane surface, which is large compared to the wavelength. The impedance of a given particle is no longer purely reactive, for it is influenced by neighbouring particles that are also vibrating and affecting the local pressure. To find the radiation impedance we have to divide the resulting pressure by the volume velocity. But, as the resulting pressure is mostly caused by neighbouring particles, it will be lagging in time due to the distance it has to travel (see Fig. 1.10). The impedance is no longer purely reactive, but has a resistive component. If the dimensions are very large, the impedance is purely resistive. This resistive component accounts for the radiation of acoustic energy.

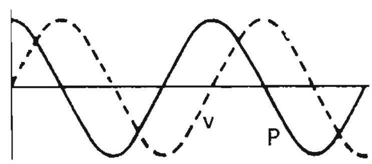


Fig. 1.9

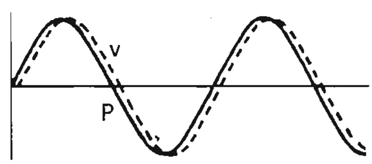


Fig. 1.10

1.4.2. MATHEMATICAL ANALYSIS

Consider a pulsating sphere of a radius r and surface area $S = 4\pi r^2$. In eq. (1.7) we can replace x by r, whence

$$\frac{d(\ln S)}{dx} = \frac{d(\ln S)}{dr} = \frac{1}{S}\frac{dS}{dr} = \frac{2}{r}$$

and the wave equation becomes

$$\frac{1}{c^2}\frac{\partial^2 \Phi}{\partial t^2} = \frac{\partial^2 \Phi}{\partial r^2} + \frac{2}{r}\frac{\partial \Phi}{\partial r}$$

Then, if we change the dependent variable from Φ to Φ_r, we obtain

$$\frac{1}{c^2}\frac{\partial^2(\Phi_r)}{\partial t^2} = \frac{\partial^2(\Phi_r)}{\partial r^2}$$

for which the general solution is

$$\Phi_r = A\,e^{j(\omega t - kr)} + B\,e^{j(\omega t + kr)}$$

where the first term represents the outgoing wave, the second term the returning or reflected wave, and the quantity k is referred to as the wave number.

$$k = \frac{\omega}{c} = \frac{2\pi}{\lambda}.$$

For a free, progressive, spherical wave

$$\Phi = \frac{A}{r}\,e^{j(\omega t - kr)}.$$

We are now in a position to calculate the mechanical, specific, or acoustic radiation impedance. First, the specific radiation impedance Z_{sA}:

$$Z_{sA} = \frac{p}{v}$$

in which $v = -\partial\Phi/\partial r$ and, from eq. (1.8), $p = \varrho\partial\Phi/\partial t$. Thus,

$$Z_{sA} = \frac{\varrho\partial\Phi/\partial t}{-\partial\Phi/\partial t} = \frac{\varrho j\omega\Phi}{\left(\frac{1}{r} + jk\right)\Phi}$$

$$= \varrho c\,\frac{jkr}{1 + jkr} \qquad (1.13)$$

(see Fig. 1.11).
Then, since the mechanical radiation impedance

$$Z_{MA} = S\,Z_{sA},$$

$$Z_{MA} = 4\pi r^2\varrho c\,\frac{jkr}{1 + jkr}.$$

The corresponding mobility circuit is given in Fig. 1.12. The acoustic radiation impedance Z_{AA} is given by

$$Z_{AA} = \frac{p}{U}\,\frac{Z_{MA}}{S^2}.$$

Equation (1.13) confirms the conclusion that we previously arrived at qualitatively; that the radiation impedance has both a resistive and a reactive component.
- For $kr \ll 1$, the impedance behaves as a mass.
- For $kr \gg 1$, the impedance behaves as a resistance.

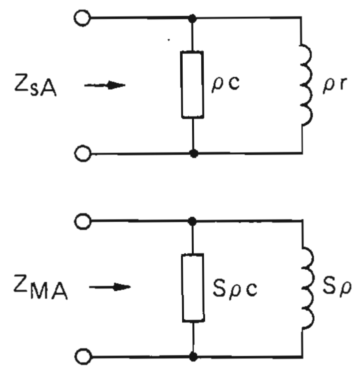


Fig. 1.11

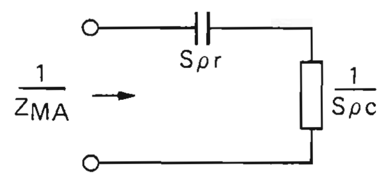


Fig. 1.12

1.4.3 SERIES/PARALLEL SUBSTITUTION

Parallel circuits, such as those of Fig. 1.11 can be replaced by series circuits as in Fig. 1.13. The impedance of the parallel circuit is given by

$$Z_{par} = \frac{R_{par}\,j\omega L_{par}}{R_{par} + j\omega L_{par}}$$

and of the series circuit by

$$Z_{ser} = R_{ser} + j\omega L_{ser}.$$

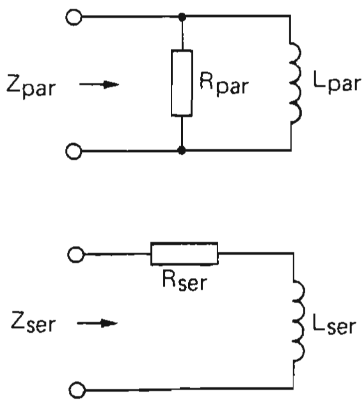


Fig. 1.13

Rationalizing the denominator in the expression for Z_{par} and separating terms gives

$$Z_{par} = \frac{R_{par}\,\omega^2 L^2_{par}}{R^2_{par} + \omega^2 L^2_{par}} + j\omega \frac{R^2_{par} L_{par}}{R^2_{par} + \omega^2 L^2_{par}}$$

where the first term is the resistive component and the second term the reactive component. Thus, since $Z_{par} \equiv Z_{ser}$, we can write,

$$R_{ser} = \frac{R_{par}\,\omega^2 L^2_{par}}{R^2_{par} + \omega^2 L^2_{par}}$$

and

$$L_{ser} = \frac{R^2_{par} L_{par}}{R^2_{par} + \omega^2 L^2_{par}}.$$

If, for example, we transform the parallel circuit representation of Z_{sA} (Fig. 1.11) into its series equivalent (Fig. 1.14), we find:

$$R = \varrho c \frac{(kr)^2}{1 + (kr)^2} \quad \text{and} \quad M_r = \frac{\varrho r}{1 + (kr)^2}.$$

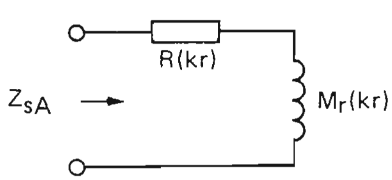


Fig. 1.14

1.4.4 LOUDSPEAKER

The mechanical radiation impedance of a loudspeaker cone is $Z_{MA} = F/v$ where F is the force necessary to overcome the resistance of the air and v is the velocity of the cone. For purposes of analysis the cone may be regarded as a piston.

Analysis of a flat, rigid piston oscillating in an infinite baffle or an endless tube is much more difficult than analysis of a pulsating sphere. Therefore we shall only summarize the results.

- For $kr \gg 1$, Z_A is the same as for the pulsating sphere and $Z_{sA} = \varrho c$.
- For all other values of kr, both the resistive and the reactive components of Z_A differ from those that apply to the pulsating sphere.
- For $kr \ll 1$, the principal component of Z_A is an air mass M_{MA}:
  for an infinite baffle, $M_{MA} = 0{,}85\, S\varrho r$
  for an endless tube, $M_{MA} = 0{,}6\ \ S\varrho r$

where r is the radius of the piston and $S = \pi r^2$. In the mobility circuit for a piston of radius r in an infinite baffle, as given in Fig. 1.15,

$$M_M = 0{,}85\,\pi r^3 \varrho$$

$$R_1 = \frac{2{,}27}{\pi r^2 \varrho c} \qquad R_2 = \frac{1}{\pi r^2 \varrho c}$$

$$C_M = \frac{0{,}6}{r \varrho c^2}$$

where the resistive component, which accounts for the sound radiation, is of the most interest.

- For $kr \gg 1$

$$\frac{1}{Z_{MA}} \approx R_1 = \frac{1}{\pi r^2 \varrho c}$$

and hence $Z_{MA} = S\varrho c$.

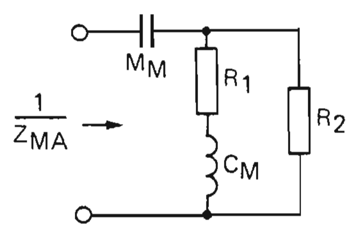


Fig. 1.15

– But for $kr < 1$, which characterizes the frequency range in which loudspeakers mostly work,

$$R_{1,2} = \frac{R_1 R_2}{R_1 + R_2} \approx \frac{0{,}7}{\pi r^2 \varrho c} = \frac{0{,}221}{r^2 \varrho c}.$$

The equivalent mobility circuit is given in Fig. 1.16 and the equivalent impedance circuit in Fig. 1.17. The latter can be transformed, as explained in Section 1.4.3, into the series circuit of Fig. 1.18, in which

$$R_{ser} = 1{,}57\,\omega^2 r^4 \frac{\varrho}{c}$$

$$M_{Mser} = 2{,}67\, r^3 \varrho$$

where R_{ser} is directly proportional to the radiated acoustic power W_A:

$$W_A = R_{ser}\, v^2. \qquad (1.14)$$

1.5 Sound intensity

The sound intensity I in a given direction is the rate of energy transmission through a unit area normal to that direction. It is commonly expressed in watts per square metre or per square centimetre. Mathematically, it is the product of sound pressure and the in-phase component of particle velocity:

$$I = \text{Re}\, p^* v \cos\theta$$

where

p^* is the complex conjugate of the r.m.s. sound pressure,
v is the complex r.m.s. particle velocity in the direction of propagation,
θ is the angle between the direction of propagation and the direction at which the intensity is taken,
and Re denotes the real part of the product.

For a free progressive plane or spherical wave, the intensity in the direction of propagation ($\theta = 0$) is

$$I = \frac{p^2}{\varrho c}. \qquad (1.15)$$

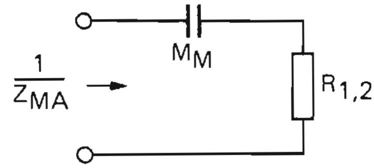


Fig. 1.16

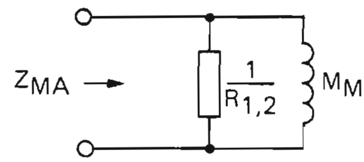


Fig. 1.17

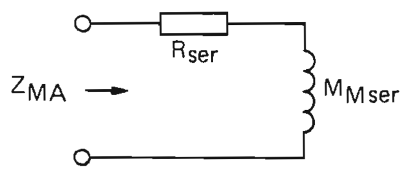


Fig. 1.18

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1.6 Power level, intensity level and sound pressure level

These are convenient ratios for comparing a measured or calculated power, intensity, or sound pressure with a fixed reference value. They are expressed in dB and are defined as follows.

$$\text{Power level, PWL} = 10 \log_{10} \frac{W}{W_{ref}}$$

where W is the measured or calculated power and W_{ref} is usually taken as 10^{-12} watts.

$$\text{Intensity level, IL} = 10 \log_{10} \frac{I}{I_{ref}}$$

where $I_{ref} = 10^{-12}$ W/m^2.

$$\text{Sound pressure level, SPL} = 10 \log_{10} \frac{p^2}{p^2_{ref}}$$

$$= 20 \log_{10} \frac{p}{p_{ref}}$$

where $p_{ref} = 10^{-5}$ N/m^2.

Under normal conditions ($\varrho = 1{,}18$ kg/m^3, $c \approx 344$ m/s), intensity level and sound pressure level are very nearly equal: IL $\approx$ SPL $- 0{,}1$ dB.

2 The moving coil direct radiator loudspeaker

2.1 Principles of operation

All moving coil loudspeakers operate according to the same general principles. Figure 2.1 shows the method of construction of a typical moving coil direct radiator loudspeaker in which the amplifier output signal is fed to the voice coil suspended in the gap of a powerful magnet system by means of a centring device, or 'spider'. The small end of the speaker cone is attached to one end of the coil assembly and the big end is attached to a frame by means of a flexible surround. Since the current flowing in the coil produces a magnetic field at right angles to the field of the magnet, the attraction and repulsion of these fields causes the coil to move to and fro at the frequency of the electrical signal. The oscillating motion of the cone to which the coil is attached produces sound waves in the surrounding air.

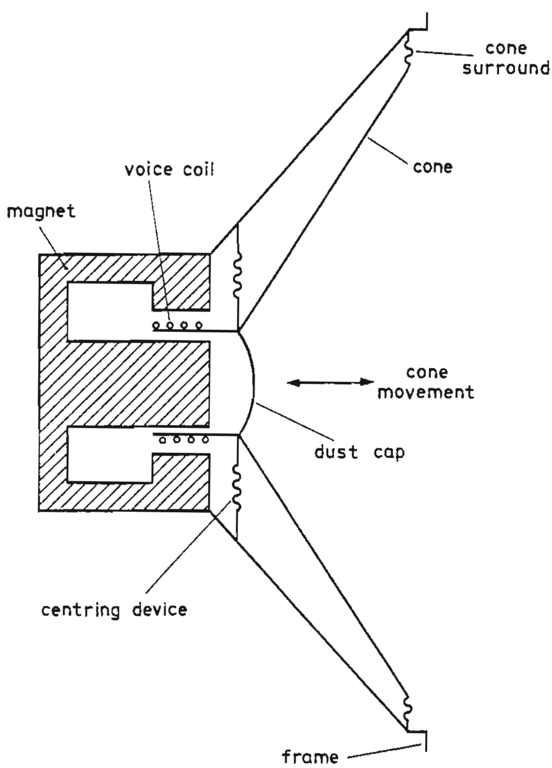


Fig. 2.1. The construction of a typical moving coil loudspeaker.

2.2 Two-stage energy conversion

To simplify the explanation of loudspeaker performance we shall treat the loudspeaker as a two-stage energy converter. Electrical energy is first converted into mechanical energy, and the mechanical energy produced is then converted into acoustic energy, as shown in Fig. 2.2. This conversion of energy should be carried out with the minimum of distortion and with maximum efficiency. To obtain maximum transfer of energy from the power amplifier to the speaker, it is desirable to match the impedances. Unfortunately, this can only be achieved over a limited frequency range due to the variation in the impedance presented to the amplifier by the loudspeaker.

If we assume that the loudspeaker is mounted in an infinite baffle, the radiation impedance Z_{AA} is the same for both sides of the baffle.

The circuit of Fig. 2.2 may be changed into one of the circuits shown in Fig. 2.3. In Fig. 2.3(*a*) and Fig. 2.3(*b*) all components are transformed to the electrical side. In Fig. 2.3(*c*) the components are referred to the mechanical side. We have, therefore, changed the electrical voltage source (e_g) with the electrical series impedance (Z_e) into a current source (e_g/Z_e) with a parallel impedance (Z_e). This principle may be used for calculations on the right side of these terminals. As a last step, we have changed the mobility circuit into an impedance circuit.

2.3 Radiation resistance

In our discussion of loudspeaker performance it will be assumed that the loudspeaker is mounted in an infinite baffle; no back radiation can therefore affect the forward radiation. Under these conditions, the air load on the cone appears as a mechanical impedance Z_{MA}. The subscript MA is used to refer to 'mechanical, air'. The mechanical impedance is represented by the radiation resistance R_{MA} and the radiation reactance X_{MA} in series. In practice, for $kr < 1$, the radiation reactance is the mass of the air load and will be referred to here as the radiation mass M_{MA}. We can therefore write:

$$Z_{MA} = R_{MA} + j\omega M_{MA}. \tag{2.1}$$

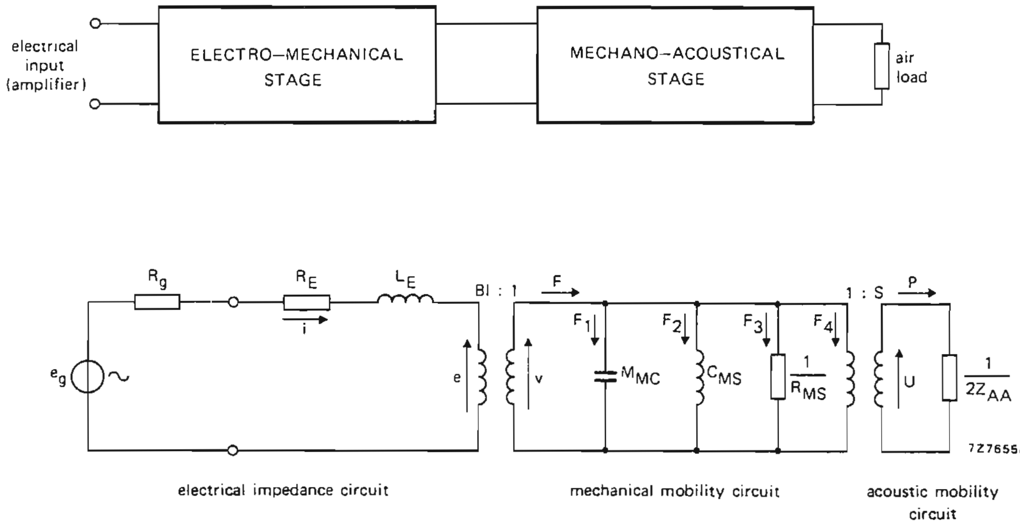


Fig. 2.2. The loudspeaker as a two-stage energy converter.

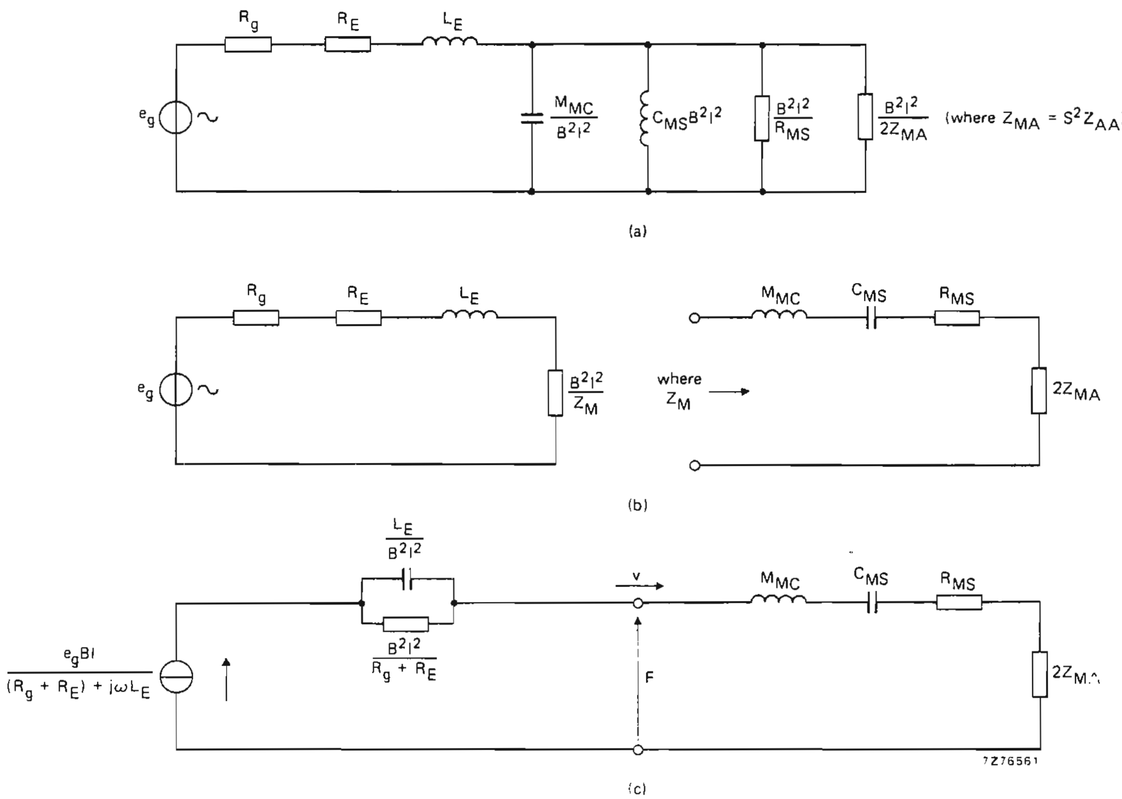


Fig. 2.3. Equivalent circuits of a moving coil loudspeaker.

It is of special interest to consider how the radiation resistance behaves, since it is this component of the radiation impedance in which the acoustic power is developed. In moving coil systems, the radiation mass M_{MA} is often neglected, since it appears in series with and is very much less than the mechanical mass of the cone M_{MC}. Figure 2.4 shows how the radiation resistance and radiation reactance vary with frequency. It can be shown (see Section 1.4.4) that the value of R_{MA} over the sloping portion of the curve is given by

$$R_{MA} = \left(\frac{1,57\omega^2 r^4 \varrho}{c}\right) \text{Ns/m (mechanical ohms)} \quad (2.2)$$

where: $\omega = 2\pi f$ (f = frequency in Hz),
r = cone radius in metres,
ϱ = density of the air = 1,18 kg/m^3
and c = velocity of sound = 344 m/s.

This is valid for $kr < 1$, which is the most important frequency region for most loudspeakers.

The radiation reactance X_{MA} is given by

$$X_{MA} = j\omega M_{MA} \quad (2.3)$$

where the radiation mass M_{MA} is

$$M_{MA} = 2,67r^3\varrho. \quad (2.4)$$

The units used in Fig. 2.4 may require some explanation. Frequency is plotted on a normalized scale, the horizontal axis representing frequency in terms of the relationship between the dimensions of the cone and the wavelength of the sound:

$$kr = 2\pi fr/c = 2\pi r/\lambda,$$

that is, the ratio of the loudspeaker circumference to the wavelength.

Up to the point where $kr = 2$, the radiation resistance increases according to eq. (2.2) proportionally with f^2. For values of $kr > 2$, however, the situation changes and eqs (2.2) and (2.3) no longer apply. The radiation resistance R_{MA} is then given by

$$R_{MA} = \pi r^2 \varrho c \quad (2.5)$$

and the radiation resistance is independent of frequency.

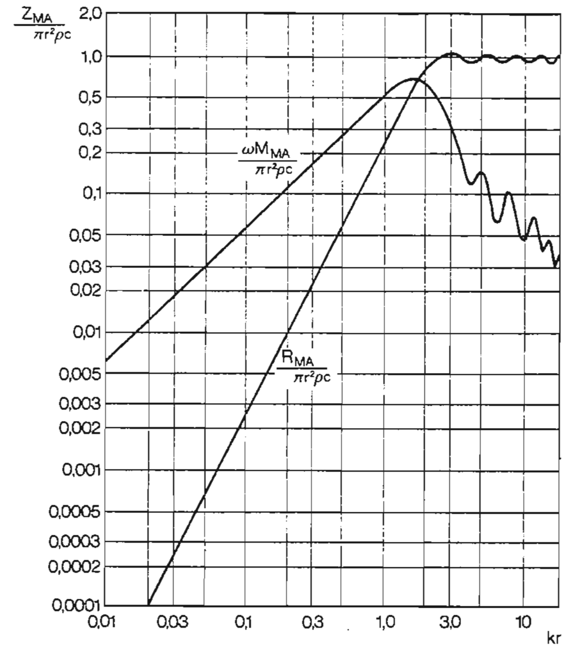


Fig. 2.4. Real and imaginary parts of the normalized mechanical impedance of the air load on one side of a plane piston of radius r mounted in an infinite baffle.
$kr = 2\pi fr/c = 2\pi r/\lambda$.

2.4 Relationship between mechanical and electrical impedances

The mechanical impedance of the cone assembly is represented in Fig. 2.3(*b*) by the mass of the total cone assembly M_{MC}, the compliance of the suspension C_{MS}, and some frictional losses R_{MS} occurring mainly in the suspension. However, the most important resistive component is the resistance of the voice coil R_E which acts in series with its inductance L_E and the output resistance of the amplifier.

From basic principles, the induced emf in the voice coil is given by

$$e = Blv \quad (2.6)$$

where e is the induced emf in the coil, in volts,
B is the radial flux density in the gap, in teslas (Wb/m^2),
l is the length of the wire on the voice coil, in metres,
and v is the voice coil velocity, in m/s.

Since the voice coil velocity can be written as

$$v = \frac{F}{Z_M}$$

where F is the force in newtons ($= Bli$) and Z_M is the mechanical impedance, we can write

$$v = \frac{Bli}{Z_M} \text{ m/s}$$

and eq.(2.6) becomes

$$e = \frac{B^2l^2i}{Z_M}, \qquad (2.7)$$

where i is the current in amperes in the voice coil. Equations (2.6) and (2.7) assume that the magnetic induction is constant over the whole voice coil length and independent of the position of the coil in the air-gap. This is not strictly true, and the length and position of the coil with regard to the air-gap is very important when considering distortion. So we define

$$Bl = \int_0^1 B\mathrm{d}l$$

at the position of rest, which remains valid for small movements of the coil.

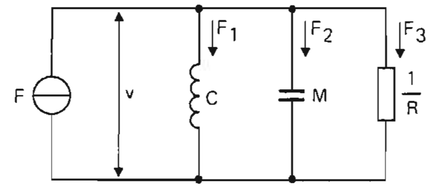
The electrical impedance Z_E due to the mechanical impedance Z_M is therefore given by

$$Z_E = \frac{e}{i} = \frac{B^2l^2}{Z_M}$$

$$Z_E = B^2l^2 \frac{1}{R_{MS} + j\left(\omega M_{MC} - \dfrac{1}{\omega C_{MS}}\right)}. \qquad (2.8)$$

The mechanical impedance of a system composed of a mass M, a compliance C and a mechanical resistance R can be represented by an equivalent impedance or mobility circuit, as in Fig. 2.5. In the impedance circuit

$$v = \frac{F}{j\omega M + (1/j\omega C) + R}$$



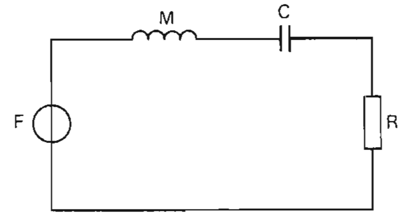


Fig. 2.5. Impedance and mobility circuits of a simple vibrating system.

and in the mobility circuit

$$F = F_1 + F_2 + F_3$$

$$= \frac{v}{j\omega C} + vj\omega M + vR = v\left(\frac{1}{j\omega C} + j\omega M + R\right).$$

In both circuits the source of the force F is assumed to be constant, but the relations also hold for a constant velocity source.

There is an interaction between electrical and mechanical impedance. The mechanical impedance will be reflected in the electrical impedance and vice versa, e.g. the cone velocity will be influenced by the electrical resistance of the voice coil in such a way that a decrease of voice coil resistance will result in an increase of effective mechanical resistance to the cone.

2.5 Effect of mechanical impedance on acoustic power

In order to determine the behaviour of a loudspeaker it is necessary to examine the effect of each of the components of the mechanical impedance. For simplicity, let us consider the equivalent circuit of the loudspeaker as being reduced

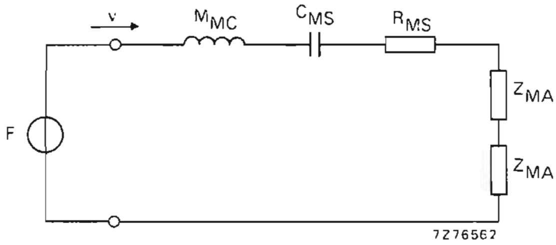


Fig. 2.6. Mechanical components determining cone velocity.

to the components shown in Fig. 2.6. This circuit is valid for a constant force source or constant current source ($F = Bli$). The overall mechanical impedance of the cone is considerably greater than that of the air load, so the current will be determined almost entirely by the cone. Let us examine the effect of each of the mechanical components assuming that the cone is perfectly rigid.

(*a*) Assume the mass impedance to be predominant. Then we may write

$$v = \frac{F}{\omega M_{MC}} = \frac{F}{2\pi f M_{MC}} \tag{2.9}$$

where F is the applied force.

Now the radiated power on one side of the infinite baffle, which is developed in the radiation resistance R_{MA}, is given by the mechanical equivalent of Ohm's Law:

$$W_A = v^2 R_{MA} \tag{2.10}$$

and substituting eq. (2.9) for v in eq. (2.10) gives

$$W_A = \frac{F^2}{4\pi^2 f^2 M_{MC}^2} R_{MA}. \tag{2.11}$$

From eq. (2.2), we have seen that $R_{MA} \propto f^2$ over the sloping part of the curve below $kr \approx 2$, so from eq. (2.11),

$$W_A \propto \frac{1}{f^2} \cdot f^2. \tag{2.12}$$

That is, the radiated power is independent of frequency. On the flat portion of the curve of Fig. 2.4, where $kr > 2$, from eq. (2.5) we see that R_{MA} is constant, therefore

$$W_A = \frac{1}{f^2} \times \text{a constant} \tag{2.13}$$

and hence the acoustic power falls at the rate of 6 dB/octave. This represents the condition of *mass control* and is shown in Fig. 2.7. This result is largely valid for a constant voltage source as in Fig. 2.3(*c*).

(*b*) Assume the resistance R to be predominant. With high damping factors, the resistance $R = R_{MS}$ may be in control for a current source, or the resistance $R = R_{MS} + B^2 l^2 / R_E$ may be in control for a constant voltage source. In these cases,

$$W_A = \frac{F^2}{R^2} R_{MA} \tag{2.14}$$

and since, from eq. (2.2), $R_{MA} \propto f^2$,

$$W_A = \text{a constant} \times f^2. \tag{2.15}$$

That is, the radiated power increases with frequency at 6 dB/octave below where $kr = 2$.

Over the flat portion of the curve where $kr > 2$,

$$W_A = \text{a constant} \times \text{a constant}. \tag{2.16}$$

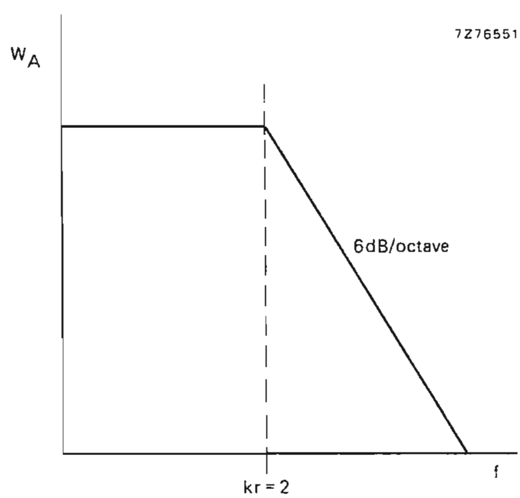


Fig. 2.7. Condition of mass control.

That is, the radiated power is independent of frequency. This condition is shown in Fig. 2.8 and is known as *constant velocity.*

(*c*) Assume the compliance to be predominant. If the compliance of the suspension C_{MS} is in control, we can write,

$$W_A = F^2 4\pi^2 f^2 C^2_{MS} R_{MA}, \tag{2.17}$$

and thus over the sloping part

$$W_A \propto f^2 f^2 = f^4 \tag{2.18}$$

That is, the radiated power increases with frequency at 12 dB/octave.

Over the horizontal portion of the curve above $kr = 2$,

$$W_A = f^2 \times \text{a constant}. \tag{2.19}$$

That is W_A rises at 6 dB/octave. This is the condition of *compliance control* as shown in Fig. 2.9, a situation not normally encountered.

From the foregoing it can be seen that to maintain a constant radiated power a number of possibilities exist. Most electro-dynamic loudspeakers operate in the mass control region, extended as far as possible. To extend this region in the low frequency range, the fundamental resonance frequency

$$f_0 = \frac{1}{2\pi\sqrt{(M_{MC} C_{MS})}}$$

must be kept as low as possible. Below f_0 we have compliance control:

$$\frac{1}{j\omega C_{MS}} > j\omega M_{MC}$$

and above f_0 we have mass control:

$$j\omega M_{MC} > \frac{1}{j\omega C_{MS}}.$$

To extend this region into the upper frequency range there are ways of circumventing the $kr = 2$ limit. For full-range loudspeakers we can use flexible cones that tend to break up starting from the frequency at which $kr = 2$. They then have the natural tendency to reduce their effective diameter as the frequency rises. Another technique is to use cross-over filters to bring into operation progressively smaller loudspeakers as the frequency rises.

Horn loudspeakers mostly operate in the constant velocity region (high damping factors) because, due to the horn, the radiation resistance R_{MA} will be constant at much lower frequencies.

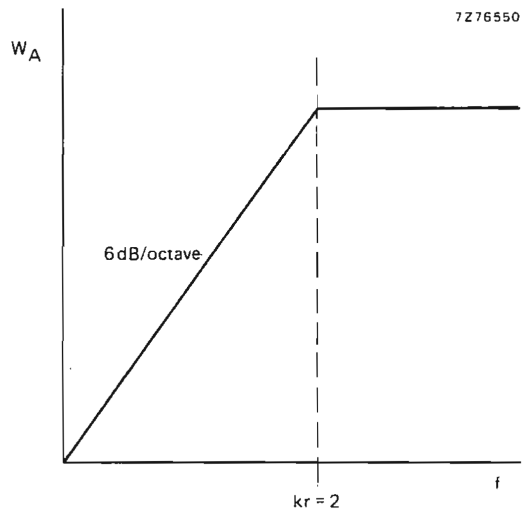


Fig. 2.8. Condition of constant velocity.

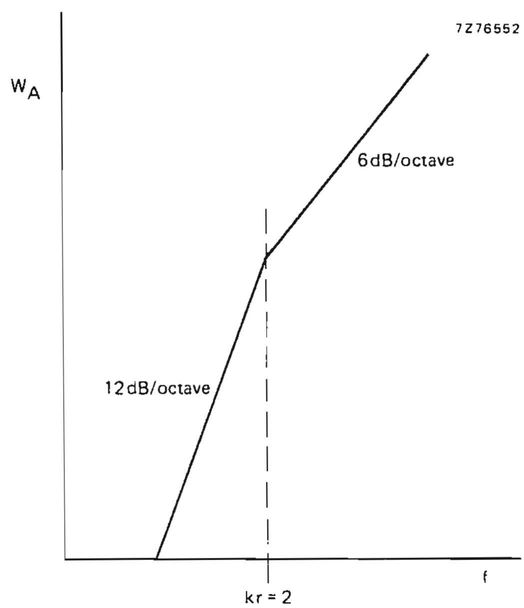


Fig. 2.9. Condition of compliance control.

2.6 Frequency response characteristics

We are now in a position to study the performance of a loudspeaker over the entire audio range. Let us divide the frequency spectrum into four parts as shown in Fig. 2.10.

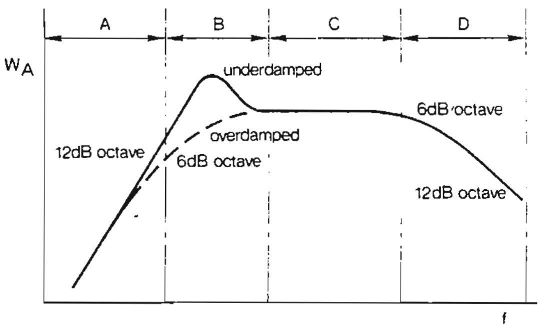


Fig. 2.10. Power output of a loudspeaker in an infinite baffle, assuming operation as a rigid piston.

At very low frequencies, the compliance is the controlling factor and the radiated power is proportional to f^4, i.e. it rises at 12 dB/octave with frequency (region A in Fig. 2.10).

At the fundamental resonant frequency the net mechanical reactance is zero since the mass reactance of the cone equals the stiffness reactance of the suspension, and the mechanical impedance is minimum; the speaker thus behaves as a series *LCR* circuit. The electrical impedance, however, rises to the maximum of a parallel *LCR* circuit. The question of damping arises now, and if the speaker is highly damped the voice coil current and the coil velocity are nearly constant. The variation of acoustic power is proportional to the radiation resistance, i.e. from eq. (2.2), to the square of the frequency. Region B in Fig. 2.10 shows the conditions at resonance.

From frequencies above resonance, up to where $kr > 2$, the net acoustic reactance is predominantly due to the mass of the cone, rather than to the air coupled to it, and from eq. (2.12) we see that the acoustic output is independent of frequency. Constant power is thus radiated over this mass-controlled region (region C in Fig. 2.10).

For still higher frequencies, where $kr \gg 2$, the condition described by eq. (2.13) applies. The radiated power falls initially at 6 dB/octave, tending towards a slope of 12 dB/octave due to the rise in the effective inductive reactance of the voice coil as the frequency increases. This is shown as region D in Fig. 2.10.

The foregoing considerations apply to the radiated acoustic power W_A. If we consider the sound pressure, the conclusions will remain the same for region A, B and a great part of region C. In these regions, the polar diagram of a loudspeaker in a infinite baffle is almost a hemisphere because the dimensions of the loudspeaker cone are small compared with the wavelength. The sound source may be considered as a point source. For high frequencies (upper part of region C and all of region D) the sound will be radiated principally in the direction of the axis. For a flat, rigid piston, the on-axis sound pressure is independent of frequency in regions C and D, but for a conical loudspeaker of depth H it starts to decrease at the frequency at which $kH = 2$ (see Fig. 2.11). Pressure waves originating from different parts of the cone have different distances to travel to reach a given point on the axis and therefore tend to cancel each other.

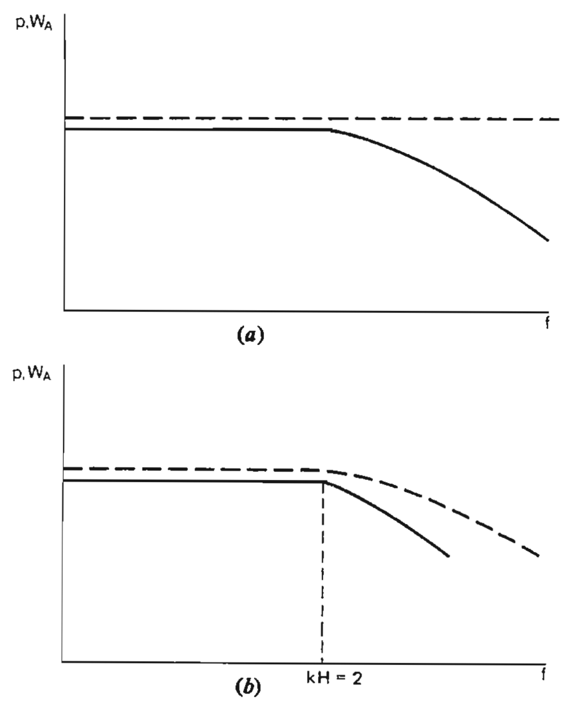


Fig. 2.11. On axis sound pressure (dashed line) and sound power (full line) radiated by: (*a*) a rigid piston in an infinite baffle; (*b*) a rigid cone in an infinite baffle.

2.7 Damping and Q-factor

The parts of a loudspeaker which actually convert the electrical energy into mechanical energy are the voice coil and the magnet system. One interesting characteristic is that the magnet and voice coil system behaves like a transformer having a turns ratio of $Bl : 1$, where B is the magnetic flux density in the air-gap and l is the length of the wire (see Fig. 2.2). Another characteristic of the system is that it inverts impedances; the mechanical damping resistance R' is related to the electrical resistance R_E by

$$R' = \frac{B^2 l^2}{R_E} . \qquad (2.20)$$

Inductances in series on one side appear as parallel capacitances on the other, and vice versa. This is the basic difference between Fig. 2.3(*a*) and Fig. 2.3(*c*). By way of a practical example, if the electrical impedance at some low frequency is noted and then the cone is touched, thus reducing its motion, the electrical impedance will decrease as a result of the increase in mechanical impedance.

Considering the transducing element as an impedance matching component, Bl will have an optimum value at some low frequency. The value is normally arranged so that the cone maintains a condition of mass control down to the frequency of resonance. At resonance, where the mass reactance of the cone equals the stiffness reactance of the suspension, a Q of something less than unity, say 0,5, is preferred since this gives a truly non-oscillatory condition and consequently the best transient performance. The mechanical circuit Q is given by:

$$Q_M = \frac{2\pi f M_{MC}}{R_{MS}} . \qquad (2.21)$$

If the resistance is predominantly due to the voice coil, we may write,

$$R' = \frac{B^2 l^2}{R_E}$$

and, putting R' for R_{MS} in eq. (2.21), we obtain the electrical Q

$$Q_E = \frac{2\pi f M_{MC} R_E}{B^2 l^2} \qquad (2.22)$$

whence

$$Bl = \sqrt{\frac{2\pi f M_{MC} R_E}{Q_E}} . \qquad (2.23)$$

The total Q-factor, Q_T, is given by

$$\frac{1}{Q_T} = \frac{1}{Q_M} + \frac{1}{Q_E} . \qquad (2.24)$$

The coil and magnet system are designed using this expression.

The acceleration of a vibrating mass-spring-resistance system as a function of frequency varies with Q in the same way as the sound pressure curves of a loudspeaker in an infinite baffle. It would seem advantageous therefore to choose Q values of about unity in order to have the most extended flat frequency response. But, as we know from mechanical vibration theory, $Q = 0,5$ corresponds to the critically damped condition. Higher values give an oscillatory motion which impairs transient response.

In calculating the optimum value of Bl the electrical resistance of the voice coil, R_E, has been used without considering the output resistance of the amplifier, R_g, which is in series with it (see Fig. 2.3(*a*)). Since R_g is normally many times smaller than R_E a high damping factor is easily achieved. (Damping factor is the ratio of load impedance to source impendance.) With modern solid-state power amplifiers a damping factor as high as 200 is not unusual. In view of the low internal resistance of the amplifier it is important that the resistance of the speaker cables does not significantly reduce the damping factor.

An interesting consequence of the effect of source resistance is shown in Fig. 2.12. Two curves are shown of the response of a typical 5″ woofer in a 7-litre box full of glass wool. One curve shows the response with a constant voltage input, the other with a constant current input. The constant voltage condition corresponds to a source resistance of zero, whereas in the constant current condition the source resistance can be taken as infinity. The effect of varying the source resistance between zero and infinity is clearly shown, a high Q resulting in the case of the high source resistance. Since a modern solid-state amplifier offers a low source resistance to the speaker, and corresponds to a nearly constant voltage generator, the underdamped condition shown in Fig. 2.12 does not normally arise (assuming that the effect of the speaker cables can be neglected).

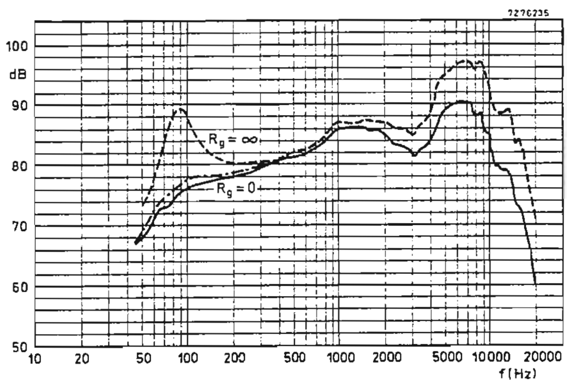


Fig. 2.12. The effect of varying source resistance on loudspeaker response. Dashed line, constant current condition, $R_g = \infty$; full line, constant voltage condition, $R_g = 0$; chain dotted line, response with typical amplifier.

2.8 Loudspeaker cones

Over the sloping part of the R_{MA} curve it can be reasonably assumed that the cone works substantially as a rigid piston. This is not the case at higher frequencies where the side of the cone becomes comparable to, or longer than, one wavelength, and we have to consider the longitudinal wave propagation in the cone material. The cone then moves with different amplitudes over different parts of its surface and it is this property that enables a single cone loudspeaker to operate over a wide frequency range instead of falling at 12 dB/octave as was shown for a rigid piston in Fig. 2.7.

Assuming that the wave is attenuated as it travels up the cone, it can be seen that the effective diameter of the cone decreases as the frequency increases. Above the knee of the R_{MA} curve, R_{MA} is proportional to the square of the effective cone diameter (see eq. (2.5)). Since the mass of the cone, M_{MC}, is also proportional to the square of the diameter, we can determine the radiated power W_A from eq. (2.9) and eq. (2.10):

$$W_A = v^2 R_{MA} \propto \frac{1}{d^4} d^2 = \frac{1}{d^2}. \qquad (2.25)$$

At higher frequencies therefore the smaller effective cone diameter tends to increase the radiated power, thus offsetting the condition shown in Fig. 2.7 for a rigid piston. The reduction in effective cone diameter is illustrated in Fig. 2.13. As the cone is more rigid at the apex than at the base, the longitudinal wave propagation is faster, and hence the wavelength greater, at the apex than at the base.

We can apply the same line of reasoning to the sloping part of the R_{MA} curve. Below the knee, R_{MA} is proportional to the fourth power of the cone diameter (see eq. (2.2)) and, since M_{MC} is proportional to the square of the diameter, we can write

$$W_A = v^2 R_{MA} \propto \frac{1}{d^4} d^4 = \text{a constant}. \qquad (2.26)$$

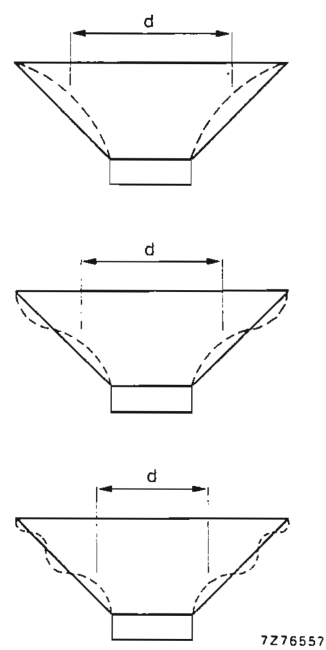


Fig. 2.13. The reduction in effective cone diameter with increasing frequency.

Hence, for a given applied force and a given cone material, the radiated power is independent of cone diameter at low frequencies.

In general, the loudspeaker cone needs to have a high stiffness-to-weight ratio and a reasonably high degree of internal friction. The cone should rapidly restore itself after the application of a waveform and, therefore, elasticity of the material is important. In this respect, metals are not particularly good. The effect of poor restoration is one of hysteresis and occurs where internal friction is high in comparison to stiffness. Hysteresis upsets both the frequency and transient response and produces distortion. Objective testing does not normally give any significant indication of distortion due to hysteresis in the cone material.

2.9 Cone surrounds

The requirements for a cone surround are that it must:

- provide a flexible support for the edge of the cone;
- provide a seal to the enclosure;
- be completely non-resonant;
- absorb the waves travelling up the cone at high frequencies.

Normally, the material of which the surround is made will be extremely soft and flexible and have high density and high internal friction. There can be few materials that have not been tried out to perform this exacting function. Acrylic coatings on polyurethane foam, textiles, and plasticized polyvinylchloride have all be used at one time or another. In the quest for greater power bandwidth, high efficiency, minimum distortion and better transient handling capability, many new materials will doubtless be tried in the future. One of the latest and most successful choices is butyl rubber.

2.10 Directivity

From the frequency response curve of Fig. 2.10, it can be seen that above a frequency where $kr \approx 2$ (normally between 700 and 3000 Hz) the acoustic output falls progressively. For a rigid piston the decrease is between 6 dB and 12 dB/octave. This is more apparent at the sides of the loudspeaker than on the axis, because of directivity, as shown in Fig. 2.14. Due to the 'horn' effect of the wider part of the cone, a larger proportion of the power output is directed along the axis of the cone at high freqencies than it is in other directions.

Directivity patterns are shown for a full-range loudspeaker in Fig. 2.15. Here the speaker is mounted only by a clamp on a turntable in an anechoic room and no baffle board is used. The output is recorded through a microphone as the loudspeaker is rotated. For the loudspeaker used, acoustic cancellation is observed at 90° and 270° rotation. Of special interest is the plot at 5000 Hz. This clearly indicates the directions in which the maximum sound output is projected at high frequencies.

At low frequencies, the acoustic output is largely omnidirectional. Coupled with reflections in the listening room, the effect of walls, floor and ceiling is to make the direction of the source almost indiscernible to the listener. But at a frequency between 10 000 Hz and 15 000 Hz, it should be expected that a full-range loudspeaker will maintain a level treble response at least to an angle of 15° off axis. In many cases, a level response can be obtained more than 30° off axis.

The upper limit of the treble response is obviously determined by the mass of the voice coil, which should be as low as possible in a full-range speaker. This may require the use of a very large magnet in which is immersed a very short coil.

2.11 Non-linearity and distortion

Amplitude distortion is caused by non-linearities in the cone suspension system and also by the cone itself. Additionally, lack of uniformity of the magnetic field as well as the change in the electrical inductance of the voice coil when it is moving, can be a cause of distortion.

The action of the suspension should be linear out to the maximum excursion of the cone, so that the cone motion is directly proportional to the force applied. With large cone movements this is sometimes difficult to achieve. Examples of cone motion are shown in Fig. 2.16.

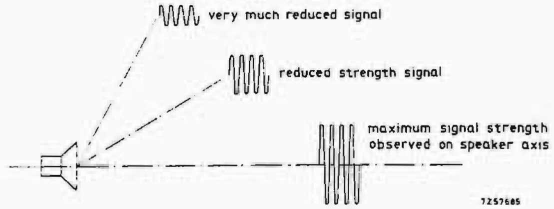
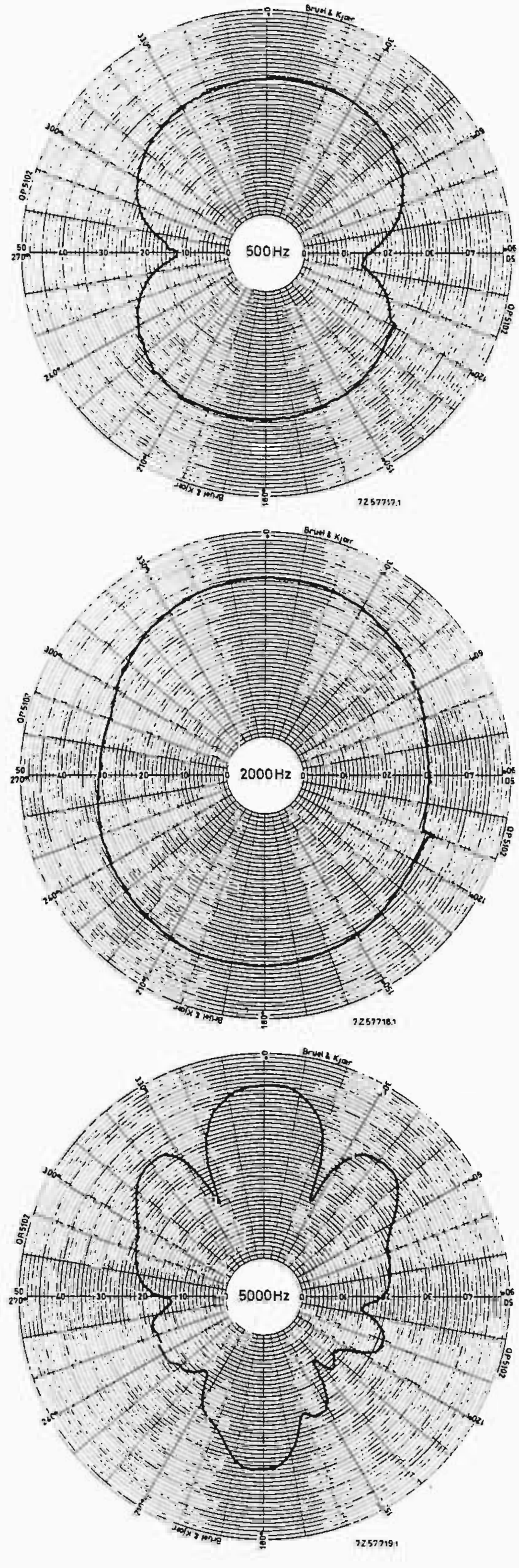


Fig. 2.14. The strength of high-frequency signals decreases at increasing angles to the axis of the loudspeaker.

Fig. 2.15. (Right-hand side of page) Polar response curves of a typical high-quality full-range loudspeaker at different frequencies. Note the fall in output at 90° and 270° on the 500 Hz curve due to acoustic cancellation.



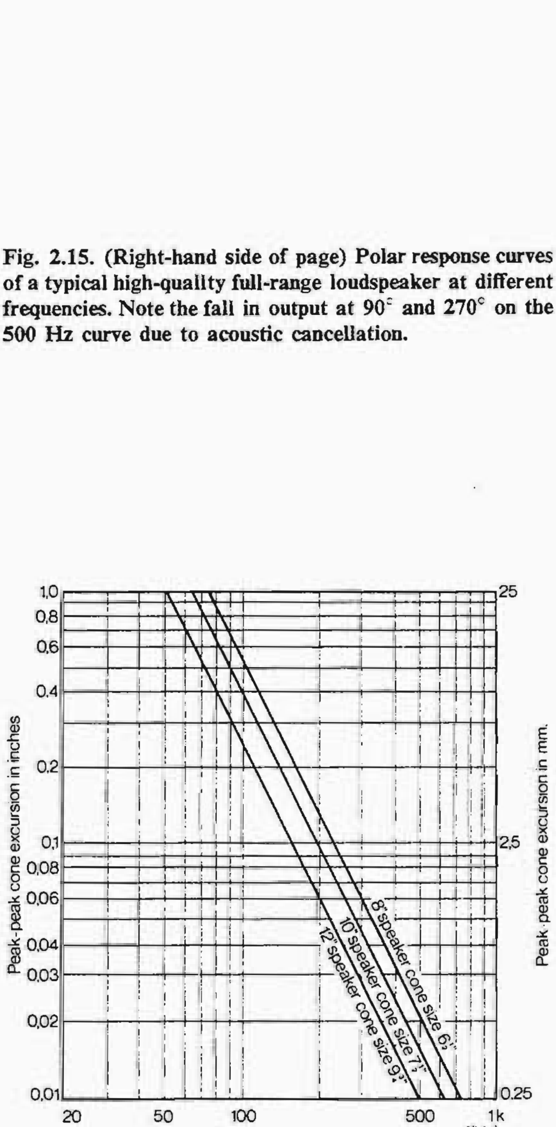


Fig. 2.16. Cone excursions of 8-inch, 10-inch and 12-inch loudspeakers mounted in an infinite baffle and radiating one acoustic watt on each side.

Most loudspeakers use paper pulp, moulded to shape, for the cone. This material can be considerably non-linear, especially if it is too thin. Doubling the cone thickness reduces the efficiency, but it also reduces distortion over most of the frequency range. The low-frequency distortion of a typical 12 inch full-range loudspeaker is shown in Fig. 2.17.

Unless the magnetic field in which the coil moves is uniform, the cone motion will not be uniform. Two methods are employed to overcome this non-linearity, as shown in Fig. 2.18. If a short coil is used, coil movement in the fringe area at the ends of the gap is avoided. If a long coil is used, one end of the coil moves into a region of high flux density as the other end moves into a region of low flux density. The product (turns × flux cut) remains constant.

So far we have been considering only amplitude distortion, but modulation distortion also occurs when a low tone, which gives rise to a large cone displacement, occurs at the same time as a high tone which requires a small displacement.

If it is assumed that the spectrum contains only two frequencies f_1 and f_2, the modulation distortion will comprise new frequencies $f_2 \pm f_1$, $f_2 \pm 2f_1$, etc. The most important are $f_2 \pm f_1$, the first-order sidebands. Amplitude distortion results primarily in even-order sidebands. For a symmetrical cone compliance and magnet system the second-order side-bands $f_2 \pm 2f_1$ are probably the greatest magnitude.

It can be shown that the modulation distortion

$$d_m = 0{,}0013\, s_1 f_2, \qquad (2.27)$$

where s_1 is the cone motion in millimetres at the lower frequency f_1, and f_2 is the frequency being modulated.* The modulation distortion is thus expressed as a percentage of the amplitude of the signal f_2. By way of example, consider a 10-inch full-range speaker with an r.m.s. cone excursion

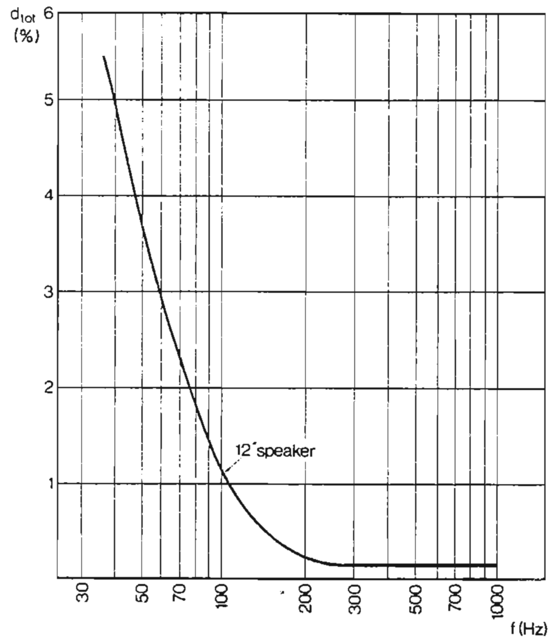


Fig. 2.17. Typical distortion as a function of frequency for a high-quality loudspeaker at 1 watt input.

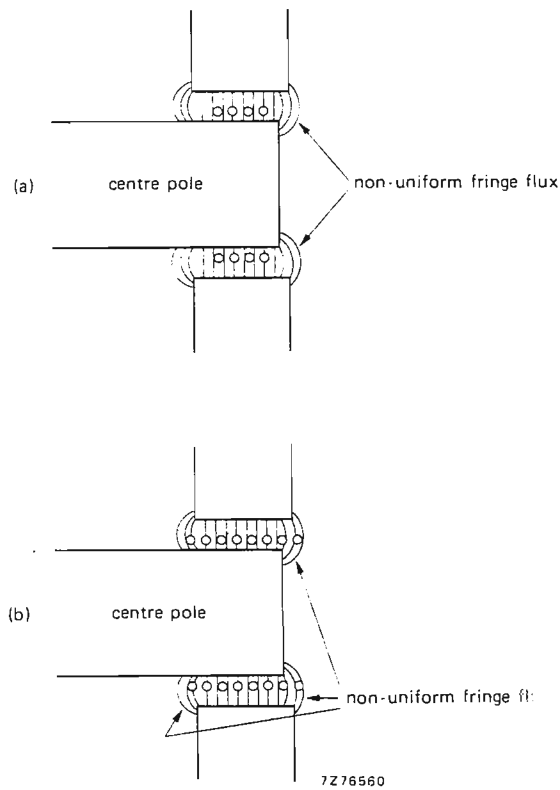


Fig. 2.18. Methods of reducing amplitude distortion due to non-uniform flux density in the air-gap. (*a*) Short voice coil, (*b*) long voice coil.

* Beers, G. L. and Belar, H. 1943. Frequency modulation distortion in loudspeakers. *Proc. I.R.E.* **31**: 132-138.

amplitude of 12,5 mm at some low frequency. At a frequency of 800 Hz,

$$d_m = 0{,}0013 \times 12{,}5 \times 800$$
$$= 13\%.$$

Amplitude distortion is also proportional to the amplitude of the cone motion, so it is important that cone motion is reduced as much as possible. The total distortion is the r.m.s. sum of the amplitude distortion and the modulation distortion,

$$d_{tot} = \sqrt{(d_a{}^2 + d_m{}^2)}. \qquad (2.28)$$

Another aspect of linearity is transient response, which is the ability of a loudspeaker to reproduce a short duration pulse without distortion of the waveshape and, particularly, without the addition of any frequencies. Good transient response requires a smooth frequency characteristic and phase response. These are normally difficult to obtain in a complex mechanical system. After removal of the driving pulse, the moving elements, excited by the coil but not necessarily rigidly coupled to it, continue to oscillate on their own (see Fig. 2.19).

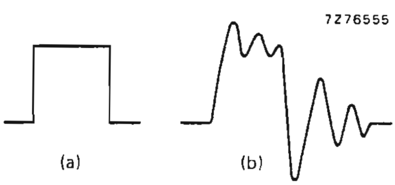


Fig. 2.19. Transient response of cone. (*a*) Input pulse, (*b*) cone motion.

2.12 Multi-way systems

Multi-way loudspeaker systems have the following advantages.

- Flatter sound pressure and power response curves, because each loudspeaker is designed to operate below the knee of its R_{MA} curve. Cone break-up can be avoided.
- High power-handling capacity, because the power spectrum is divided amongst several loudspeakers; thus the combination is able to handle more power than one loudspeaker alone.
- Better polar diagram, because each loudspeaker operates in a region where the wavelength is long compared with the loudspeaker dimensions; thus each loudspeaker may be regarded as a point source. Only at the crossover frequencies may this give rise to difficulty. At any point in the listening area where the distances to two loudspeakers operating at the same frequency differ from each other by half a wavelength, the sound pressure at that frequency will be diminished.

The plastic domed tweeter is very popular for high frequencies. This straddles the knee of the R_{MA} curve, and whilst it experiences no difficulties on the sloping portion, there may be problems above the knee where flexure is required; any damping is normally a result of internal friction in the material.

The design of the dividing network, or crossover network, should always be carried out experimentally. To use formulae indiscriminately which express the values of inductance and capacitance in terms of cross-over frequency and nominal impedance is very unsatisfactory. The amplifier impedance is almost zero and the impedance of loudspeakers is complex. Only a systems approach will give the best results; objective treatment alone stands a poor chance of success.

Ferrite cores may be used to reduce the physical size of the cross-over filter inductors. Two important aspects of their application should be considered: hysteresis, and saturation. Unless due regard is given to all the characteristics, hysteresis in the core of a cross-over filter inductor can result in complete loss of sound definition. Further, it should be remembered that at high power levels the current flowing in inductors is considerable and a ferrite core can easily be driven into saturation.

2.13 Loudspeaker magnets

Equation (2.22) showed that the Q of the loudspeaker varies inversely as the square of Bl, the product of flux density in the air-gap and length of wire in the voice coil. Efficiency, therefore, is closely related to the flux density in the air-gap, and for an efficient loudspeaker a strong magnet is a necessity.

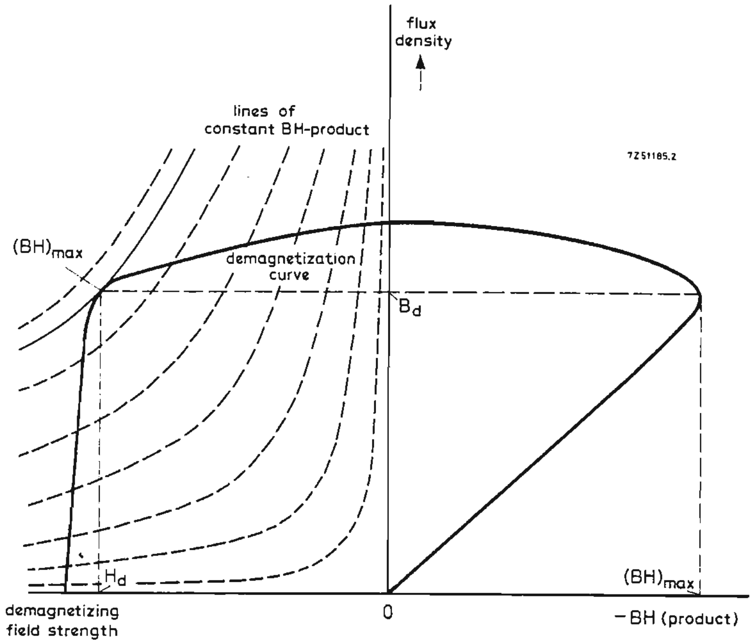


Fig. 2.20. Demagnetization curve with contours of constant *BH*-product and energy-product curve.

The classical *B-H* curve is of no interest here, since we are concerned with permanent magnets, and the demagnetization curve of Fig. 2.20 is of more importance. The *BH*-product indicates the energy in the material for a given value of *B*, and the maximum value of the *BH*-product on the demagnetization curve represents the ideal operating point for the magnetic material under static conditions. To achieve stable operation of a magnet system, which is a combination of magnet and air-gap, the operating point ought to be chosen a little higher up the demagnetization curve. This is shown in Fig. 2.21.

Under practical conditions the demagnetization of the material is not constant and the variation of flux density follows a line called the recoil line. This is shown in Fig. 2.22.

The operating point P_1 may drop below the knee of the demagnetizing curve, to P_2, because of an external demagnetizing field, an increase of the air-gap, or a temperature decrease that changes the demagnetization curve. Then, when the previous conditions are restored, the operating point will move from P_2 to P_3, following a recoil line parallel to part of the demagnetization curve. The values of *B* and *BH* will be lower at P_3 than at P_1, and to avoid this situation P_1 must be chosen a safe distance above the knee but without sacrificing too much *BH*-product, for the only way to restore the required *BH*-product will then be to use a larger magnet.

Many magnetic materials are used for loudspeaker magnet systems; amongst the most popular are anisotropic Ferroxdure, and anisotropic Ticonal.

Ferroxdure is made from a special form of powdered iron oxide which has been pressed and sintered. It can be ground to a higher order of accuracy. Ticonal may contain some or all of the following materials: titanium, cobalt, nickel, iron, aluminium, copper. Ticonal magnets are cast into shape and can also be ground. The

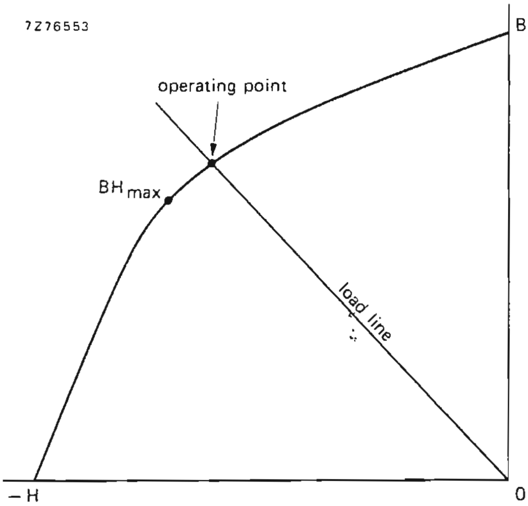


Fig. 2.21. An operating higher than the point of maximum *BH*-product gives better stability.

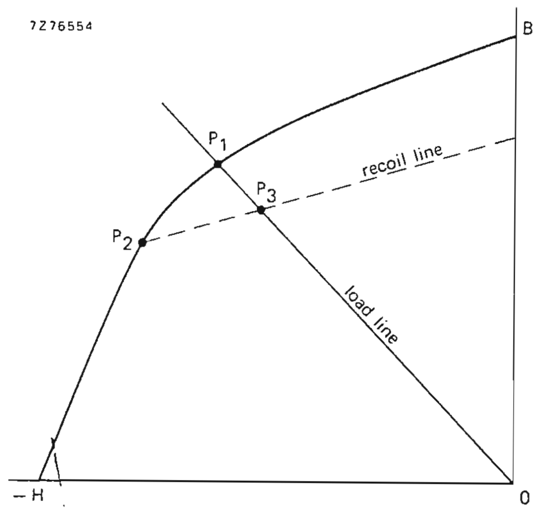


Fig. 2.22. If the operating point falls below the knee of the curve it returns to the load line via a recoil curve.

main differences between these materials are that whilst Ferroxdure magnets are characterized by high coercivity and resistivity, Ticonal magnets have higher values of remanent magnetism and energy product.

For a given air-gap, the length of the magnet is proportional to *H*, and the cross-sectional area is proportional to *B*. Two basic configurations thus occur:

Ferroxdure – large section, short length.
Ticonal – small section, greater length.
These are illustrated in Fig. 2.23.

In view of the trend towards shallower enclosures in domestic loudspeaker systems, loudspeakers with Ferroxdure magnets are now becoming increasingly popular. They also have the important advantage that they are less expensive. One advantage of a Ticonal magnet system is its lower external stray field due to the metal shielding of the pot; this is important where a loudspeaker is mounted in a colour television receiver close to the picture tube.

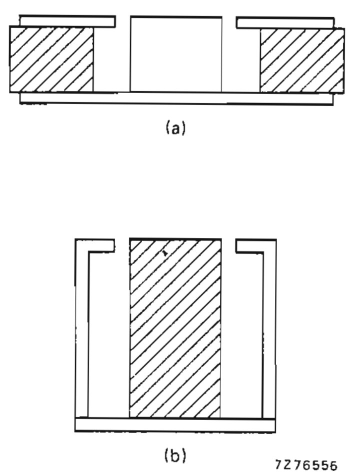


Fig. 2.23. Magnet system configuration in common use. (*a*) Ferroxdure magnet, (*b*) Ticonal magnet.

Designing hi-fi speaker systems - part 2

D. Hermans* and M. D. Hull**

In the first article in this series we considered the behaviour of the basic moving-coil direct-radiator loudspeaker operating unmounted. We shall now discuss the effect of a sealed enclosure on loudspeaker performance, the use of two or more loudspeakers in a multi-way system and the design of the cross-over networks that such a system demands.

3 The design of sealed enclosure systems

3.1 The infinite baffle

Consider a loudspeaker mounted on a small baffle board. When the cone moves forward air is compressed in front of it and rarefied behind it as shown in Fig. 3.1. The compressed air spills around the edge of the baffle; the air impedance is low (as for an unmounted loudspeaker) and thus the radiated sound pressure is low. The effect is most pronounced when the distance the sound wave has to travel from the front to the back of the loudspeaker is half a wavelength. At 40 Hz, for example, if the loudspeaker was mounted centrally the baffle would have to be at least 4,25 m square to prevent any appreciable cancellation. The larger the baffle the better the isolation between the front and the back of the loudspeaker. Complete isolation is obtainable with an infinitely large baffle – obviously an impracticable solution.

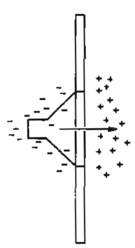


Fig. 3.1. A loudspeaker mounted on a small baffle board is subject to acoustic short-circuiting at low frequencies.

Complete isolation can be achieved, however, by folding the baffle around the back of the loudspeaker to form a closed box in which the rear radiation is completely suppressed. The closed box is known by various names

- infinite baffle (a misnomer)
- closed box baffle
- acoustic suspension
- sealed enclosure.

The term 'sealed enclosure' places stress on the most important feature of its construction, and it is the term used throughout this article. Although a sealed enclosure and an infinite baffle are often considered synonymous, there is one major difference between them. The air in the enclosure is constrained and therefore behaves as a spring as the cone moves in and out. This is not, of course, the case with an infinite baffle.

3.2 Equivalent circuits for sealed enclosure systems

Let us now consider the effect of mounting a loudspeaker in a sealed enclosure. If the enclosure is very large, the displacement of the air due

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** Philips Electronic Components and Materials Division, Eindhoven, The Netherlands.

to the movement of the loudspeaker cone will be insignificant compared with the volume of air in the enclosure. If, however, the enclosure is small, the movement of the cone will take place against a comparatively large change in pressure within the enclosure and there will be a marked difference in performance. The enclosure air acts as an added stiffness to the loudspeaker cone and this effect is more pronounced at small volumes. According to an equation in the first article,* the acoustic compliance of the enclosed volume V_B is $C_{AB} = V_B/\gamma P_0$.

Since at low frequencies the cone behaves as a rigid piston, we can now re-draw the equivalent circuit shown in Fig. 3.2(*a*) so that all elements are transferred to the acoustic side of the circuit. Mechanical elements discussed earlier** are divided by the square of the effective cone area to yield acoustic impedances in m.k.s. acoustic ohms. The electrical impedance due to the inductance of the voice coil (L_E) may be neglected below 500 Hz. The double radiation impedance $2Z_{MA}$ in Fig. 3.2(*a*) must now be replaced by two impedances: the radiation impedance from the front of the box (M_{AR}, R_{AR}) and the acoustic loading impedance due to the enclosure (C_{AB}, R_{AB}, M_{AB}). The new equivalent circuit is shown in Fig. 3.2(*b*).

* Section 1.3.4, page 6.

** Section 2.4, page 12.

The elements that make up the circuit of Fig. 3.2(*b*) should now be considered. Firstly e_g represents the open-circuit voltage of the amplifier, B is flux density in the air gap in teslas (1 tesla = 1 Wb/m^2 = 10^4 gauss), and l is the length of wire on the voice coil in metres. R_E is the d.c. resistance of the voice coil in ohms and R_g is the output impedance of the amplifier in ohms. (The occurrence of R_g in the denominator of one of the resistances reflects the influence of the amplifier output impedance on the damping of the system.) The effective area of the cone in square metres is represented by A.

The mechanical part of the loudspeaker has been reduced to three terms, as follows. Previously we have used M_{MC} to represent the

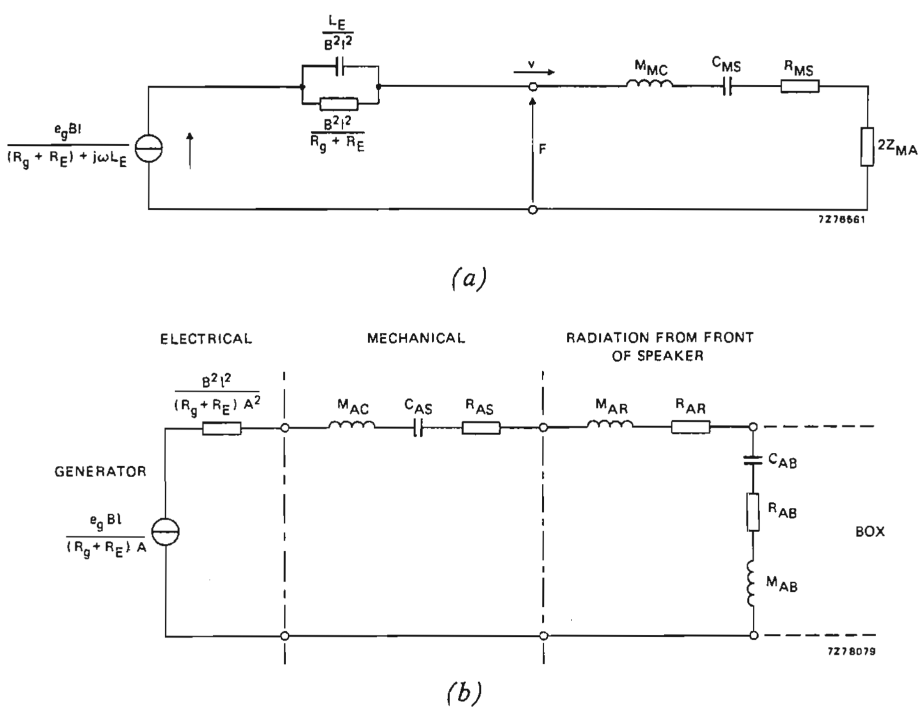
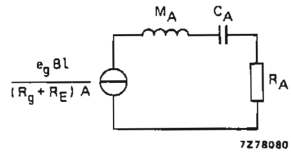


Fig. 3.2. (*a*) Equivalent circuit of a moving-coil loudspeaker. (*b*) Low-frequency equivalent circuit of the same loudspeaker mounted in a sealed enclosure – the cone operating as a piston. This circuit is valid up to about 500 Hz.



$$C_A = \frac{C_{AS}C_{AB}}{C_{AS} + C_{AB}} \qquad M_A = M_{AC} + M_{AR} + M_{AB}$$

$$R_A = \frac{B^2l^2}{(R_g + R_E)A^2} + R_{AS} + R_{AR} + R_{AB}$$

Fig. 3.3. Simplified version of Fig. 3.2(*b*)

mechanical mass of the coil and the cone. Dividing M_{MC} by A^2 we convert this quantity to an acoustic mass, M_{AC}, expressed in kg/m^4. Similarly, R_{MS} in Fig. 3.2(*a*) now becomes R_{AS} (the acoustic resistance of the suspension system in m.k.s. acoustic ohms) and C_{MS} becomes C_{AS} (the acoustic compliance of the suspension system in m^5/N). All these quantities can be determined in the laboratory as we shall see later.

Looking now at the remainder of the circuit elements of Fig. 3.2(*b*), M_{AR} represents the acoustic radiation mass of the air load in kg/m^4 and R_{AR} is the acoustic radiation resistance in m.k.s. acoustic ohms. The values of these quantities are related to the size of the baffle, not the enclosure volume, since they represent forward radiation. The quantity C_{AB}, on the other hand, represents the acoustic compliance in m^5/N and R_{AB}, the acoustic resistance of the enclosure. The latter is measured in m.k.s. acoustic ohms and depends on the absorption within the enclosure. M_{AB} is the acoustic mass of the air load on the back of the cone.

Figure 3.2(*b*) can be further simplified to the form shown in Fig. 3.3, which is a simple series circuit having a resonant frequency f_0' given by

$$f_0' = \frac{1}{2\pi\sqrt{M_A C_A}} \text{ Hz,} \tag{3.1}$$

where

$$M_A = M_{AC} + M_{AR} + M_{AB} \text{ kg/m}^4 \tag{3.2}$$

and

$$C_A = \frac{C_{AS}\ C_{AB}}{C_{AS} + C_{AB}} \text{ m}^5\text{/N.} \tag{3.3}$$

(The symbols f_0 and ω_0 were adopted in part 1 of this series of articles, for the resonant frequency of an unmounted loudspeaker. The symbols f_0' and ω_0' are used here to denote the resonant frequency of a loudspeaker mounted in an enclosure.)

3.3 Values of mass and compliance

3.3.1 DETERMINATION OF ACOUSTIC MASS

Equation 3.1 is fundamental to the design of a sealed enclosure system. First, let us consider the quantity M_A representing the total acoustic mass. This quantity is the sum of M_{AC}, the acoustic mass of the coil and cone, M_{AR}, the acoustic mass of the air load on the radiating side (front) of the cone, and M_{AB}, the acoustic mass of the air on the rear of the cone. These quantities are defined by:

$$M_{AC} = \frac{M_{MC}}{A^2} \text{ kg/m}^4, \tag{3.4}$$

where M_{MC} is the mechanical mass of the voice coil and cone in kg, and A is the effective area of the cone in m^2.

$$M_{AR} = \frac{M_{MR}}{A^2} \text{ kg/m}^4, \tag{3.5}$$

where M_{MR} is the mechanical mass of the air load on the front of the cone in kg.

$$M_{AB} = \frac{M_{MB}}{A^2} \text{ kg/m}^4, \tag{3.6}$$

where M_{MB} is the mechanical mass of the air load on the rear of the cone in kg.

Note that the acoustic quantities can be derived from their mechanical counterparts. The procedure is to measure the resonant frequency with the loudspeaker unmounted. A known mass of a few grams is then added to the cone and the new lower resonant frequency is measured. The total mass can then be calculated and, by subtracting the mass of the air load, the mass of the moving parts is easily determined. The method of calculation is given in Table 3.1.

Equations (3.7) to (3.10) show the procedure using acoustic quantities, whereas eqs (3.11) to (3.14) use the mechanical quantities with which most readers will be more familiar. The essential

Table 3.1 Determination of coil and cone mass

Acoustic quantities		Mechanical quantities	
1. $f_0 = \dfrac{1}{2\pi\sqrt{(M_{AC} + M'_{AR})C_A}}$,	(3.7)	$f_0 = \dfrac{1}{2\pi\sqrt{(M_{MC} + M'_{MR})C_M}}$,	(3.11)
where: M_{AC} = moving system mass in kg/m^4, M'_{AR} = total air load mass for both sides of cone in kg/m^4, C_A = compliance of suspension in m^5/N.		where: M_{MC} = moving system mass in kg, M'_{MR} = total air load for both sides of cone in kg, C_M = compliance of suspension in m/N.	
2. Add mass M_{AX} ($= \dfrac{\text{mass in kg}}{A^2}$ kg/m^4).		Add mass m kg.	
3. New resonant frequency $f_m = \dfrac{1}{2\pi\sqrt{(M_{AC} + M'_{AR} + M_{AX})C_A}}$.	(3.8)	$f_m = \dfrac{1}{2\pi\sqrt{(M_{MC} + M'_{MR} + m)C_M}}$.	(3.12)
4. Dividing eq. (3.7) by eq. (3.8) and manipulating $M_{AC} = \dfrac{M_{AX} f^2_m}{f^2_0 - f^2_m} - M'_{AR}$,	(3.9)	Dividing eq. (3.11) by (3.12) and manipulating $M_{MC} = \dfrac{m f^2_m}{f^2_0 - f^2_m} - M'_{MR}$,	(3.13)
where: $M'_{AR} \approx \dfrac{2 \times 0{,}16}{r}$ kg/m^4.	(3.10)	where: $M'_{MR} \approx 2 \times 1{,}58 r^3$ kg.	(3.14)

Note: M'_{AR} and M'_{MR} are the sum of the front and rear values given by eqs (3.15) and (3.16) in Table 3.2 and r is the cone radius in m.

Table 3.2 Determination of mass of air load

mounting	front		rear	
	acoustic M_{AR} (kg/m^4)	mechanical M_{MR} (kg)	acoustic M_{AB} (kg/m^4)	mechanical M_{MB} (kg)
unmounted, free-space or anechoic room	$\dfrac{0{,}16}{r}$ (3.15)	$1{,}58r^3$ (3.16)	$\dfrac{0{,}16}{r}$ (3.15)	$1{,}58r^3$ (3.16)
infinite baffle	$\dfrac{0{,}32}{r}$ (3.17)	$3{,}15r^3$ (3.18)	$\dfrac{0{,}32}{r}$ (3.17)	$3{,}15r^3$ (3.18)
small sealed enclosure	$\dfrac{0{,}23}{r}$ (3.19)	$2{,}27r^3$ (3.20)	$\dfrac{0{,}375k}{r}$ (3.21)	$3{,}75kr^3$ (3.22)

r = effective cone radius in m, k = mass loading factor from Fig. 3.5.
Numbers in parenthesis are equation numbers referred to in the text.

difference is that the acoustic masses are equal to the mechanical masses divided by A^2, and the acoustic compliance is equal to the mechanical compliance multiplied by A^2. This accounts for the factor of m^4 by which the respective units differ.

3.3.2 MASS OF THE AIR LOAD

In calculations on speaker performance the mass of the air is taken into account. Its inertia has to be overcome by the power supplied to the loudspeaker. When the enclosure is very large the effect on the forward radiation impedance is that of an infinite baffle. When the enclosure has a volume of less than about 220 litres (8 ft^3) the behaviour is that of a piston in the end of a long tube.

In this series of articles we are primarily concerned with small enclosures and our discussion will not represent the general case. For convenience the acoustic and mechanical masses represented by the air load on the cone are given in Table 3.2.

Due to the influence of the sealed enclosure on the sound radiation from the cone, the relative areas of the cone and the baffle board have to be taken into account. Table 3.2 summarizes the equations which should be used to find the air load under different conditions. The mechanical masses represented by the air loads given in Table 3.2 may be obtained directly from Fig. 3.4. The expressions given in Table 3.2 for the rear side air loading of a cone in a small sealed enclosure contain a constant, k (the mass loading factor). This is proportional to the ratio of effective cone area to baffle board area. The value of k as a function of relative area is plotted in Fig. 3.5.

3.3.3 COMPLIANCES OF THE SUSPENSION AND ENCLOSURE

Taking the compliance of the loudspeaker suspension system first, the acoustic compliance

$$C_{AS} = C_{MS}A^2 \text{ m}^5/\text{N}, \qquad (3.23)$$

where C_{MS} is the mechanical compliance of the suspension in m/N and A is the effective area of the cone.

The mechanical compliance can easily be determined by adding weights to the cone with the loudspeaker in a vertical position and measuring the cone displacement with a depth micrometer:

$$C_{MS} = \frac{\text{displacement (mm)}}{\text{added mass (g)} \times 9{,}8} \text{ m/N}. \qquad (3.24)$$

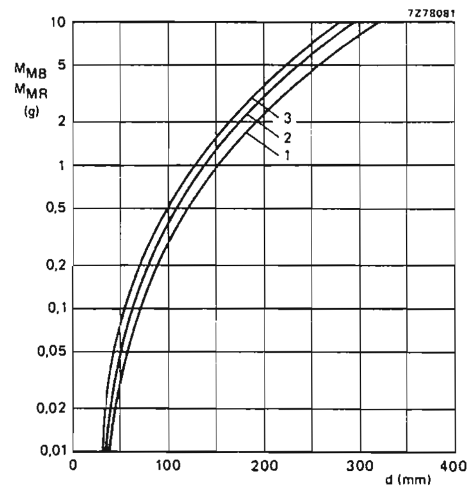


Fig. 3.4. Variation of air-load mass on one side of the cone with effective speaker diameter. Curve 1, sealed enclosure, front side only. Curve 2, infinite baffle, each side. Curve 3, sealed enclosure, rear side (multiply by k – Fig. 3.5).

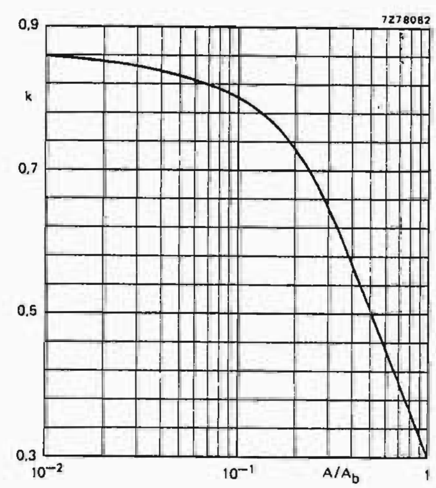


Fig. 3.5. Variation of mass loading factor k with the ratio A/A_b (effective cone area/baffle board area).

Alternatively, eq. (3.11) may be used if the masses M_{MC} and M'_{MR} are known:

$$C_{MS} = \frac{1}{4\pi^2 f_0^{\,2}(M_{MC} + M'_{MR})} \text{ m/N.} \qquad (3.25)$$

The acoustic compliance of the enclosure is given by

$$C_{AB} = \frac{V_B}{\gamma P_0} \text{ m}^5\text{/N,} \qquad (3.26)$$

where V_B is the volume of air in the enclosure in m^3, P_0 is normal atmospheric pressure (10^5 N/m^2) and γ (the ratio of the specific heat of the air at constant pressure to that at constant volume) is 1,4 for adiabatic compression as in a lined (not filled) enclosure.

We are dealing with acoustic quantities and therefore we can convert them to mechanical equivalents by dividing compliances or multiplying masses by A^2. Taking eq. (3.26) we then get

$$C_{MB} = \frac{C_{AB}}{A^2} = \frac{V_B}{\gamma P_0 A^2} \text{ m/N,} \qquad (3.27)$$

where C_{MB} represents the mechanical compliance of the enclosure.

From the law for adiabatic expansion and compression, the velocity of sound is given by the expression.

$$c = \sqrt{\frac{\gamma P_0}{\varrho}} \qquad (3.28)$$

thus $\gamma P_0 = \varrho c^2$, where ϱ is density.

Substituting this value for γP_0 in eq. (3.27) we get

$$C_{MB} = \frac{V_B}{\varrho c^2 A^2} \text{ m/N} \qquad (3.29)$$

hence the stiffness of the enclosure

$$S_B = \frac{\varrho c^2 A^2}{V_B} \text{ N/m.} \qquad (3.30)$$

This equation shows that the stiffness depends not only on the volume of the enclosure but also on the effective area of the cone.

3.4 Values of resistive elements

In Fig. 3.3 we combined the resistive components of Fig. 3.2 into a single element

$$R_A = \frac{B^2 l^2}{(R_g + R_E)A^2} + R_{AS} + R_{AB} + R_{AR} \qquad (3.31)$$

in m.k.s. acoustic ohms.

Let us now examine each of the terms in this expression. First of all it must be realized that these are acoustic quantities which is why the quantity A^2 appears in the denominator of the first term. Multiplying throughout by A^2 we obtain

$$R_M = \frac{B^2 l^2}{(R_g + R_E)} + R_{MS} + R_{MB} + R_{MR} \qquad (3.32)$$

which expresses eq. (3.31) in m.k.s. mechanical ohms. The values of the terms in eq. (3.31) can either be determined directly by calculation, or, using mechanically equivalent terms, by measurement.

The acoustic radiation resistance R_{AR} for small enclosures is given by

$$R_{AR} \approx \left(\frac{f}{10}\right)^2 \text{ m.k.s. acoustic ohms} \qquad (3.33)$$

where the frequency, f, is such that $2\pi r < \lambda$, i.e. $kr < 1$. (r is the cone radius.)

The acoustic resistance of the suspension is given by

$$R_{AS} = \frac{R_{MS}}{A^2} \text{ m.k.s. acoustic ohms} \qquad (3.34)$$

where R_{MS} is the mechanical resistance of the suspension. The method of determining R_{MS} will be dealt with later.

The acoustic resistance of the enclosure, R_{AB}, is dealt with in the next section.

3.5 Internal resonances of the enclosure

When the depth of the enclosure equals one half of the wavelength, the first fundamental mode of vibration occurs. The reactance of the enclosure is given by

$$X_{AB} = \omega M_{AB} - \frac{1}{\omega C_{AB}}. \qquad (3.35)$$

At resonance ωM_{AB} is greater than $1/\omega C_{AB}$; the reactance X_{AB} becomes positive and reaches a

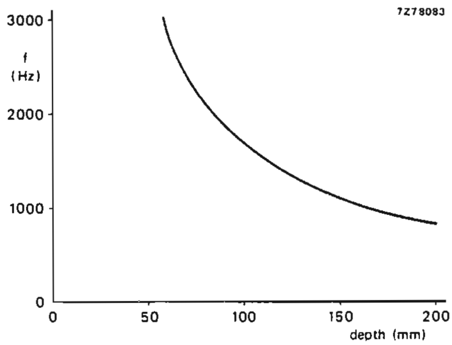


Fig. 3.6. Frequency of fundamental resonance of enclosure as a function of enclosure depth.

high value. This will greatly reduce the radiated power unless steps are taken to lower the value of X_{AB} at the resonant frequency. The frequency of fundamental resonance is plotted as a function of enclosure depth in Fig. 3.6.

To absorb energy at this and all higher frequencies, an acoustic lining is fitted inside the enclosure; preferred materials are, bonded mineral wool, acetate fibre, glass wool and bonded hair felt. Foamed plastic materials such as polyurethane foam, unless of the 'open' variety, are too dense in the sense that no direct air flow is possible through them.

At low frequencies, in small enclosures, a thickness of about 25 mm of absorptive material lining the sides, top, bottom and back of the enclosure can generally be used.

The impedance presented to the rear of the cone is

$$Z_{AB} = R_{AB} + jX_{AB} \tag{3.36}$$

where the components of X_{AB} are defined by eqs (3.21) and (3.26). The term R_{AB} represents the acoustic resistance of the enclosure and is given by

$$R_{AB} = \frac{R_{AM}}{\omega^2 C_{AB}{}^2 R_{AM}{}^2 + 1 + \dfrac{V_B}{\gamma V_M} + \dfrac{V_B{}^2}{\gamma^2 V_M{}^2}}. \tag{3.37}$$

$R_{AM} = R_f/3A_M$ = one-third of the total flow resistance of the material lining the box divided by the area of the material. For a 25 mm layer of a lightweight material of area A_M square metres, $R_{AM} \approx 33/A_M$ m.k.s. acoustic ohms. V_B is the total volume of the enclosure in m^3, and V_M is the volume of the acoustic lining material in m^3. Equation (3.37) applies only where the lining does not occupy more than 10% of the total volume.

3.6 Damping and Q-factor

We have shown that a loudspeaker mounted in a sealed enclosure may be treated as a series electrical circuit. Thus we may write

$$Q_T = \frac{\omega_0' M_A}{R_A}, \tag{3.38}$$

where Q_T is the total circuit magnification factor, $\omega_0' = 2\pi f_0'$ where f_0' is the resonant frequency, and M_A and R_A are defined by eqs (3.2) and (3.31) respectively.

In practice the value of Q_T may be readily determined by measuring the 3 dB points on the cone velocity curve as a function of frequency about resonance. From eq. (2.6)* we know that $e = Blv$ and since Bl is a constant for a particular loudspeaker, velocity v is proportional to e. Using the circuit of Fig. 3.7, the voltage across the voice coil

$$e = V_1 - iR_E = V_1 - \frac{V_2}{1000} R_E.$$

Since the values V_1 and V_2 may be read directly from the instruments, and R_E is the d.c. resistance of the voice coil, values of e may be plotted against frequency.

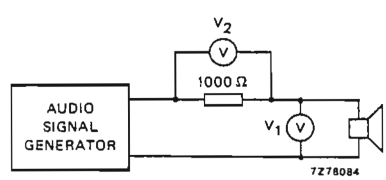


Fig. 3.7. Circuit for determining resonant frequency and Q.

* Section 2.4, page 12.

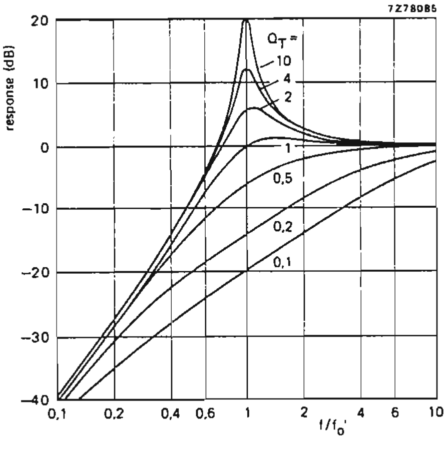


Fig. 3.8. Normalized frequency response of a typical low-frequency loudspeaker for different values of Q_T.

The value of Q_T for a constant current is then given by

$$Q_T = \frac{f_0'}{\Delta f}$$

where Δf is the 3 dB bandwidth. Figure 3.8 illustrates the response at resonance of a typical loudspeaker mounted in a sealed enclosure for different values of Q_T.

Let us now consider the effect of the amplifier output resistance R_g on the value of Q_T. In the method of determination of Q_T just discussed, the inclusion of the 1 kΩ series resistor produces a nearly constant current through the loudspeaker, i.e. $R_g \to \infty$. Thus the first term in eq. (3.31) becomes

$$\frac{B^2 l^2}{(R_g + R_E)A^2} \approx 0.$$

In practice, however, a modern solid-state amplifier has a very low output resistance, i.e. $R_g \to 0$, and the first term in eq. (3.31) becomes

$$\frac{B^2 l^2}{(R_g + R_E)A^2} \approx \frac{B^2 l^2}{R_E A^2}.$$

We can therefore distinguish two values of Q_T:

1) Q_{T1}, representing the constant current condition when $R_g \to \infty$

$$Q_{T1} \approx \frac{\omega_0' M_A}{(R_{AB} + R_{AS} + R_{AR})}, \qquad (3.39)$$

2) Q_{T2}, representing the constant voltage condition when $R_g \to 0$

$$Q_{T2} \approx \frac{\omega_0' M_A}{\dfrac{B^2 l^2}{R_E A^2} + R_{AB} + R_{AS} + R_{AR}}, \qquad (3.40)$$

the reciprocal of which can be expressed in the form

$$\frac{1}{Q_{T2}} = \frac{B^2 l^2}{\omega_0' M_A R_E A^2} + \frac{R_{AB} + R_{AS} + R_{AR}}{\omega_0' M_A}$$

$$= \frac{B^2 l^2}{\omega_0' M_A R_E A^2} + \frac{1}{Q_{T1}}. \qquad (3.41)$$

Now M_A is the acoustic mass in kg/m^4, and, multiplying by A^2 we can express this in terms of M_M, the total mechanical mass in kg, where

$$M_M = M_{MC} + M_{MR} + M_{MB}, \qquad (3.42)$$

in which M_{MC} is the mechanical mass of the coil and the cone, M_{MR} is given by eq. (3.20), and M_{MB} is given by eq. (3.22).

We can therefore rewrite eq. (3.41) substituting M_M for $M_A A^2$,

$$\frac{1}{Q_{T2}} = \frac{B^2 l^2}{\omega_0' M_M R_E} + \frac{1}{Q_{T1}}, \qquad (3.43)$$

where B is the magnetic flux density in teslas and l is the length of wire on the voice coil in metres.

The first term on the right-hand side of eq. (3.43) represents the damping due to the voice coil resistance and the magnet system. Denoting this by $1/Q_E$, we can write

$$\frac{1}{Q_{T2}} = \frac{1}{Q_E} + \frac{1}{Q_{T1}} \qquad \text{(cf eq. (2.24) *)}.$$

The importance of these equations must be stressed. Published frequency response curves are invariably obtained using constant voltage sources and thus compare with the results obtainable when the loudspeaker is fed from an amplifier with a low output resistance. Such curves do not, therefore, illustrate the effect on Q_T when measurements are made using a constant current source (Fig. 3.7) rather than the usual constant

* Section 2.7, page 17.

voltage source and comparison of the results obtained becomes very difficult indeed.

From eq. (3.40) it is apparent that, with a given loudspeaker, there are only two possibilities for altering Q_T. Either alter the baffle board in order to modify the value of M_A by altering the value of k, (eq. (3.21)). Or alter one of the resistive components R_{AB}, R_{AS} or R_{AR}.

The first of these, R_{AB}, is the acoustic resistance of the enclosure – see eq. (3.37). R_{AS}, the acoustic resistance of the suspension, is fixed in the manufacture of the speaker. R_{AR}, the acoustic radiation resistance, is a function of frequency – see eq. (3.33). Therefore, all we can do to control Q_T disregarding the amplifier output resistance, is to change R_{AB} by modifying the value of R_{AM} – eq. (3.37).

We considered eq. (3.37) in terms of a 25 mm lining of lightweight absorbent material, the generalized equation being difficult to handle. It is obvious that we cannot increase the area of the lining once every part of the inside walls has been covered. The next step is to stuff the enclosure with absorbent material, and this is where calculation gives way to a more empirical approach.

A new situation exists; repeating eq. (3.26):

$$C_{AB} = \frac{V_B}{\gamma P_0}.$$

The acoustic compliance of the enclosure, C_{AB}, is inversely proportional to γ. For adiabatic processes γ has a value of 1,4 at normal temperature and pressure. But what happens when the enclosure is full of damping material?

If the air space is completely filled with a soft, lightweight material, compression and expansion become isothermal. The speaker drives energy into the enclosure and the material converts it into heat. This shows up a overdamping at the low end of the frequency response characteristic.

Under isothermal conditions the speed of sound falls from 344 m/s to 292 m/s and all the equations in which c has been assumed to be constant are modified. The fact that the value of γ in eqs (3.26) and (3.27) is no longer 1,4 but 1,0 means that filling the enclosure has the same effect on the compliance as a 40% increase in volume.

Obviously, the practical method of designing a sealed enclosure loudspeaker system is to follow objective methods up to a certain point and to perform the final touches by filling, or partly filling the enclosure to obtain the desired characteristics.

One last word on the subject of materials. We have previously rejected denser materials which will not permit a direct air flow through them. These materials cannot be entirely ruled out as far as lining the enclosure is concerned, but it is a matter of art and experience rather than calculation, that will give the required Q_T for the system concerned. The value of γ will be somewhere between 1,0 and 1,4. The reader who investigates this area of activity will doubtless evolve some empirical design rules of his own but he must be careful that his chosen material does not radically affect the volume of the enclosure.

One difficulty facing a designer is the decision about what value of Q_T to aim for. There is no analytical basis for choice and arguments which favour many values between 0,5 and 1,4 can be found in the literature. There is nothing more offensive than a loudspeaker system with too high a value of Q_T which 'rings', and there is also nothing more unsatisfactory than a 'dead' one. In the latter case, probably underdamping is tolerable. In the former, the high value may give the promise of good bass only to prove that at high loudness levels it simply cannot handle transients.

For satisfactory transient response, it is suggested that

$$Q_T < \frac{f_0'}{30}. \tag{3.44}$$

How much less than $f_0'/30$ is a matter for the designer to decide but anything greater than this in a high-fidelity application may be unsatisfactory. Some speaker systems sound livelier than others, some more solid. Every manufacturer sets his own standard over this question. Like pianos, loudspeaker systems have their own tonal qualities.

In the end a pair of ears and an open mind are used to assess the final result. When the bass begins to sound constricted or overdamped, remove some stuffing from the enclosure. When the bass sounds relaxed and ample, but not sloppy, it is correct.

3.7 Effect of enclosure compliance on resonant frequency

We may write the expression for the resonant frequency of an unmounted loudspeaker in an anechoic room as

$$f_0 = \frac{1}{2\pi C_{AS}\sqrt{(M_{AC} + 2M_{AR})}}, \tag{3.45}$$

where M_{AC} is the acoustic mass of the moving system as before. M_{AR} is the acoustic mass of the air load on each side of the cone and C_{AS} is the acoustic compliance of the suspension as defined in eq. (3.23).

When we take the same loudspeaker and mount it in a sealed enclosure, eq. (3.45) is modified by the acoustic mass of the air load due to the enclosure, and the effect of the enclosure acoustic compliance. Referring back to Fig. 3.2, we can rewrite eq. (3.45) in the form

$$f_0' = \frac{1}{2\pi}\sqrt{\frac{C_{AS} + C_{AB}}{C_{AS}C_{AB}(M_{AC} + M'_{AR} + M_{AB})}} \tag{3.46}$$

where M'_{AR} is the new value of air load on the front of the cone given by eq. (3.19), M_{AB} is the acoustic mass of the air load on the rear of the cone, and C_{AB} is the acoustic compliance of the enclosure given by eq. (3.26).

Taking the ratio of eqs (3.45) and (3.46) we may now determine the change in resonant frequency due to the effect of the enclosure.

$$\frac{f_0'}{f_0} = \sqrt{\left\{\left(1 + \frac{C_{AS}}{C_{AB}}\right)\frac{M_{AC} + 2M_{AR}}{M_{AC} + M'_{AR} + M_{AB}}\right\}}. \tag{3.47}$$

Since $M'_{AR} \approx 1{,}4M_{AR}$, we can write eq. (3.47) in the form

$$\frac{f_0'}{f_0} = \sqrt{\left\{\left(1 + \frac{C_{AS}}{C_{AB}}\right) \times \frac{M_{AC} + M'_{AR} + 0{,}6M_{AR} + M_{AB} - M_{AB}}{M_{AC} + M'_{AR} + M_{AB}}\right\}}$$

$$= \sqrt{\left\{\left(1 + \frac{C_{AS}}{C_{AB}}\right) \times \left(1 + \frac{0{,}6M_{AR} - M_{AB}}{M_{AC} + M'_{AR} + M_{AB}}\right)\right\}}. \tag{3.48}$$

Let us now examine the values of the acoustic masses. From eq. (3.15)

$$2M_{AR} = \frac{2\times 0{,}16}{r} = \frac{0{,}32}{r}$$

from eq. (3.19),

$$M'_{AR} = \frac{0{,}23}{r}$$

and from eq. (3.21),

$$M_{AB} = \frac{0{,}375k}{r}.$$

The value of k in eq. (3.21) depends on the proportion of the baffle area that the loudspeaker occupies. If the area of the speaker is about one-third of the baffle board area, k has a value of about 0,65 (see Fig. 3.5). If we take this value as an example

$$M_{AB} \approx \frac{0{,}24}{r}.$$

Thus, if the loudspeaker occupies a third or less of the baffle board area, we can write the following approximation for eq. (3.48)

$$\frac{f_0'}{f_0} \approx \sqrt{\left\{0{,}87\left(1 + \frac{C_{AS}}{C_{AB}}\right)\right\}}, \tag{3.49}$$

which represents the ratio of the resonant frequency with a sealed enclosure to the resonant frequency in free space without a baffle.

It makes no difference whether acoustic or mechanical units are used to express the compliance, as it is the ratio of these quantities that is of interest. If for greater simplicity we refer to stiffnesses, we can rewrite eq. (3.49) in the form

$$\frac{f_0'}{f_0} \approx \sqrt{\left\{0{,}87\left(1 + \frac{S_b}{S_s}\right)\right\}}, \tag{3.50}$$

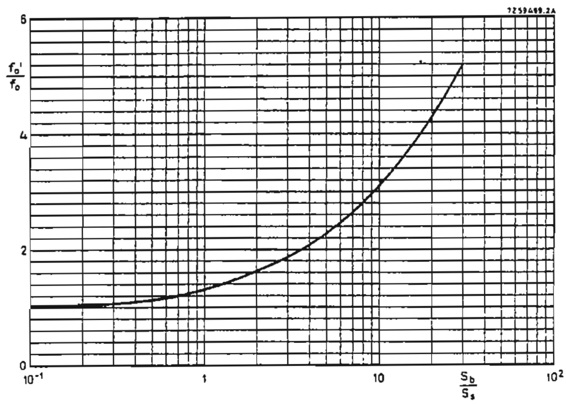


Fig. 3.9. Variation of resonant frequency with enclosure to speaker stiffness ratio.

where S_b is the reciprocal of the enclosure compliance given by eq. (3.29),

$$S_b = \frac{\varrho c^2 A^2}{V_B} \text{ N/m} \tag{3.51}$$

and S_s is the stiffness of the speaker suspension. Figure 3.9 shows the variation of f_0'/f_0 as a function of the ratio S_b/S_s.

3.8 Power considerations

3.8.1 RADIATED SOUND PRESSURE

The sound pressure level (SPL) is defined by

$$\text{SPL} = 20\log_{10} \frac{P}{P_{ref}} \text{ dB,} \tag{3.52}$$

where P is the measured pressure and P_{ref}, the reference sound pressure, is 2×10^{-4} μbar (2×10^{-5} N/m).

The acoustic power level (PWL) of a sound source, which is responsible for the production of the required sound pressure level, is given by

$$\text{PWL} = 10\log_{10} \frac{W}{W_{ref}} \text{ dB,} \tag{3.53}$$

where W is the acoustic power in watts, and W_{ref} is 10^{-12} watts. Thus, a source radiating one acoustic watt has an acoustic power level of 120 dB.

At low frequencies the loudspeaker acts as a spherical source of sound and we may now consider the levels which exist in the case of a speaker system in a particular living room. In order that we can restrict our treatment of the subject to the most likely circumstances which will exist, we shall consider stereo reproduction in a small room. Let us assume a 'wall' of sound is created by loudspeakers in corner position.

It can be shown that the sound pressure level in the near field close to the wall is given by

$$\text{SPL} = \text{PWL} + 10\log_{10}\left(\frac{1}{A_W} + \frac{4}{R}\right) \tag{3.54}$$

where A_W is the area of the radiating wall in m²; and $R = \alpha A/(1-\alpha)$, where A is the total surface area of the room in m², and α is the average absorption coefficient, about 0,15 for an average room.

Let us take a room 5 metres long, 4 metres wide and 3 metres high. In the first case we will place the speakers at the end of the room and calculate the acoustic power required to produce a sound pressure level of 90 dB close to the wall.

$$A = 94 \text{ m}^2$$
$$R = (0{,}15\times94)/0{,}85 = 16{,}6 \text{ m}^2$$
$$A_W = 12 \text{ m}^2$$

therefore

$$\frac{1}{A_W} + \frac{4}{R} = \frac{1}{12} + \frac{4}{16{,}6} = 0{,}324$$

and

$$10\log_{10}\left(\frac{1}{A_W} + \frac{4}{R}\right) = -4{,}9 \text{ dB.}$$

From eq. (3.53)

$$\text{PWL} = 10\log_{10} \text{W} + 120 \text{ dB,} \tag{3.55}$$

and substituting this for PWL in eq. (3.54) we obtain

$$\text{SPL} = 10\log_{10} \text{W} + 120 - 4{,}9 \text{ dB}$$

thus

$$10\log_{10} \text{W} = \text{SPL} - 120 + 4{,}9 \text{ dB}$$
$$= 90 - 120 + 4{,}9$$
$$= -25{,}1$$
$$\log_{10} \text{W} = -2{,}51$$
$$\text{W} = 3{,}09 \text{ mW.}$$

Thus approximately 3 mW of acoustic power must be radiated by the source to produce a sound pressure level of 90 dB close to the wall.

If we move the loudspeakers to the longer wall of the room, the acoustic power level required rises to 3,16 mW since a greater area of radiating wall has to be covered.

Now the sealed enclosure may be treated as if it were a spherical source of sound so long as the circumference of the cone is less than a wavelength. The sound pressure P at a distance r from the enclosure in a free field is then given by

$$P = \frac{f\varrho U_c}{2r}\ \mathrm{N/m^2}, \tag{3.56}$$

where f is the frequency in Hz and U_c is the r.m.s. volume velocity of the cone in m^3/s. At low frequencies, above resonance, where the loudspeaker is mass-controlled, we can define the reference volume velocity $U_{c\ ref}$ by the expression

$$U_{c\ ref} = \frac{e_g Bl}{(R_g + R_E)2\pi A f M_A}. \tag{3.57}$$

Whence the reference sound pressure, P_{ref}, is

$$P_{ref} = \frac{e_g Bl}{(R_g + R_E)4\pi r A M_A}, \tag{3.58}$$

where r is the distance from the loudspeaker in m, A is the effective area of the cone in m^2, and M_A is as given by eq. (3.2).

We can determine the actual volume velocity U_c from the equivalent circuit of Fig. 3.2,

$$U_c = \frac{e_g Bl}{A(R_g + R_E)\sqrt{\{R_A{}^2 + (\omega M_A - 1/\omega C_A)^2\}}}$$

where

$$(\omega M_A - 1/\omega C_A)^2 = \omega_0'^2 M_A{}^2 \left(\frac{\omega}{\omega_0'} - \frac{\omega_0'}{\omega}\right)^2$$

and, from eq. (3.1)

$$\omega_0'^2 = \frac{1}{M_A C_A}.$$

Furthermore, from eq. (3.38)

$$Q_T = \frac{\omega_0' M_A}{R_A},$$

so we can write

$$U_c = \frac{e_g Bl}{(R_g + R_E)\omega_0' A M_A \sqrt{\left\{\frac{1}{Q_T{}^2} + \left(\frac{\omega}{\omega_0'} - \frac{\omega_0'}{\omega}\right)^2\right\}}}. \tag{3.59}$$

By substituting eq. (3.59) in eq. (3.56) and dividing by eq. (3.58) we can derive the following expression for the ratio of the radiated sound pressure to the reference sound pressure

$$20\log_{10}\frac{P}{P_{ref}} = 20\log_{10}\frac{\omega}{\omega_0'} - 10\log_{10}\left\{\frac{1}{Q_T{}^2} + \left(\frac{\omega}{\omega_0'} - \frac{\omega_0'}{\omega}\right)^2\right\} \tag{3.60}$$

where ω is the frequency at which the response is to be found.

When measurements are made using a microphone at a given distance in front of the loudspeaker, the frequency response characteristic follows eq. (3.60). The reference level must always be stated if actual values of pressure are required.

An important feature illustrated by eq. (3.60) is that the sound pressure level at resonance differs from the reference sound pressure level above resonance by

$$20\log_{10}\frac{P}{P_{ref}} = -10\log_{10}\frac{1}{Q_T{}^2} = 20\log_{10} Q_T. \tag{3.61}$$

3.8.2 POWER HANDLING CAPACITY

The power handling capacity of a loudspeaker is the maximum continuous power the loudspeaker is designed to handle. This must not be confused with the music power rating, described in Sect. 3.8.6, which is usually measured in terms of pulsatory loading representing music and speech at the low frequency end of the audio spectrum.

Power handling capacity is tested by applying a weighted noise voltage to the loudspeaker terminals, and measuring the r.m.s. voltage V across the voice coil. The peak noise voltage is clipped at a level equal to twice the r.m.s. voltage.

The loudspeaker must be able to withstand this signal for 100 hours continuously and meet all its specifications afterwards (except resonant frequency which may decrease). Power handling capacity is defined

$$\text{PHC} = \frac{V^2}{R},$$

where R is the loudspeaker impedance as stated by the manufacturer.

The same principle applies to multi-way systems but here a speaker power handling capacity, PHC_1, may be defined for the individual speakers as well as a system power handling capacity, PHC_2, for the total system.

$$\text{PHC}_2 = \frac{V_1^2}{Z} \quad \text{and} \quad \text{PHC}_1 = \frac{V_2^2}{R},$$

where V_1 is the test voltage applied to the input terminals of the cross-over filter, and V_2 is the voltage across the voice coil of one of the loudspeakers. (See Fig. 3.10.)

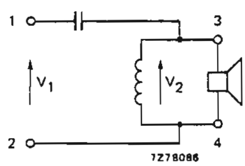


Fig. 3.10. Measurement of speaker and system power.

It is possible to build a loudspeaker system in which all the loudspeakers are capable of handling the maximum power specified for the total system. This would be expensive and thus the high-frequency speakers are commonly of a lower power handling capacity than the low frequency loudspeakers. Remembering that the highest fundamental frequency produced by conventional musical instruments is a little over 4000 Hz it is evident that a tweeter, which will not come into operation until about 3000 Hz, will handle mainly harmonics. As the harmonics of a note are very much diminished in intensity compared to the fundamental we can safely use medium and high-frequency loudspeakers of reduced power handling capacity.

To establish how much power each speaker in a system will be required to handle many things have to be considered. There are many variables, the most obvious being the type of sound that the speaker may be called upon to handle. Thus safety margins must be observed. A great deal of research has been carried out on the energy distribution through the frequency range of various types of music. The result is the establishment of what have now become clearly defined standards.

In Europe the IEC DIN standards clearly define the noise spectrum which is to be used for testing loudspeakers and loudspeaker systems. This is shown in Fig. 3.11. As is obvious to the most casual listener, the energy content of modern pop music is distributed rather differently across the audio spectrum than that of classical orchestral works. The bold line in Fig. 3.11 is a new standard recommended in order to cater for this difference.

Figure 3.12 is also the result of studying many recordings. The curve can be used to determine the approximate apportionment of total power (PHC_2) between the speakers of a multi-way system. If, for instance, a 2-way system with a cross-over frequency of 1200 Hz is specified, the point at which the curve intersects 1200 Hz corresponds to 75% on the ordinate. This indicates that the division of energy between the woofer and the tweeter is about 75:25. In a 20 W system the woofer should thus be able to handle at least 15 W and the tweeter at least 5 W. A safety margin must be allowed on top of these figures as the ratio depends on the frequency content of the input signal and this can be modified to quite a

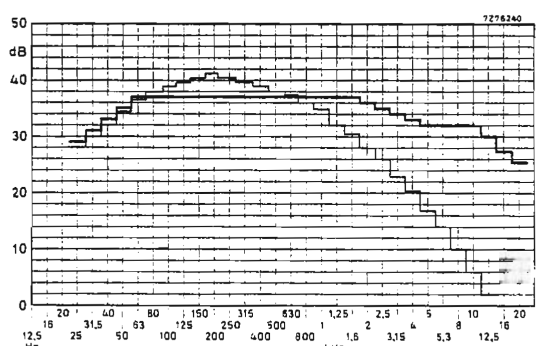


Fig. 3.11. DIN 45500 noise curve (thin line) as used for measuring and specifying power handling capacity. The alternative IEC curve (bold line) is currently under consideration.

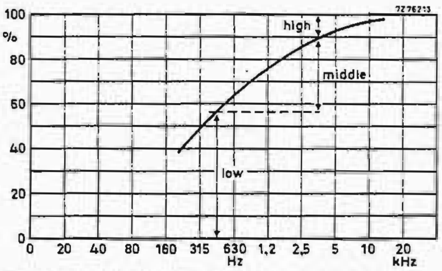


Fig. 3.12. Typical power distribution in a multi-way system. The curve is based on measurements made on many recordings of music. It is thus typical and allowance should be made for variations in the frequency content of the sound to be reproduced by the system, the characteristics of the amplifier driving the system and the setting of the tone controls on the amplifier.

considerable degree by varying the setting of the amplifier tone controls. Figure 3.12 can thus only be taken as a guide but, nonetheless, quite a useful one. Used to determine the typical power distribution in a 3-way, 40 W system with cross-over frequencies of 630 Hz and 2500 Hz, it gives us the minimum values of 25,6 W, 8,8 W and 5,6 W for the woofer, squawker and tweeter respectively.

Finally, a warning is perhaps in order. Mistakes are sometimes made in testing the loudspeakers by not allowing for the reduced power handling capacity of the higher range speakers. Sweeping a tone of fixed amplitude up through the frequency range of the system is a common method of measuring the frequency response of the system. Unless the very much reduced power handling capacity of the higher range speakers is taken into account, damage can be caused.

3.8.3 EFFICIENCY

The acoustic power in watts radiated from both sides of the loudspeaker cone is:

$$W = |v|^2 2R_{MR} \tag{3.62}$$

where v is the voice coil velocity in m/s, and R_{MR} is the mechanical radiation resistance in m.k.s. mechanical ohms.

Assuming that the inductive reactance of the voice coil is negligible compared to $(R_g + R_E)$, the voice coil velocity is given by

$$v = \frac{e_g Bl}{(R_g + R_E)(R_M + jX_M)} \tag{3.63}$$

in which

$$R_M = \frac{B^2 l^2}{(R_g + R_E)} + R_{MS} + 2R_{MR}$$

$$X_M = \omega M_{MD} + 2X_{MR} - \frac{1}{\omega C_{MS}},$$

where M_{MD} is the mass of the diaphragm, X_{MR} is mechanical radiation reactance, and C_{MS} is the mechanical compliance of the suspension. Squaring the modulus of v and substituting in eq. (3.62) then gives

$$W = \frac{2e_g{}^2 B^2 l^2 R_{MR}}{(R_g + R_E)^2 (R_M{}^2 + X_M{}^2)}. \tag{3.64}$$

The maximum power available from the amplifier is obtained when the internal resistance of the amplifier is equal to the d.c. resistance of the speaker; that is,

$$W_E = \frac{e_g{}^2}{4R_g}. \tag{3.65}$$

Dividing eq. (3.64) by eq. (3.65) and multiplying by 100 we obtain the efficiency of the loudspeaker related to the maximum power output of the amplifier:

$$\eta = \frac{W}{W_E} \times 100 = \frac{800 B^2 l^2 R_g R_{MR}}{(R_g + R_E)^2 (R_M{}^2 + X_M{}^2)}\ \%. \tag{3.66}$$

This is known as the power available efficiency.

To enable a frequency response curve to be plotted without showing the acoustic power being radiated, it is useful to define a reference efficiency. The reference power available efficiency for both sides of the cone is

$$\eta_{ref} = \frac{800 B^2 l^2 R_g R_{MR}}{\omega^2 (R_g + R_E)^2 (M_{MC} + 2M_{MR})^2}. \tag{3.67}$$

At low frequencies,

$$R_{MR} = \frac{\omega^2 A^2 \varrho}{2\pi c}$$

so eq. (3.67) becomes

$$\eta_{ref} = \frac{800 R_g B^2 l^2 A^2 \varrho}{2\pi c (R_g + R_E)^2 (M_{MC} + 2M_{MR})^2}. \tag{3.68}$$

This gives the actual response in the flat region of the frequency characteristic above resonance.

The ratio of the response at medium and low frequencies given by eq. (3.66), where the radiation is not directional, to the reference power available efficiency is

$$\frac{\eta}{\eta_{ref}} = \frac{\omega^2(M_{MC} + 2M_{MR})^2}{R_M{}^2 + X_M{}^2}. \tag{3.69}$$

An interesting situation occurs at resonance when $X_M = 0$, and eq. (3.69) becomes

$$\frac{\eta}{\eta_{ref}} = \frac{\omega_0{}^2(M_{MC} + 2M_{MR})^2}{R_M{}^2} = Q_T{}^2. \tag{3.70}$$

Thus, if we know the reference power available efficiency, we can easily determine the efficiency at resonance, since from eq. (3.70)

$$\eta = \eta_{ref} Q_T{}^2. \tag{3.71}$$

The values of the efficiencies given by eqs (3.66), (3.67) and (3.68) are for both sides of the loudspeaker and should be halved for forward radiation only.

When a loudspeaker is mounted in a sealed enclosure, the effect of the rear-side impedance modifies the above equations. The radiation impedance is also affected by the size of the baffle board. The enclosure compliance, which depends upon the volume of the enclosure, also affects the rear-side impedance. If we were to confine ourselves to a purely mathematical treatment of power handling capacity and efficiency, the resulting equations would become very difficult to handle. We shall resort to methods of approximation which produce sufficiently accurate results for all practical purposes.

A loudspeaker driven by an amplifier having $R_g = 0$ is a special case for which the efficiency may be defined as

$$\eta = \frac{W_A}{W_{in}},$$

where $W_A = v^2 R_{MR}$, the radiated acoustic power, and $W_{in} = i^2 R_E$, the applied electrical power dissipated in the voice coil resistance.

Above the resonant frequency the radiated acoustic power can be expressed as

$$W_A = v^2 R_{MR} = \left(\frac{Bli}{\omega M_M}\right)^2 R_{MR}.$$

For a large box R_{MR} can be taken to be the same as for an infinite baffle,

$$R_{MR} = 1{,}57\omega^2 r^4 \varrho / c.$$

Whence

$$\eta = \left(\frac{Bli}{\omega M_M}\right)^2 \frac{1{,}57\omega^2 r^4 \varrho}{c} \times \frac{1}{i^2 R_E}$$

$$= \frac{1{,}57 B^2 l^2 r^4 \varrho}{M_M{}^2 R_E c}, \tag{3.72}$$

where $M_M = M_{MC} + M_{MR} + M_{MB}$, and the resistance of the voice coil $R_E = \sigma l/s$ where σ = specific resistivity of the wire, l = length of wire, and s is the cross-sectional area of the wire.

Furthermore, the mass of the voice coil, M_C, is given by

$$M_C = ls\beta$$

where β is the density of the wire. By substitution therefore

$$\frac{l^2}{R_E c} = \frac{M_C}{\sigma\beta}$$

whence

$$\eta = \frac{1{,}57 B^2 M_C r^4 \varrho}{\beta M_M{}^2 c}. \tag{3.73}$$

3.8.4 BASIS OF POWER RATINGS

In Europe, it is convenient to adopt the German Standards (DIN) for determining speaker performance, since no satisfactory alternative exists which deals with the problem in sufficient depth. We do not intend to deal here with anything more than the basic principles involved in specifying loudspeaker systems in terms of hi-fi. Readers wanting a detailed analysis of the requirements should consult DIN 45500 and DIN 45573.

Besides the nominal power handling capacity which we have already dealt with, there are two distinct power levels to be considered:

- operating power;
- music power.

Each serves a very different purpose and there is no direct relationship between them.

3.8.5 OPERATING POWER

Operating power is the power input to the system to produce a sound pressure of 12 μb at 1 m distance (or 4 μb at 3 m). Taking 2×10^{-4} μb as the reference sound pressure level, 12 μb ≈ SPL 96 dB (4 μb ≈ SPL 86 dB).

This simplified definition establishes a reference for further acoustic calculations. To determine the operating power, the loudspeaker should be measured in half free field conditions (infinite baffle) and the average sound pressure level (on axis) between 100 Hz and 4000 Hz (or another indicated appropriate frequency range) should be 96 dB at a distance of 1 m.

The operating power is, naturally, in electrical watts and is simply determined by increasing the electrical input until the required sound pressure at the appropriate distance is reached.

3.8.6. MUSIC POWER

Music power, or maximum power handling capacity is the peak power which can be applied to the loudspeaker or system for short periods (2 seconds maximum) without any audible distortion being suffered. The loading is applied as a sinusoidal signal between 250 Hz and the lower frequency limit. In general the music power rating is much higher than the continuous power handling capacity.

3.9 Enclosure proportions and construction

An advantage of sealed enclosure systems is that there is no optimum volume or shape. Baffle board proportions of 5:3 and 4:3 are common but there is no reason why the proportions should not be increased to 2:1 or more, provided that a satisfactory speaker layout can be achieved.

The first normal mode of internal vibration occurs when the half wavelength becomes equal to the enclosure depth. A minimum enclosure depth can be calculated from the woofer cross-over frequency, e.g., if the cross-over frequency is 500 Hz, $\lambda/2 = 344$ mm and a depth less than this should be chosen. However, with an absorptive lining in the enclosure, if the depth is more than $\lambda/4$ the rear side reactance X_{AB} is positive and the loading at the back of the cone approaches that of an infinite baffle. Hence the internal depth of the enclosure should lie between a half and a quarter-wavelength of the cross-over frequency.

Offsetting the loudspeakers from the centre of the baffle may be advantageous. Mounting them flush with the baffle board, especially the squawker and the tweeter, and using sufficiently thick material for the walls of the enclosure to avoid panel resonances will overcome any disadvantages arising from mounting the woofer centrally.

3.10 Determination of enclosure volume

3.10.1 EFFECT OF ENCLOSURE VOLUME ON RESONANT FREQUENCY

In Section 3.7 we discussed the effect of enclosure compliance on resonant frequency. For a given percentage rise in resonant frequency, the volume of the enclosure for a particular speaker can be determined with the aid of eqs (3.50) and (3.51).

The volume required for a given percentage rise, Δ, in resonant frequency is

$$V = \frac{0{,}87\varrho c^2 A^2}{S_s\left\{\left(1 + \frac{\Delta}{100}\right)^2 - 0{,}87\right\}} \text{ m}^3, \qquad (3.74)$$

where S_s is the stiffness of the speaker and A is the effective area of the cone.

3.10.2 ALLOWANCE FOR SPEAKER VOLUME

The method of mounting loudspeakers affects the volume of the enclosure. If they are attached to the back of the baffle board, they may reduce the volume significantly. If they are attached to the front the effect they have on enclosure volume depends on the size of the holes cut in the baffle board to accommodate them and the size of the speakers themselves.

3.11 Construction of enclosure

It is essential that the design of an enclosure incorporates maximum rigidity. If rigidity and airtightness are maintained the enclosure will perform according to its acoustic design characteristics. Panel vibration and poor sealing will introduce distortion and severely affect the power handling capabilities of the woofer, resulting in unbalanced speaker outputs.

Blockboard, plywood, and a wide variety of panel materials may be used. Solid timber is not recommended on account of warping. The corners may be jointed in any way preferred providing that the enclosure remains air-tight. Particular attention in this respect should be paid to the area around the input plug where loss of sealing can easily occur.

A variety of damping materials are available; glass wool, kapok and cotton waste have good absorption coefficients and an indication of the relative characteristics of a number of materials is given in Fig. 3.13. For use in all types of environmental conditions glass wool or fibre is preferred as variation in its characteristics is negligible. Although glass wool is about three times as expensive as kapok its use is more than justified on the grounds of stability. In view of its high absorption coefficient a thin layer can be used to line the inside of the enclosure. With small enclosures where the largest dimension is of the order of 450 mm, a 12 mm layer of absorbent material may be adequate.

4 Cross-over filters

The sound radiation from a moving coil loudspeaker becomes more directional with increasing frequency. Although the on-axis sound pressure might remain constant, the off-axis sound falls off at high frequencies. The advantage of a multi-way system is that the speakers can be driven in the region where they operate most satisfactorily. Outside that frequency region the signals can be channelled to loudspeakers designed to operate over the appropriate frequency range. To obtain the best performance from a multi-way system the electrical signal input to the system must be divided so that each speaker receives only the signals it is required to reproduce.

The individual frequency characteristic of the loudspeakers must overlap each other slightly, to maintain a continuous response throughout the audio range. The point at which signals cease being fed to one loudspeaker and are channelled to the loudspeaker reproducing signals in the adjacent frequency range, is called the cross-over frequency. When determining the cross-over frequency and the degree of attenuation either side of it, the frequency characteristics

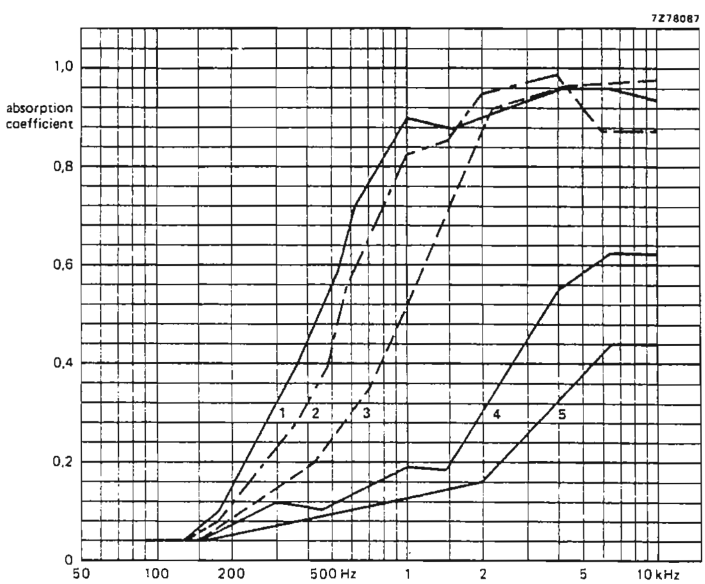


Fig. 3.13. Relative absorption of common damping materials. Curve 1, blown kapok. Curve 2, cotton waste. Curve 3, glass wool. Curve 4, wool waste. Curve 5, polyester fibre.

of the loudspeakers must be taken into account. It is important for each loudspeaker to generate the correct amount of acoustic energy for the part of the audio spectrum it is handling.

4.1 Design requirements

The design of a cross-over network depends on a number of factors:

- the desired cross-over frequency;
- the power handling capacity of the speakers;
- the directivity of the speakers and its frequency dependence;
- the sensitivity (efficiency) of the speakers;
- the desired frequency characteristic of the complete system.

Taking the last point first, the aim of the design is normally to obtain as linear a frequency characteristic as possible over the required frequency range. In combination with a linear amplifier characteristic, the overall frequency characteristic should be flat, with the amplifier tone controls at their neutral positions. In the following discussion it will be assumed that a flat characteristic is required.

4.2 Choice of cross-over frequency

The choice of cross-over frequency is a compromise between the frequency response characteristics of the individual speakers and the smoothness of transition between one speaker and another at the cross-over frequency. It also depends on the power requirements of the system. If at the cross-over frequencies of our choice, the maximum power handling capacity of one of the speakers is likely to be exceeded, adjusting the cross-over frequency can give us the safety margin required. The relationship between power distribution and cross-over frequency is explained in Sect. 3.8.2. Where the adjustment of cross-over frequency does not give the required safety margin, we must use a different loudspeaker with a higher power rating or two or more of the same type of loudspeaker suitably connected with regard to impedance.

There are two basic methods of signal division:

- electronic cross-over systems;
- passive filter networks.

Where an electronic cross-over system is used, individual adjustment of signal levels to each speaker is normally provided and correct tonal balance can be easily achieved. With a passive network the transition between speakers cannot be as smooth as with an electronic system, and a loss of 3 dB or more may occur at the cross-over frequency. Concerning ourselves primarily with passive networks on grounds of economy, the choice of cross-over frequency should be based on careful consideration of individual loudspeaker performance characteristics.

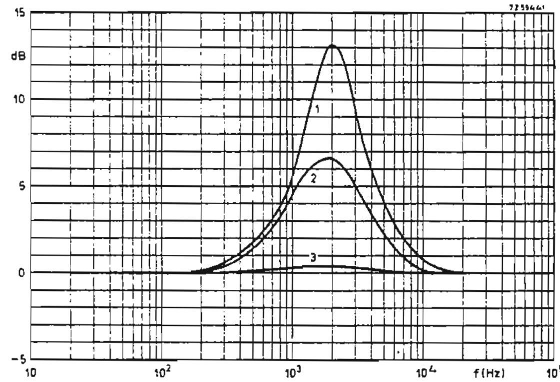


Fig. 4.1. Amplitude/frequency characteristic of a presence control. Curve 1, maximum; curve 2, half; curve 3, minimum.

Artificially boosting the mid-range at a selected frequency around 2000 Hz enhances the liveliness and presence of the reproduced sound. A presence control is sometimes fitted on high-fidelity amplifiers to provide a variable amount of lift in the response, as shown in Fig. 4.1. It is desirable to avoid discontinuities in the loudspeaker response under any circumstances, but if one occurs in the region of 2000 Hz it can be particularly annoying.

From the foregoing it can be concluded that, provided discontinuities can be avoided, the choice of cross-over frequency depends on the loudspeaker specifications mentioned earlier.

4.3 Types of passive networks

There are two basic types of passive filter network. The first consists of separate high-pass and low-pass filters arranged in series or parallel to provide a four-terminal output from a two-terminal input. The second, known as a constant resistance network, looks identical to the first but has different component values. The advantage of the constant resistance type is that not only does the input impedance remain constant over the frequency range but, in the case of 2-way speaker systems similar components can have similar values.

The 'classic' approach to filter design is based on transmission line theory, using a fictitious iterative impedance and iterative parameters. Iterative impedance is rather like the characteristic impedance of a transmission line. Terminating a filter with this impedance causes an identical impedance to appear, reflected, at the input. But in practice, characteristic impedance always has a real or complex value, and can even be made constant or frequency independent. Iterative impedance, on the other hand, cannot be simulated by any real impedance; at cut-off frequency it can be zero or infinity, in a passband it is real and resistive and varies in value, and in a stop band it is an imaginary, positive or negative reactance. In view of the similarity of characteristic and iterative impedance, it is mistakenly assumed that terminating the last section of a classic filter in a constant resistance will cause the correct impedance to be reflected back through any number of filters designed for the right value. However, in the vicinity of cut-off this is no longer true, and the use of conventional half-section low-pass and high-pass filters of the m-derived type has given way to the constant resistance types in high fidelity applications.

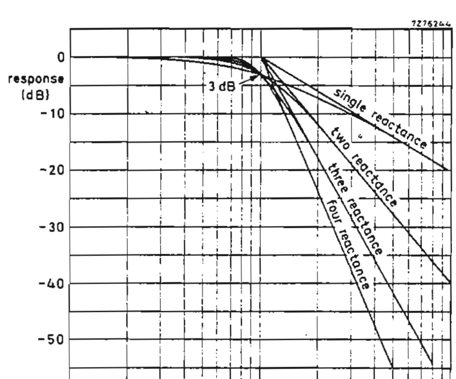


Fig. 4.2. Basic form of a constant resistance type response in a low-pass section of a cross-over filter according to the number of elements used.

While filters designed on the 'classical' basis require correct termination at both ends, constant resistance filters are not critical of input termination. If the outputs are correctly terminated, the input impedance is a constant resistance and the response will be unaffected by the source impedance. Since a modern solid-state amplifier can be considered as a constant voltage source having zero impedance, it follows that performance is unaffected by 'matching' amplifier output impedance and loudspeaker system input impedance. What is affected is the power conversion in the speaker system.

4.4 Constant resistance networks for two-way systems

Cross-over filter networks for high fidelity applications are characterized by the following features:

- attenuation at the cross-over frequency is 3 dB;
- the slope of the transfer characteristic at the cross-over frequency is half the ultimate slope;
- the ultimate slope is asymptotic to a straight line drawn at the cross-over frequency having a slope of 6 dB/octave multiplied by the number of reactive elements (see Fig. 4.2).
- when two filters having complementary characteristics are fed from a common source and the two outputs are correctly terminated, the total power at the outputs will be constant over the passband;
- when two complementary filters are correctly terminated, the impedance presented at their common input will be a constant resistance equal to each terminating resistance;
- the phase response at the cross-over frequency is half the ultimate value;
- the phase difference between the complementary outputs is constant, depending on the number of reactive elements.

The transfer characteristics of multi-reactance networks are illustrated in Fig. 4.2.

Constant resistance networks are derived from the circuits given in Fig. 4.3. If the component values are chosen to make $R_o = \sqrt{(L/C)}$, the impedance presented at the input terminals is constant and equal to R_o at all frequencies. At frequencies below the cross-over frequency, f_1,

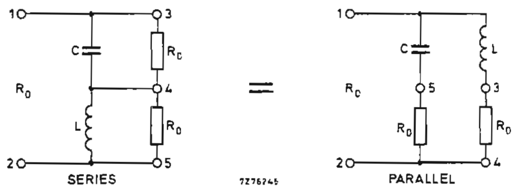


Fig. 4.3. When component values are selected to make $R_0 = \sqrt{(L/C)}$, the impedance presented at the input terminals is a resistance R_0.

all the input power is delivered to terminals 3 and 4; at frequencies above f_1, all the input power is delivered to terminals 4 and 5. At either side of frequency f_1 the slope of the attenuation characteristic approaches 6 dB/octave. This is normally too low to be of any real value, and can be improved by increasing the number of reactive elements in the filter section. Loudspeakers of current design normally require filters with an attenuation characteristic of 12 dB/octave for high fidelity applications.

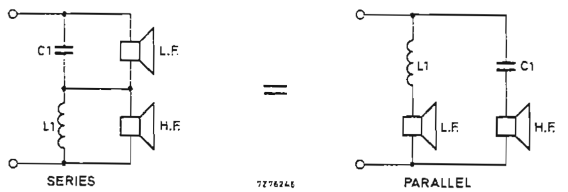
The values of the inductances and capacitances can be determined in the simple case of the 6 dB/octave filter by multiplying R_o and f_1:

$$R_o = \sqrt{\frac{L}{C}} \qquad f_1 = \frac{1}{2\pi\sqrt{(LC)}} \qquad R_o f_1 = \frac{1}{2\pi C}$$

whence

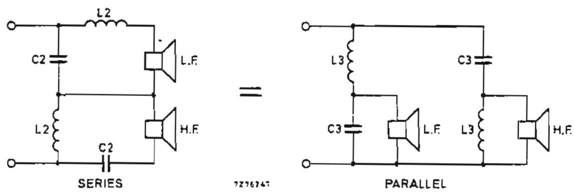
$$C = \frac{1}{2\pi f_1 R_o} \quad \text{and} \quad L = \frac{R_o}{2\pi f_1}.$$

In the case of single reactance filters, therefore, the reactance of each component is made equal to R_o at the cross-over frequency. For filters having two reactances per section (12 dB/octave types), the components have values that make their reactances equal to $R_o\sqrt{2}$ in the parallel case and $R_o/\sqrt{2}$ in the series case. This means that, in the same filter, both inductances have the same value and both capacitances have the same value. Figure 4.4 shows the arrangements for cross-over filters for two-way systems. Component values are given for different cross-over frequencies in Table 4.1 for 6 dB/octave filters, and in Table 4.2 for 12 dB/octave filters. Figure 4.5 shows two practical circuit arrangements where the cross-over frequency is 1000 Hz.



6 dB/octave

$$C_1 = \frac{1}{2\pi f_1 R_0} \qquad L_1 = \frac{R_0}{2\pi f_1}$$

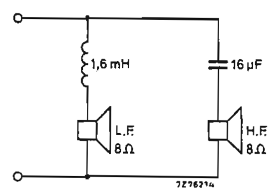


12 dB/octave

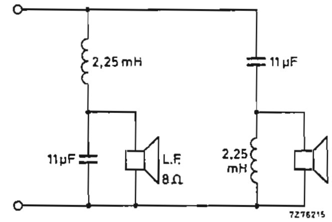
$$C_2 = \frac{\sqrt{2}}{2\pi f_1 R_0} \qquad C_3 = \frac{1}{2\pi f_1 R_0 \sqrt{2}}$$

$$L_2 = \frac{R_0}{2\pi f_1 \sqrt{2}} \qquad L_3 = \frac{R_0\sqrt{2}}{2\pi f_1}$$

Fig. 4.4. Constant resistance cross-over filter networks for 2-way systems with cross-over frequency f_1.



(a)



(b)

Fig. 4.5. (*a*) Cross-over filter for two-way system. Attenuation is 6 dB/octave (symmetrical). Cross-over frequency is 1000 Hz. (*b*) Cross-over filter for two-way system. Attenuation is 12 dB/octave (symmetrical). Cross-over frequency is 1000 Hz.

Table 4.1 Component values or the 6 dB/octave filters of Fig. 4.4

f_1 (Hz)	R_0* (Ω)	L_1 (mH)	C_1 (μF)
	5	1,6	64
500	10	3,2	32
	20	6,4	16
	5	1,1	45
700	10	2,3	23
	20	4,5	11
	5	0,8	32
1000	10	1,6	16
	20	3,2	8
	5	0,7	26
1200	10	1,3	13
	20	2,6	7
	5	0,5	20
1600	10	1,0	10
	20	2,0	5
	5	0,4	16
2000	10	0,8	8
	20	1,6	4
	5	0,3	13
2400	10	0,7	7
	20	1,3	3

Table 4.2 Component values for the 12 dB/octave filters of Fig. 4.4

f_1 (Hz)	R_0* (Ω)	L_2 (mH)	C_2 (μF)	L_3 (mH)	C_3 (μF)
	5	1,1	90	2,2	45
500	10	2,2	45	4,5	22
	20	4,5	22	9	11
	5	0,8	64	1,6	32
700	10	1,6	32	3,2	16
	20	3,2	16	6,4	8
	5	0,5	45	1,1	22
1000	10	1,1	22	2,2	11
	20	2,2	11	4,5	5,5
	5	0,47	37	0,94	19
1200	10	0,94	19	1,87	9,4
	20	1,87	9	3,75	4,7
	5	0,35	28	0,7	14
1600	10	0,7	14	1,4	7
	20	1,4	7	2,8	3,5
	5	0,28	22	0,56	11
2000	10	0,56	11	1,1	5,5
	20	1,1	5,5	2,2	2,8
	5	0,23	19	0,47	9,4
2400	10	0,47	4,7	0,94	4,7
	20	0,94	9,4	1,87	2,3

4.5 Networks for three-way systems

Networks for three-way systems require an additional filter section for the mid-range loudspeaker. Here two cross-over frequencies are involved. One at which the woofer rolls off and the squawker rolls on, the other at which the squawker rolls off and the tweeter rolls on. For symmetrical filters, in which roll-on and roll-off slopes are alike, the component values for passive filter networks of constant resistance design may be calculated from Fig. 4.6.

An alternative approach is to use two 2-way filters together to form a 3-way filter. The principle is shown in Fig. 4.7. The frequency at which the woofer rolls off is the cross-over frequency of filter A. Signals of a higher frequency are fed to the input of filter B via the 'tweeter' terminals of filter A. This higher range of frequencies is divided between the squawker and the tweeter in accordance with the cross-over frequency of filter B. The simplicity of this method is clear and excellent results are easily obtained.

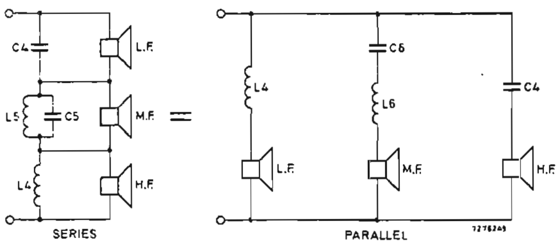
4.6 The effect of loudspeaker impedance

We have assumed so far that a loudspeaker presents a constant and purely resistive load to the cross-over filter. In practice the load will vary with frequency due to the inductance and motional impedance of the voice coil.

At resonance the impedance is high, falling sharply as frequency increases then rising slowly again. The effect of the changing impedance on the filter characteristic coupled with the frequency response of the loudspeaker suggests that a smoother overall transition through the cross-

* Corresponding to nominal loudspeaker impedances of 4 Ω, 8 Ω and 16 Ω respectively.

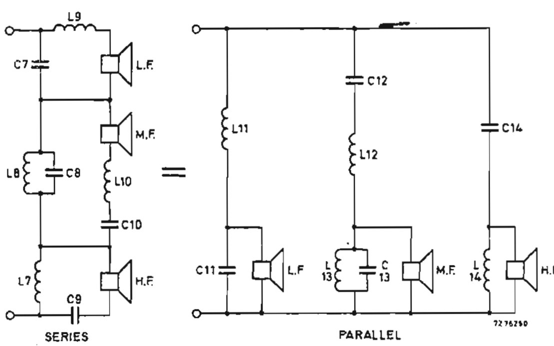
over region could be obtained if the low-frequency speaker were made to roll-off at 6 dB/octave and the high-frequency speaker were made to roll-on at 12 dB/octave. A filter with different roll-on and roll-off slopes is said to be asymmetric. There is another means of compensating for the variation of loudspeaker impedance.



6 dB/octave

$$L_4 = L_5 = \frac{R_0}{2\pi f_1} \qquad L_6 = \frac{R_0}{2\pi f_2}$$

$$C_4 = C_5 = \frac{1}{2\pi f_2 R_0} \qquad C_6 = \frac{1}{2\pi f_1 R_0}$$



12 dB/octave

$$C_7 = C_{10} = \frac{\sqrt{2}}{2\pi f_1 R_0} \qquad C_8 = C_9 = \frac{\sqrt{2}}{2\pi f_2 R_0}$$

$$L_7 = L_{10} = \frac{R_0}{2\pi f_2 \sqrt{2}} \qquad L_8 = L_9 = \frac{R_0}{2\pi f_1 \sqrt{2}}$$

$$C_{11} = C_{12} = \frac{1}{2\pi f_1 R_0 \sqrt{2}} \qquad C_{13} = C_{14} = \frac{1}{2\pi f_2 R_0 \sqrt{2}}$$

$$L_{11} = L_{13} = \frac{R_0\sqrt{2}}{2\pi f_1} \qquad L_{12} = L_{14} = \frac{R_0\sqrt{2}}{2\pi f_2}$$

Fig. 4.6. Constant resistance cross-over filter networks for three-way systems.

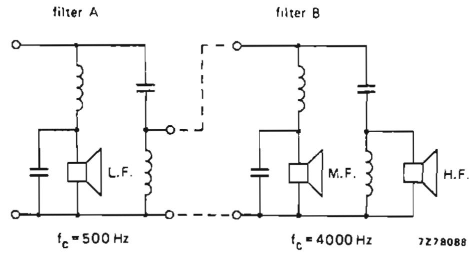


Fig. 4.7. Alternative approach to the design of three-way filters.

The loudspeaker impedance seen by the cross-over network can be made fairly constant by using the compensating network illustrated in Fig. 4.8. The impedance of both speakers rises with increasing frequency and appears more inductive. The impedance can be made resistive by adding a series *RC* circuit in parallel with the loudspeaker. Although this is not perfect, calculation and experiment enable a value of *R* and *C* to be chosen which will produce a reasonably constant load at least one octave either side of cross-over which is the most critical region.

A method of using the inductive component of the loudspeaker impedance may be practised where an inductor is in series with a loudspeaker as shown in Fig. 4.9. The value of the inductor can be reduced by an amount equal to the inductive component of the voice coil impedance measured at the cross-over frequency. This brings about a reduction in the value of the filter component and the constant resistance properties of the cross-over filter can be maintained.

4.7 Phase response

It is essential that the phase response of a cross-over filter is carefully considered. The cross-over network, composed of reactive elements, introduces phase changes into the system. In single-element sections, as the frequency increases from the cross-over frequency the phase change between the input and the low-frequency output approaches −90°, while the high-frequency output tends to become in phase with the input. As the frequency decreases from the cross-over frequency the phase of the high-frequency output approaches +90° relative to the input, while the low-frequency output tends to become in phase with the input. As can be seen in Fig. 4.10, the phase difference between input and outputs at the cross-over frequency is 45° and there is a constant phase difference of 90° between the two outputs.

Figure 4.11 shows that the situation is similar for 12 dB/octave filters employing two-element sections but the difference between the outputs is 180° and there is a phase difference of 90° between the input and the outputs at the cross-over frequency.

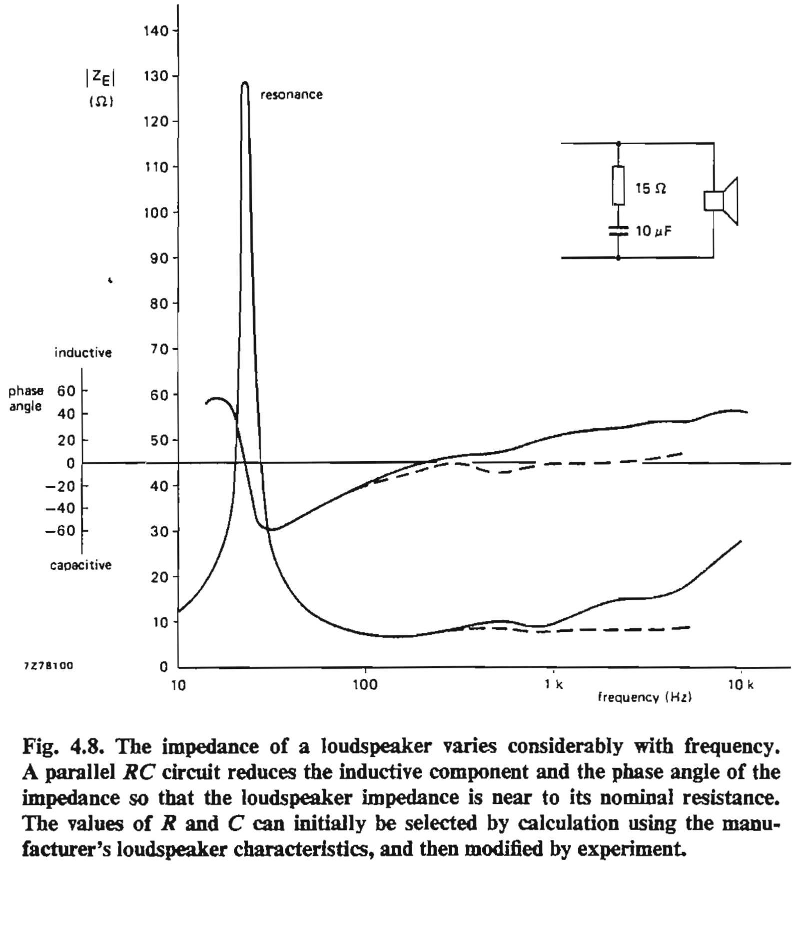


Fig. 4.8. The impedance of a loudspeaker varies considerably with frequency. A parallel *RC* circuit reduces the inductive component and the phase angle of the impedance so that the loudspeaker impedance is near to its nominal resistance. The values of *R* and *C* can initially be selected by calculation using the manufacturer's loudspeaker characteristics, and then modified by experiment.

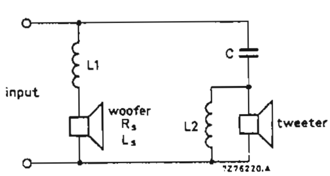


Fig. 4.9. Inductor L_1 is in series with the voice coil of the woofer. The value of the component can be reduced in practice to a value L_1' where $L_1' = L_1 - L_S$. Inductance L_S is that of the voice coil at cross-over frequency. L_1 is the theoretical value calculated during the filter design assuming a resistive filter termination.

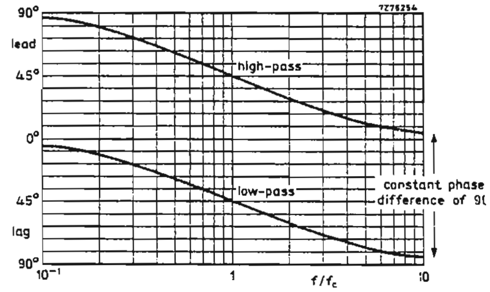


Fig. 4.10. Phase difference of outputs relative to input for a single-element section filter terminated by the appropriate resistance.

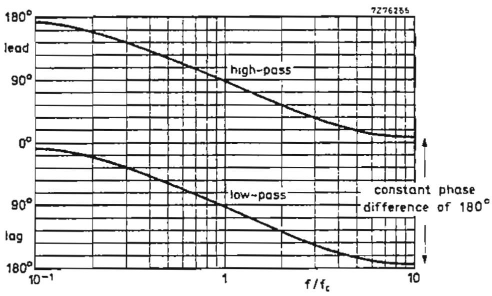


Fig. 4.11. Phase difference between outputs and input of a two-element cross-over network terminated with the appropriate resistance.

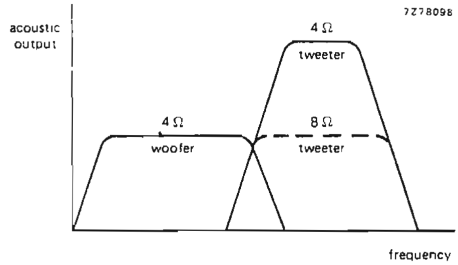
Something must be done therefore to prevent anti-phase conditions leading to acoustic cancellation. With a network having 180° phase difference between its outputs it is a simple matter to reverse the connections to one of the speakers. Electrically the voice coils will be fed in anti-phase but, as one is reversed, the cone motions are in phase.

Since the matter of phase is of such importance it is usual to give an indication of the polarity of the voice coil terminals. Philips loudspeakers have one terminal marked with a red dot. When a d.c. voltage is applied to the voice coil so that the red connection is positive, the voice coil will move forwards.

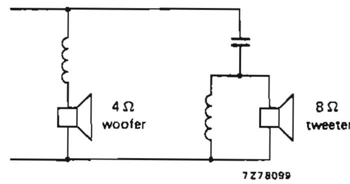
The curves of Figs 4.10 and 4.11 only apply when the networks are resistor terminated. It follows that all we have said about phase response will be modified by the phase angle (the phase difference between the voltage across the voice coil and the current through it) of the loudspeaker impedance at the cross-over frequency. An example of the variation of the angle and magnitude of loudspeaker impedance with frequency is shown in Fig. 4.8.

4.8 Obtaining a smooth response

In order to obtain a smooth transition from one speaker to another it is important to select speakers with sensitivities within about 2 dB of each other. A greater difference will cause an audible step in the response as shown in Fig. 4.12(*a*).



(*a*)



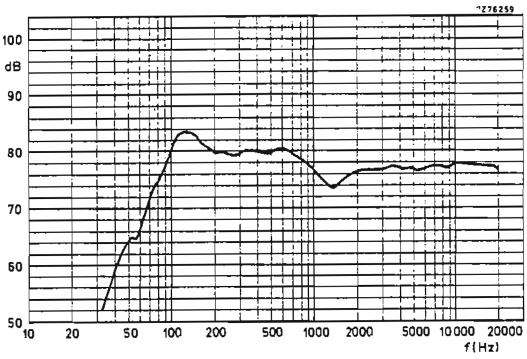
(*b*)

Fig. 4.12. Speakers with different sensitivities when used in the same system give rise to a step in the overall system response (*a*). This can be overcome by exchanging the higher sensitivity speaker with a similar one of a greater impedance (*b*). The system has a smoother overall response and an increased power handling capacity.

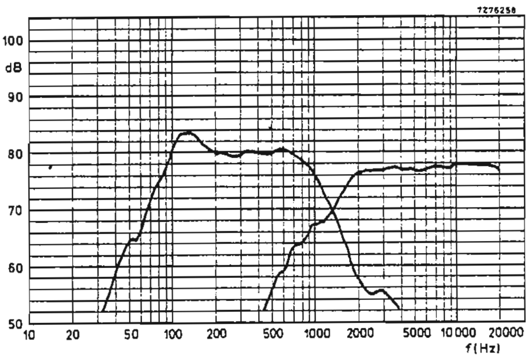
Where speaker sensitivities do differ and a speaker of the same sensitivity as those in the rest of the system cannot be obtained, the higher sensitivity speaker can be exchanged for a similar speaker of higher impedance (see Fig. 4.12(*b*)). This will then operate at a reduced power level and acoustic output. The same effect can be achieved by inserting a low-value resistor, of a suitable power rating, in series with the higher sensitivity speaker. A combination of both methods can be used.

It is also necessary to ensure that one speaker is capable of taking over the signal from the other at the cross-over frequency. If two speakers have responses which do not overlap appreciably, a gap will occur in the overall response when they are connected through a filter which introduces a further 3 dB attenuation at cross-over. This is shown in Fig. 4.13.

The ultimate aim of cross-over filter design is to obtain the best matching of the acoustic characteristics of the different speakers in the system.



(a)



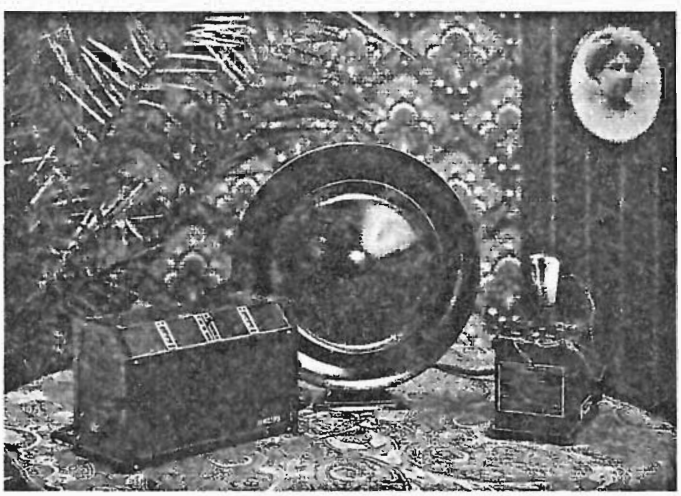
(b)

Fig. 4.13. (*a*) Frequency response of two speakers with insufficient overlap in their characteristics. (*b*) Combined response, note the dip in the curve at cross-over.

This may necessitate some modifications to the standard types of filter already discussed. An example of this is illustrated in Figs 4.14 and 4.15. The sound pressure curves of a woofer and tweeter, measured on axis, are shown in Fig. 4.14(*b*). Using a conventional type of filter with a cross-over frequency of 2000 Hz leads to a system response curve with a peak in the cross-over region because the frequency/response characteristics of both speakers are relatively high in this region. This is illustrated in Fig. 4.14.

The peak can be avoided by choosing an earlier roll-off for the woofer and a later roll-on for the tweeter. This is illustrated in Fig. 4.15 and it is clear that there is now a distinction between the −3 dB points of the filter and the cross-over frequency of the filter with a purely resistive termination (2000 Hz) or with a loudspeaker termination (1700 Hz).

A cross-over filter should be designed in conjunction with the speakers and the enclosure to be used. Following preliminary analysis and the construction of a prototype, the system should be tested with the cross-over filter outside the enclosure. Changes in filter design or the reversal of speaker connections can then be made between tests until the best results are obtained.



In contrast to modern loudspeakers, that shown above is a moving iron type with a polarized magnet system and has an impedance of 1500 to 2000 Ω. Beside it are the receiver unit (left) and the 150 V HT supply unit (right); the complete broadcast receiver as built by Philips in 1927.

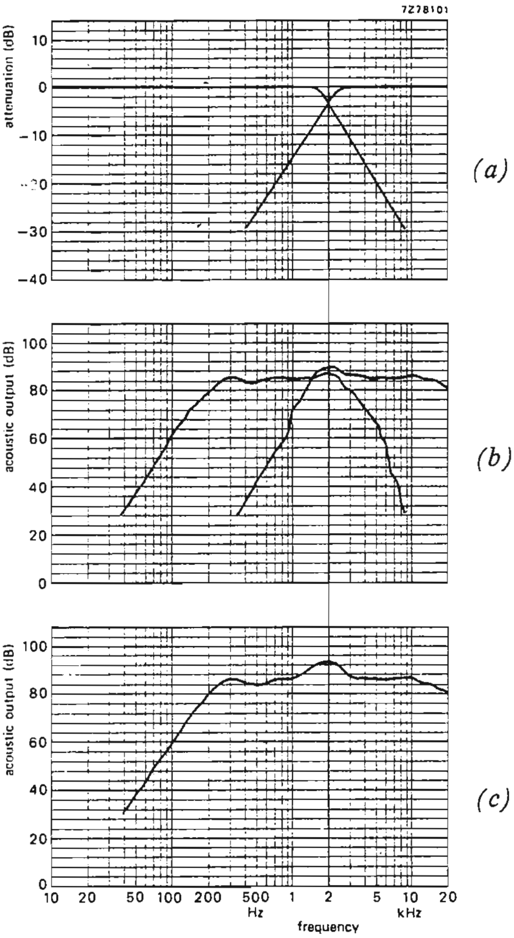


Fig. 4.14. (*a*) Ideal characteristic of a cross-over filter. (*b*) Frequency response of two speakers both showing a peak in the cross-over region. (*c*) Combined response of the two speakers using the cross-over filter of (*a*), note peak at cross-over.

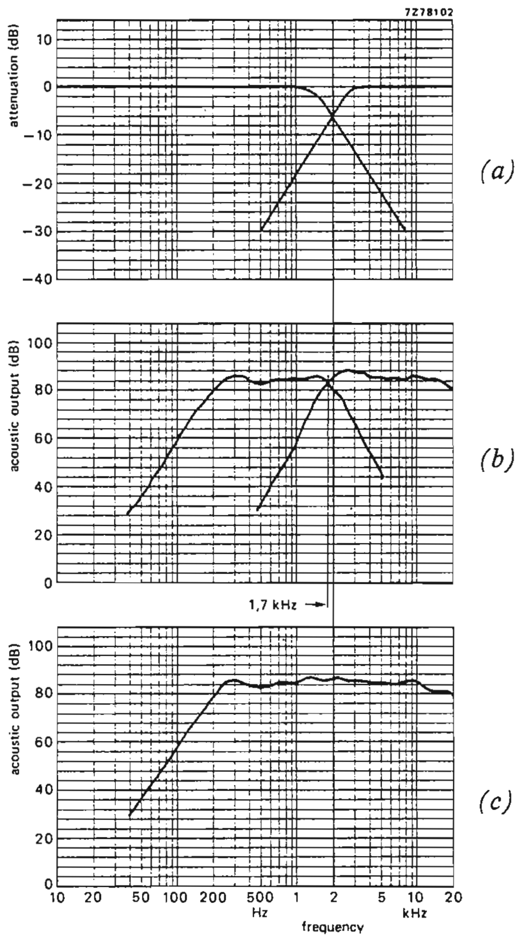


Fig. 4.15. (*a*) Characteristic of filter of Fig. 4.14(*a*) redesigned to have an earlier roll-off for the woofer and a later roll-on for the tweeter. (*b*) The cross-over frequency of the filter terminated by the speakers becomes 1700 Hz and the overall frequency response of the system, (*c*), is much smoother.

Designing hi-fi speaker systems - part 3

D. Hermans*

This is the last of a series of articles on sealed-enclosure loudspeaker systems. The first two parts dealt with the mechanical and electrical design of a moving-coil direct-radiator loudspeaker, its performance both unmounted and mounted in a sealed enclosure, the use of two or more loudspeakers in a multi-way system and the cross-over networks demanded by such systems. In part 3 we discuss the measurement and specification of loudspeaker/loudspeaker system performance and include a section on digital measurement technique. The series ends with some discussion of direct and reflected sound, listening room and hall acoustics.

5 Specifications and measurements

5.1 National and international standards

In Europe, DIN 45573 and DIN 45500 are the standards mainly used to define hi-fi. A small label, affixed to hi-fi equipment is commonly used to indicate that the appropriate standard has been complied with. It is not proposed to provide a translation of these standards in this publication but a number of interesting points arising from them will be discussed.

High fidelity standard DIN 45500 defines the requirements for frequency response measured in a half-free (2π steradian) field by the use of a standard curve (Fig. 5.1). When a frequency response curve has been determined for a loudspeaker system, the curve of Fig. 5.1 is overlaid on it. The two curves must obviously be drawn to the same scale. The middle line of the standard is adjusted to the average of the loudspeaker system response and the system response is defined as conforming to the hi-fi standard if it lies within the upper and lower limits of the standard.

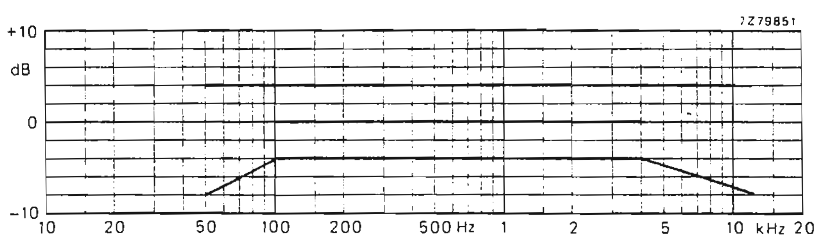


Fig. 5.1. High fidelity standard DIN 45500. The loudspeaker response should lie between the lower and upper limits when its mean level coincides with the central straight line.

* Philips Loudspeaker Development Laboratories, Dendermonde, Belgium.

Fig. 5.2 shows the frequency response curve of a typical 3-way sealed enclosure loudspeaker system. Figure 5.3 shows that the system conforms to DIN 45500.

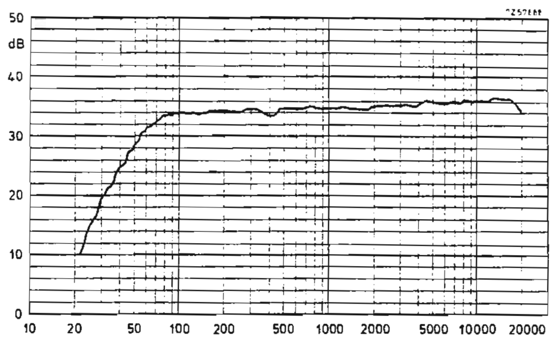


Fig. 5.2. Response curve of a high fidelity, multi-way, sealed enclosure, loudspeaker system.

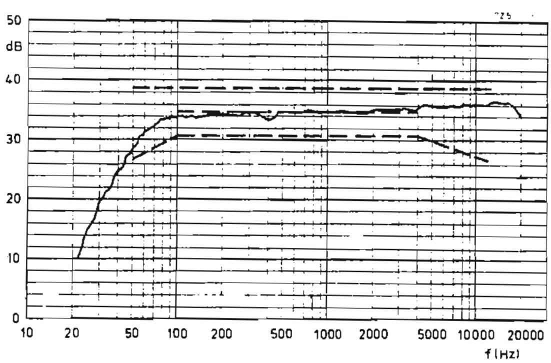


Fig. 5.3. The response curve of Fig. 5.2 superimposed on the DIN 45500 characteristic (Fig. 5.1) shows that the loudspeaker system conforms to the high fidelity standard.

The standard curve is often drawn on a transparency or a transparent plastic sheet for easy use in the laboratory. It can then be laid directly over a loudspeaker system response on recorder paper, provided both the standard and the response are to the same scale.

There are many types of loudspeakers and loudspeaker systems presenting a wide choice of performance intended for different acoustical environments. Predictably, various test methods have evolved to test the loudspeaker in various environments. Two conditions are often referred to

(1) The *half-free field* or *2π steradian field* in which the acoustical conditions on the forward side of the loudspeaker approach those of free space. The condition is illustrated in Fig. 5.4 where the loudspeaker is baffle mounted and fixed in the ground in an anechoic environment.

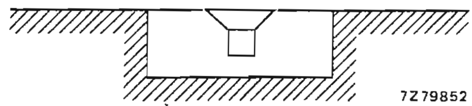


Fig. 5.4. Half-free field measurement on a single loudspeaker.

(2) The *free-field* or *4π steradian field* is defined in IEC publication 268-5 (sub-clause 3.3). The acoustical conditions approach those of free space.

In both cases the measurement is performed in anechoic conditions so that they reflect the intrinsic characteristics of the loudspeaker, unaffected (as far as possible) by reflections. However, it is possible to obtain pure loudspeaker characteristics from measurements in a normal environment where some reflection is present by using modern signal processing techniques. This is discussed in Section 6.

The half-free field and free field are widely used standard conditions and, in fact, conversion can be made between the two methods. Fig. 5.5 shows a curve which may be used, under certain circumstances, to convert a 4π steradian characteristic measured on-axis on a sealed enclosure or bass reflex loudspeaker system to the equivalent 2π steradian characteristic for the system. The conditions under which the conversion is valid are that the enclosure has a maximum front area of about 30×60 cm, the smallest dimension of which (d) is used to derive the transition frequency $f_0 = 120/d$ Hz. The dimension d must be measured in metres and the sealed enclosure should not exceed a volume of 40 to 50 litres. It is also important that where a conversion has been used to obtain a particular type of characteristic, this is explained by the manufacturer concerned.

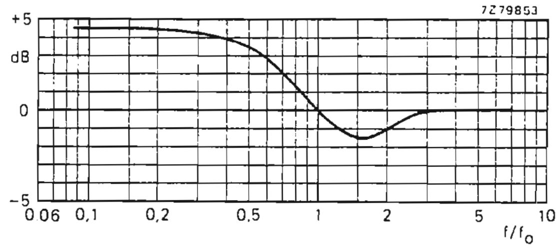
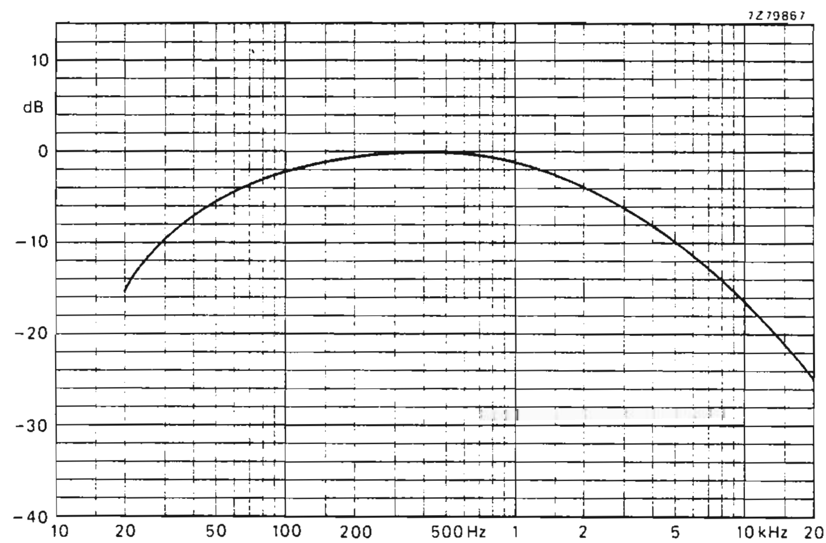


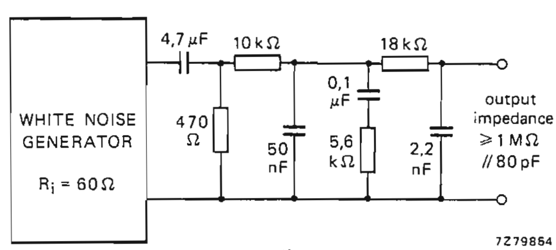
Fig. 5.5. In some circumstances a loudspeaker response curve measured in a 4π steradian field can be converted to the equivalent half-free (2π steradian) field curve using the curve shown above.



(a)

The determination of power handling capacity has been discussed in Sect. 3.8.6. The appropriate weighted signal is derived from a source of 'white noise' which is filtered to give the required frequency distribution. The response of the filter is described in DIN 45573 and illustrated in Fig. 5.6. The standard is also under consideration by the IEC.

The frequency range of a system is defined as the range between the bass and treble frequencies which are 8 dB below the mean response.



(b)

Fig. 5.6. (a) Relative amplitude of frequencies in a noise signal used for testing loudspeakers. (b) The filter necessary to derive this response from a "white" noise signal.

5.2 Standard test conditions

The standard atmospheric conditions for testing are:

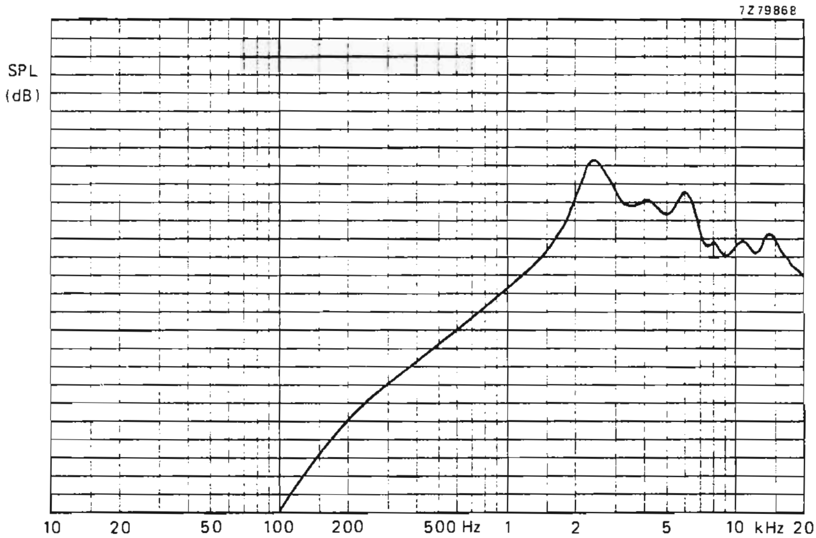
Temperature, 20-25 °C
Relative humidity, 45-75%
Atmospheric pressure,
860-1060 mbar (86-106 kPa)

The acoustic environments can, as has already been stated, take different forms.

5.2.1 UNMOUNTED TESTS

An easy and reproducible test is to measure the response of an unmounted loudspeaker in an anechoic room. The test lends itself easily to quality control measurements. A response curve obtained using this method to test a 4-inch, full-range speaker is given in Fig. 5.7(a).

(a)



(b)

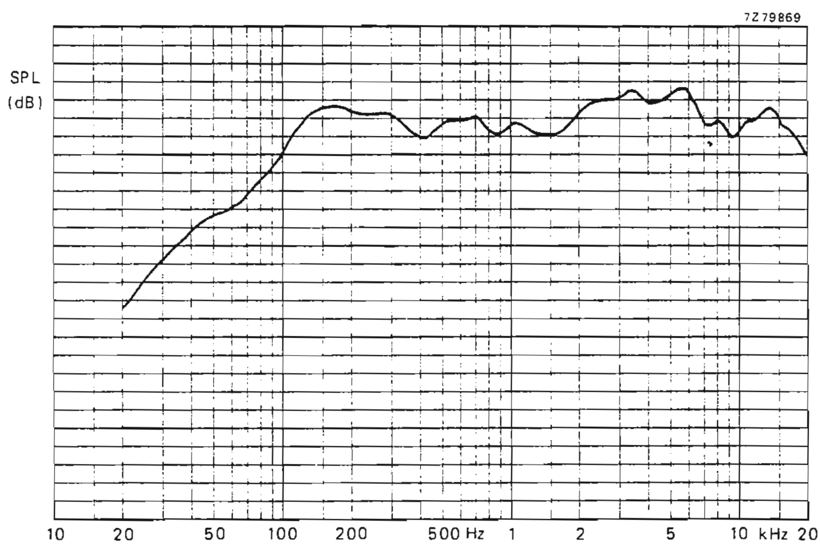


Fig. 5.7. The frequency response of a 4-inch full-range loudspeaker type AD40500M4. Curve (a) is a 4π steradian measurement made with the loudspeaker unmounted. Curve (b) is a 2π steradian measurement made with the loudspeaker mounted on an IEC baffle. Both are on-axis characteristics measured in an anechoic room.

5.2.2 MOUNTED TESTS

Loudspeakers can be tested mounted on a baffle or in a sealed enclosure under half-free or free field conditions. Figure 5.7 offers a comparison between the unmounted test and the result when the same loudspeaker is mounted on an IEC baffle. There are also tests which seek to simulate the response of a mounted loudspeaker in a typical setting - a living room for example.

Figure 5.8 enables some comparison to be drawn between a sealed enclosure mounted loudspeaker system in an anechoic room, in a reverberant room, in a half-free field condition and in a simulated living room environment.

For various reasons, the measurements were made using different power levels and different microphone-speaker distances so a direct comparison is not really possible but the effects of the different environments are clear enough. The familiar anechoic room response (a) is completely broken up by reflections in a reverberant room (b) and bears some comparison to the effect obtained when performing the measurement in a normal living room (d). The 2π steradian field measurement (c) exhibits a drop in response around 400 to 500 Hz.

A typical living room is, of course, rather difficult to define and no generally accepted standard has yet emerged. The results of living room measurements should be accompanied by a rather fuller description of the test set-up than would be necessary were one able to quote a suitable, universally acclaimed, specification. One aspect that certainly has a marked effect on the measurements obtained is the deployment of the loudspeaker system with respect to the walls, floor and ceiling of the room. Figure 5.9 compares on-axis sound pressure level/frequency characteristics measured with a loudspeaker mounted in several different positions in a living room type environment. The loudspeaker mounted on the floor (a) shows some boost in the lower frequency zone around 60 to 70 Hz and around 200 Hz, when compared to measurements made with the loudspeaker mounted 0,6 metres off the floor (dashed line). With the speaker moved back against the wall (b) the speaker characteristics shows further boost at the lower frequencies, which is increased again when the speaker is set in the corner of the room (c). The sound is reflected from adjacent surfaces forming 'image' sources so that the effect is of a larger more directional source particularly as far as the low frequencies are concerned.

The 'living room' conditions used by Philips for loudspeaker tests are the result of an exhaustive study of conditions in genuine living rooms. The measurement conditions are designed to avoid the setting up of standing waves and the reverberation time is an average taken from many measurements made in genuine living rooms. Although, of course, one would be very unlikely to come across the average living room, the simulation of an average living room is a useful exercise. Figure 5.10 is the result of reverberation measurements in many European living rooms.

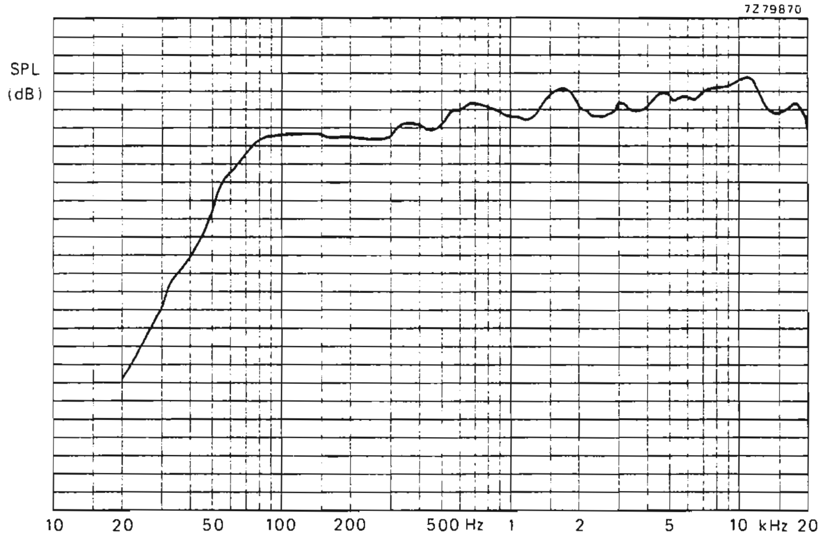
5.3 Resonance frequency and rated impedance

The simple circuit arrangement shown in Fig. 5.11 is used to measure both the resonance frequency and the rated impedance. A one ohm resistor is connected in series with the loudspeaker, or system, and a variable frequency signal generator. The frequency is swept slowly from 0 to 20 000 Hz and the resonance frequency is indicated by the first minimum voltmeter reading.

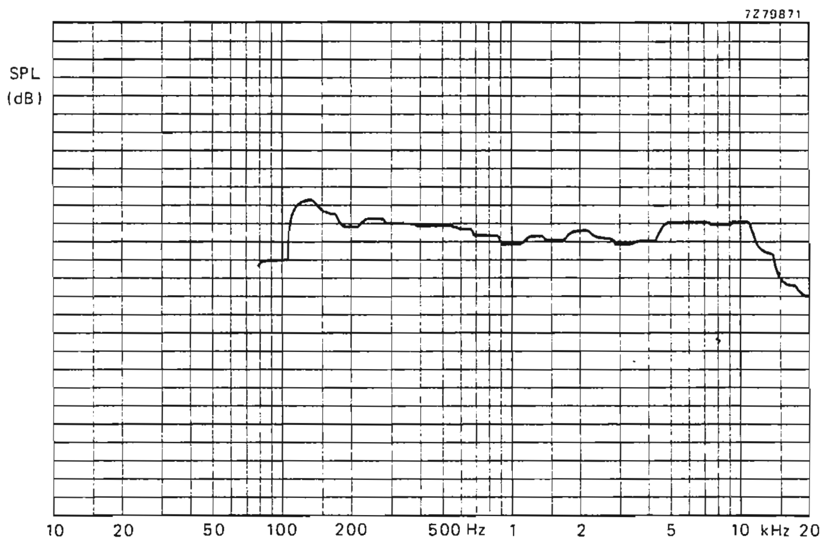
The rated impedance ascribed to a particular loudspeaker type by a manufacturer, is a nominal value. The modulus of the lowest value of electrical impedance measured on a loudspeaker at any frequency above the bass resonance frequency is no less than 80% of the loudspeaker's rated impedance. A typical speaker impedance curve is shown in Fig. 5.12 illustrating the point at which speaker impedance is measured.

5.4 Speech coil resistance

The speech coil resistance is normally measured with a d.c. ohmmeter and will normally be about 15 to 20% lower than the rated impedance.

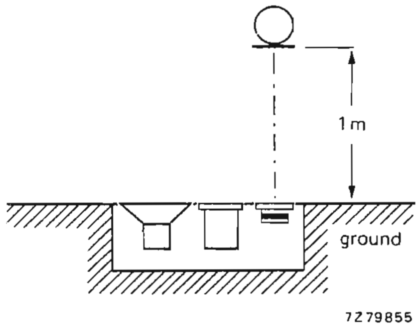


(a)



(b)

Fig. 5.8. Frequency response of a complete, 3-way high fidelity sealed enclosure loudspeaker system measured under various conditions. (a) Characteristic obtained in an anechoic room. (b) The characteristic obtained in the reverberation room. (c) The half-free field measurement. (d) The living room measurement.



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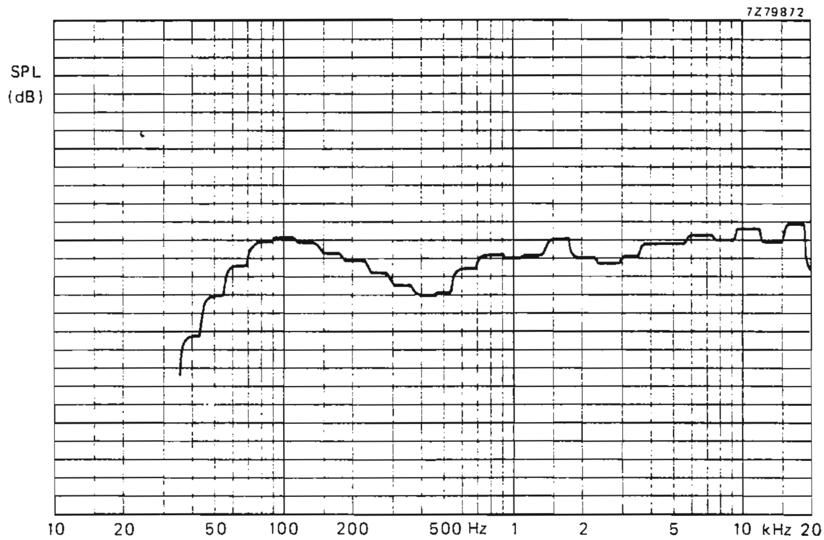
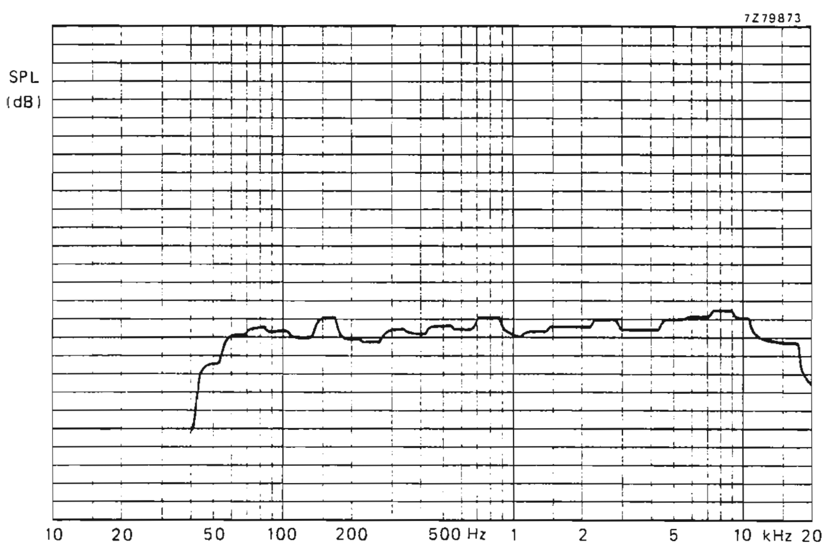
ground
7Z79855



7Z79872
SPL
(dB)
10
20
50
100
200
500 Hz
1
2
5
10 kHz 20

(c)



7Z79873
SPL
(dB)
10
20
50
100
200
500 Hz
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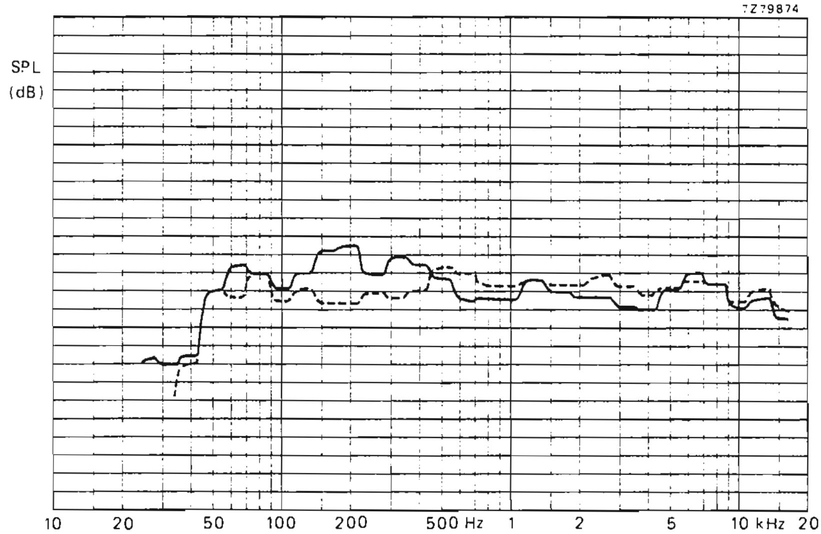
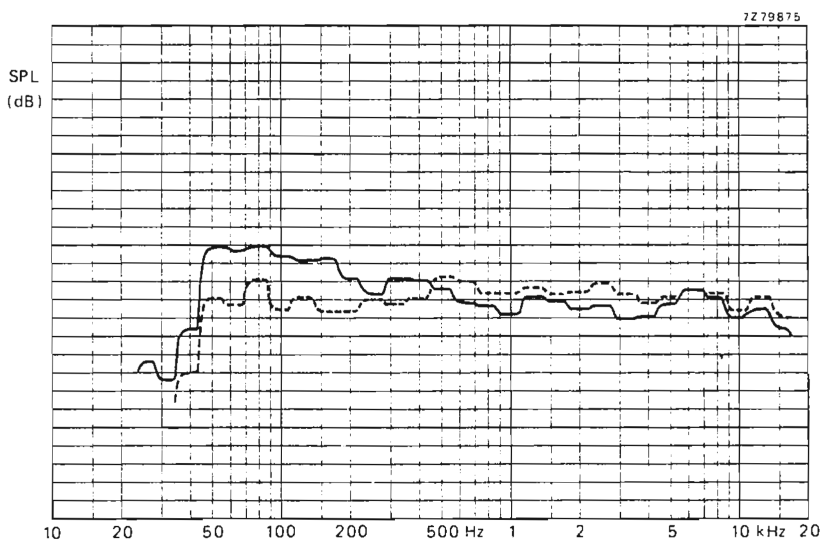
2
5
10 kHz 20

(d)

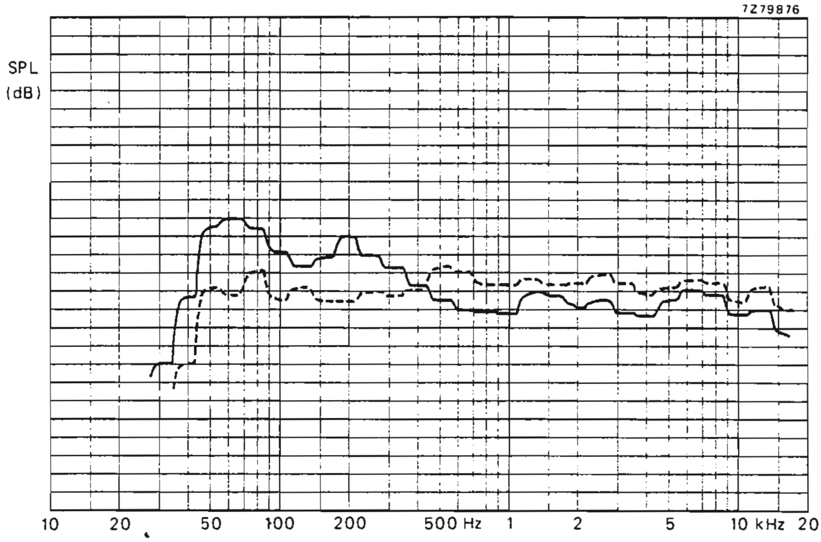


(a)



(b)

Fig. 5.9. The most noticeable effect of the proximity of walls and floors etc. is base lift. The characteristic of a loudspeaker placed 0,6 m from the floor and 0,8 and 2 m from the walls is shown as a dashed line for comparison with the curve obtained when it is (a) moved down onto the floor (b) moved back against the wall and (c) moved into the corner of a room.



(c)

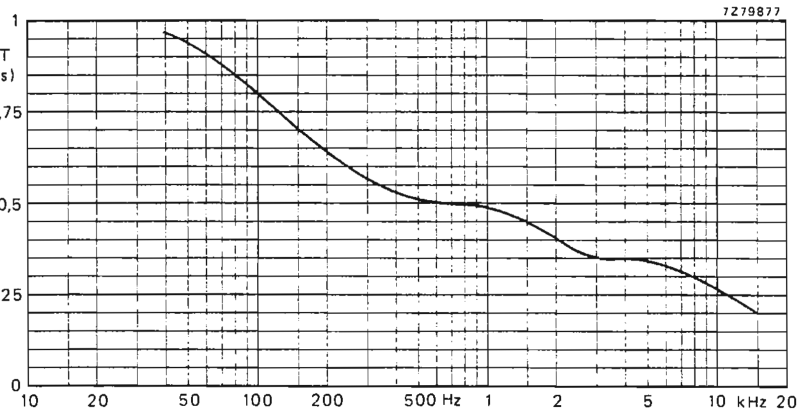


Fig. 5.10. Average reverberation time in living rooms derived from many measurements made in European living rooms.

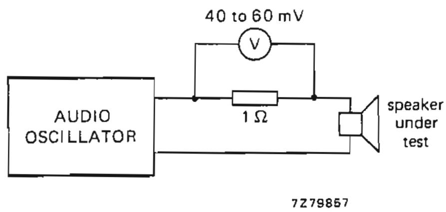


Fig. 5.11. The measurement of resonance frequency and rated impedance can be determined with this simple circuit. The oscillator should have a low output impedance.

5.5 Magnetic flux density

The magnetic flux density in the air gap is measured over the air-gap height with a differential search coil and a reflecting galvanometer.

5.6 Distortion

To determine distortion, a sinusoidal signal is applied to the loudspeaker so that an average SPL of 96 dB relative to 2×10^{-4} μbar (20 μPa) is obtained between 100 and 8000 Hz at a distance of 1 metre. The fundamental frequency is filtered out and the resulting signal represents the distortion. The nature and causes of loudspeaker signal distortion are discussed in Section 2.11.

5.7 Directivity

Loudspeaker directivity, and its measurement, have been discussed earlier (Section 2.10). The loudspeaker, clamp-mounted on the centre of a turntable, is energized by a constant-frequency constant-amplitude signal and the loudspeaker is rotated through 360° while the microphone, fixed at a distance of 1 metre from the loudspeaker, feeds the varying acoustic response to a polar recorder. Unmounted, the loudspeaker exhibits the effects of acoustic short-circuiting at low frequencies and increasing directivity at high frequencies. Figure 2.15 illustrates these effects clearly for a quality full-range speaker. Figure 5.13 illustrates the polar response of a 1-inch dome tweeter at five different frequencies.

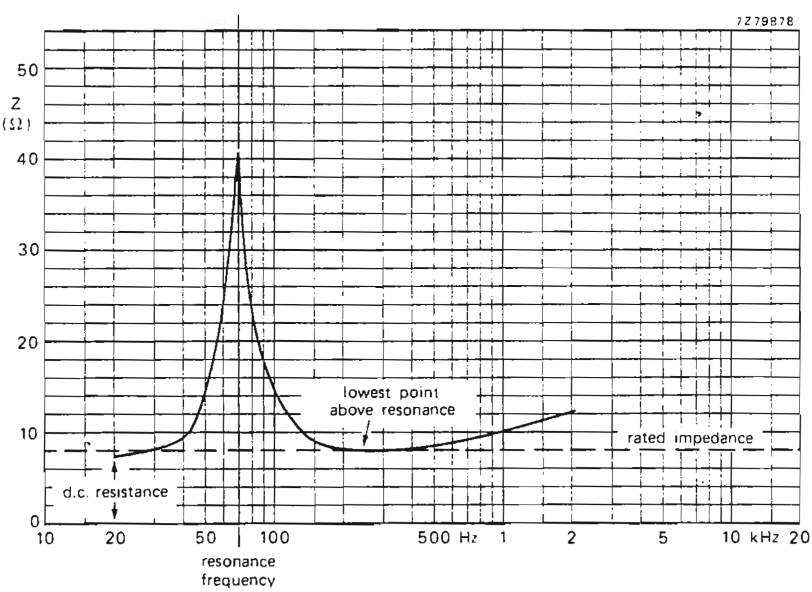
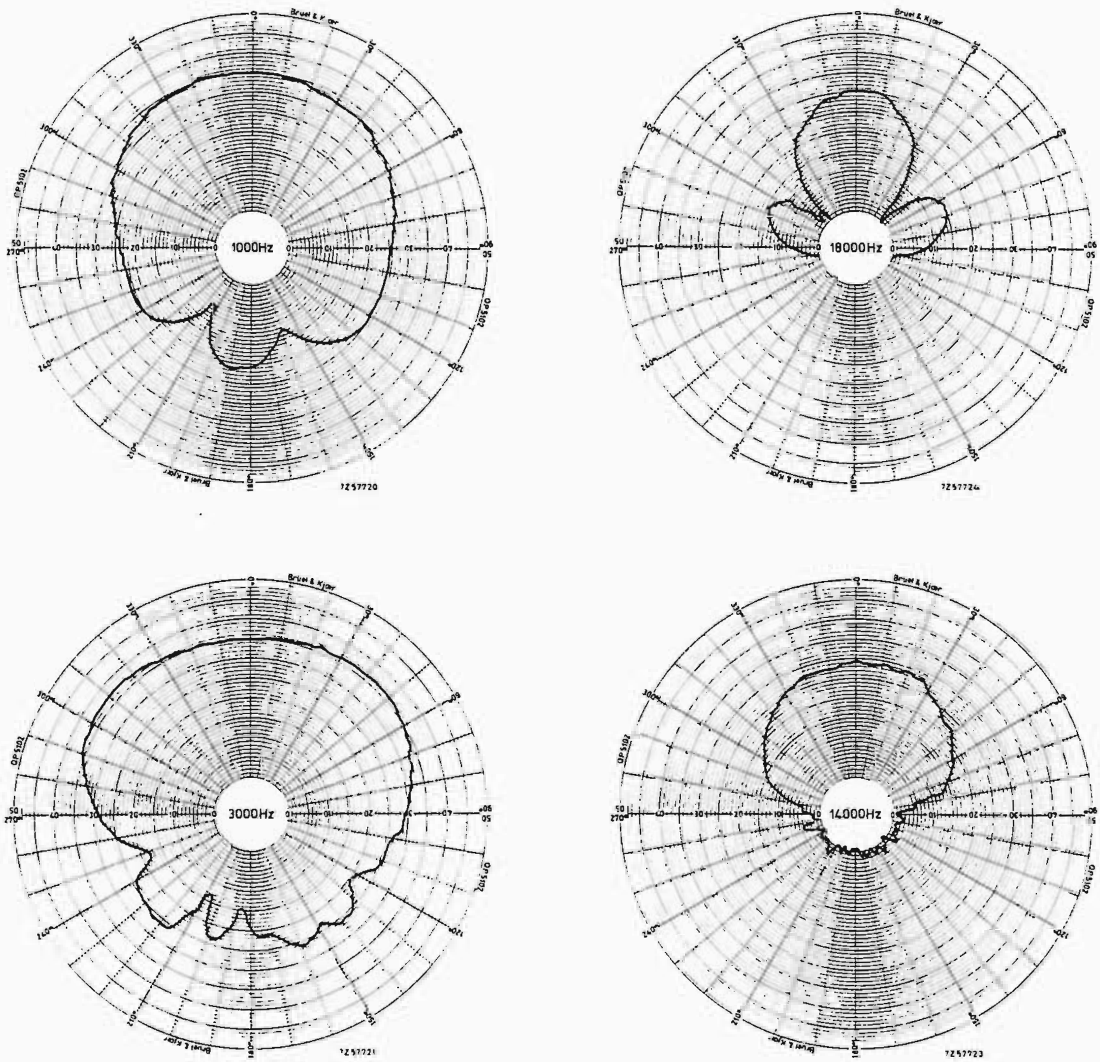


Fig. 5.12. Rated impedance is the modulus of the lowest value of electrical impedance above the resonance frequency.



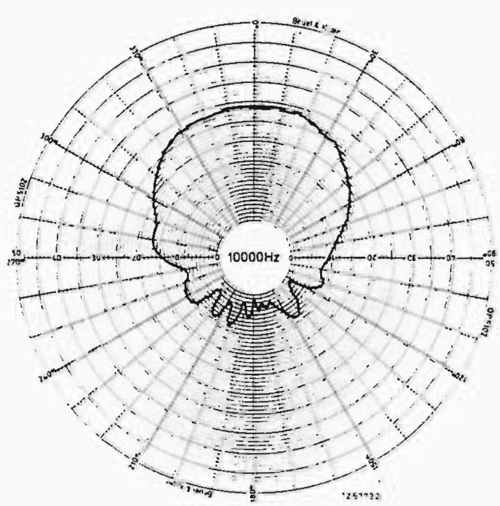


Fig. 5.13. Polar response of a 1-inch dome tweeter at five different frequencies.

Again directivity increases with frequency and the polar response at 18 000 Hz shows marked directivity, two well-defined side lobes and no measurable back radiation. At 1000 Hz the polar response is far more even, being at a maximum at the front of the loudspeaker with two minima at the rear.

5.8 Measurement of Q_T

The total quality factor, Q_T, of a loudspeaker (or system of loudspeakers) mounted in an enclosure may be determined as outlined in Section 3.6. Expressions are given which are relevant to both the constant current condition (high signal source impedance) and the constant voltage condition (low signal source impedance). The latter condition is more relevant to modern, solid-state amplifiers.

5.9 Dynamic mass

The dynamic mass of a loudspeaker M_d is comprised of the mass of the air load M'_{MR} and that of the moving parts of the loudspeaker M_{MR}. To determine the dynamic mass of a loudspeaker it is first necessary to find out its resonance frequency, f_0, then the resonance frequency, f_m, which is observed when a mass m of a few grams is added to the speaker cone. The dynamic mass is then given by

$$M_d = M_{MC} + M'_{MR} = \frac{mf^2_m}{f^2_0 - f^2_m} \qquad (5.1)$$

(see Section 3.3.1).

5.10 Mass of the speech coil and cone

The mass of the moving system of a loudspeaker can be obtained simply by weighing those parts, but where one is confronted with an assembled loudspeaker the weight of the moving parts can be calculated from eq. (5.1): $M_d = M_{MC} + M'_{MR}$. The mass of the air load, M'_{MR}, is given by eq. (3.14) as $2 \times 1{,}58r^3$ kg, where r is the effective cone radius in metres. The mass of the moving parts, M_{MC}, is then derived by subtracting M'_{MR} from M_d.

5.11 Mechanical compliance of suspension

If the resonance frequency and the dynamic mass are known, since, from eqs (3.11) and (5.1),

$$f_0 = \frac{1}{2\pi \sqrt{(M_d C_M)}}$$

then,

$$C_M = \frac{1}{\omega_0^2 M_d} \text{ m/N.} \qquad (5.2)$$

5.12 Transducing constant Bl

The force F exerted when a current i flows through the speech coil is given by the expression

$$F = Bli \text{ newtons.}$$

If a loudspeaker is mounted cone upwards, and a known weight m is added to the cone, the cone will be displaced downwards. The direct current, i', which restores the cone to its original position enables the transducing constant to be evaluated from the following expression:

$$Bl = \frac{9{,}8m}{i'}. \qquad (5.3)$$

The added mass is m kg, the factor of 9,8 being needed to convert this to newtons.

5.13 Mechanical resistance of the suspension

The mechanical radiation resistance of the suspension can be measured by making measurements on an unmounted loudspeaker. From eq. (3.38) we know that

$$R_A = \frac{\omega_0' M_A}{Q_{T1}}$$

from which, for an unmounted loudspeaker we get

$$R_{MS} + R_{MR} = \frac{\omega_0(M_{MC} + 3{,}15\, r^3)}{Q_{T1}} \qquad (5.4)$$

where R_{MR} is the mechanical radiation resistance of an unmounted loudspeaker and is about $8{,}45 \times 10^{-6} + r^6 f^4$ m.k.s. mechanical ohms and can thus be ignored.

6 Digital testing techniques

It is hardly surprising that the measurement of loudspeaker performance is yet another application where the computer offers advantages of speed and reliability over traditional methods. Loudspeaker or speaker system characteristics can be derived from a mathematical analysis of the system response to a single pulse input. Further, the loudspeaker response sampling time can be controlled to ensure that reflected sound is excluded from the analysis, and averaging techniques can be used to offset random background noise. The isolated anechoic room is no longer as essential in the measurement of loudspeaker performance.

6.1 Principles of operation

A loudspeaker (or a complete system) can be completely characterized by a complex transfer function $T = Sy/Sx$. Sx and Sy are the Fourier transforms of the input and output as illustrated in Fig. 6.1.

Figure 6.2 is a simplified illustration of the analysis process. A fast sampling technique is used to acquire and store the loudspeaker output signal. The result of an analogue-to-digital conversion of each sample is stored in memory and, after sampling, the Fourier transforms of both input and output signals are evaluated to obtain the transfer function, T. From this various loudspeaker characteristics can be evaluated and displayed or plotted on suitable peripheral equipment, namely

- the sound pressure level (SPL) which corresponds with the modulus $|T|$,
- the phase response (φ) which corresponds with the phase operator of T,
- the time delay which corresponds to $d\varphi/d\omega$.

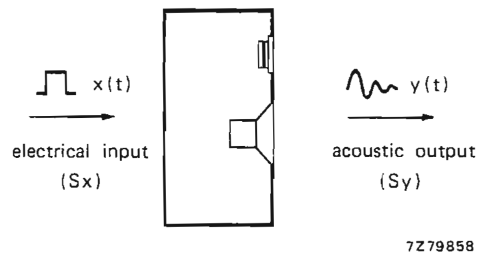


Fig. 6.1. A loudspeaker – or loudspeaker system – can be characterized by a complex transfer function. $T = Sy/Sx$.

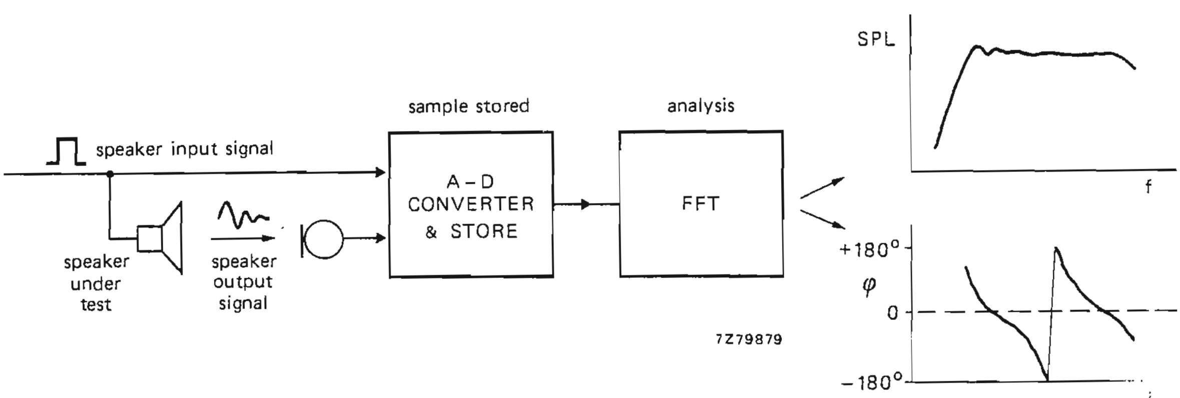


Fig. 6.2. Loudspeaker (or system) characteristics can be produced by the analysis of the response to a pulse input. The speaker response is stored as a sequence of digital numbers and the stored data analysed using a fast Fourier transform.

6.2 Input signal

The most convenient input signal for the purposes of analysis is a square pulse as this comprises a distribution of sinusoidal signals with a frequency/amplitude relationship conforming to the function $|\sin x/x|$. Appropriate choice of the pulse width thus ensures a fairly flat frequency/amplitude characteristic for the audio-frequency part of the input signal. Referring to Fig. 6.3, a pulse width of 10 µs ($1/t$ = 100 kHz) ensures a fairly flat response between 0 Hz and 20 kHz (P and Q in Fig. 6.3(b)) so that the output requires little or no correction for non-linearity in the input signal. Even better linearity is obtained with a shorter pulse but the pulse width and amplitude must be large enough to impart sufficient energy to the loudspeaker for the output to be useful for analysis.

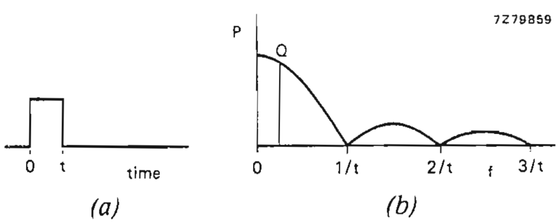


Fig. 6.3. The input signal is a rectangular pulse (a), this having sinusoidal components which correspond in magnitude to the function $|\sin x/x|$ (b) in the frequency domain. A 10 µs pulse ($1/t = 10^5$ Hz) gives a reasonably flat audio-frequency (0 to 20 kHz) distribution.

6.3 Sampling

The SPL generated by the loudspeaker is sensed by a suitable microphone and applied to an analogue-to-digital converter which performs conversions at regular time intervals Δt. The sampled data is stored. The faster the sampling process (i.e. the smaller Δt in Fig. 6.4(a)) the more accurately the loudspeaker output is characterized by the stored data.

Constraints inevitably exist which limit the sampling rate, but it can be shown mathematically that slightly more than two samples per cycle are sufficient to fully describe a sinewave. In other words the minimum sampling frequency is slightly more than twice the maximum frequency (usually 20 kHz) that can be plotted.

Sampling time – the period over which the waveform is observed – is also subject to contradictory requirements. Obviously a long sampling period is essential to detect and process a low-frequency signal component. However, practical considerations limit the sampling 'window' to a reasonable time, T, and, sampling has to cease before any reflected sound arrives at the microphone so that the sound sampled is only direct sound from the loudspeaker. Further to this, the window dimension, T, dictates the spacing, Δf, of the processed output data in the frequency domain – see Fig. 6.4(b).

6.4 Eliminating random noise

When making measurements, noise problems can arise if the input signal has insufficient energy. In this case repetitive measurements can be made, the results being cumulatively summed in store. Random sound data tend to average out while significant data are added. The result is a very close approximation to the pure loudspeaker response but it is important to dimension the time window correctly to ensure that the sound being received does not include any reflected sound.

6.5 Computation of the transfer function

Using an FFT algorithm, both amplitude (normal anechoic sound pressure) and phase can be computed and plotted on suitable terminal equipment. The time delay ($d\varphi/d\omega$) can also be derived fairly simply from the same process thus presenting a comprehensive set of characteristics very rapidly.

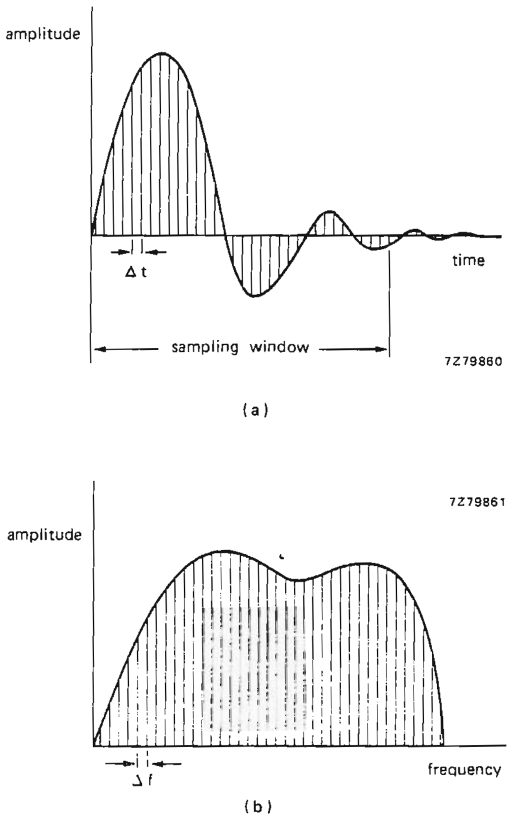


Fig. 6.4. The loudspeaker output (a) is sampled at Δt intervals within a time window T. The frequency response (b) obtained from the analysis of the loudspeaker output is constructed from data spaced in the frequency domain by Δf. The frequency response shown is plotted linearly; it could also be plotted logarithmically. Sampling frequency limits the highest detectable frequency ($f_{max} < 0.5\ \Delta t$); T limits the lowest detectable frequency and determines the interval Δf.

Besides the benefit of speed it should be noted that when a suitable sampling window is used this technique produces a genuine free field characteristic. This is not often the case with the anechoic room where standing waves are usually present at low frequencies.

A further consideration in favour of the digital technique is that the data is easily stored on whatever computer storage medium is convenient, and can be extracted and replotted for comparison, further analysis and so forth, without recourse to the test set-up.

6.6 Decay response

Digital techniques lend themselves to an analysis of the decaying waveform and hence of loudspeaker resonances and enclosure reflections which are otherwise difficult or impossible to obtain. Sound is a pressure variation functioning in frequency and time. A three dimensional model, frequency-time-SPL, can be plotted using digital signal processing techniques.

The speaker is energized by a signal comprising the complete audio spectrum with a reasonably flat amplitude/frequency characteristic and the normal (axial) SPL is observed. If at time t the signal applied to the speaker is reduced instantaneously to zero, the speaker output decays in a complex manner, the decay being characterized by resonances set up within the electrical/mechanical/acoustical system.

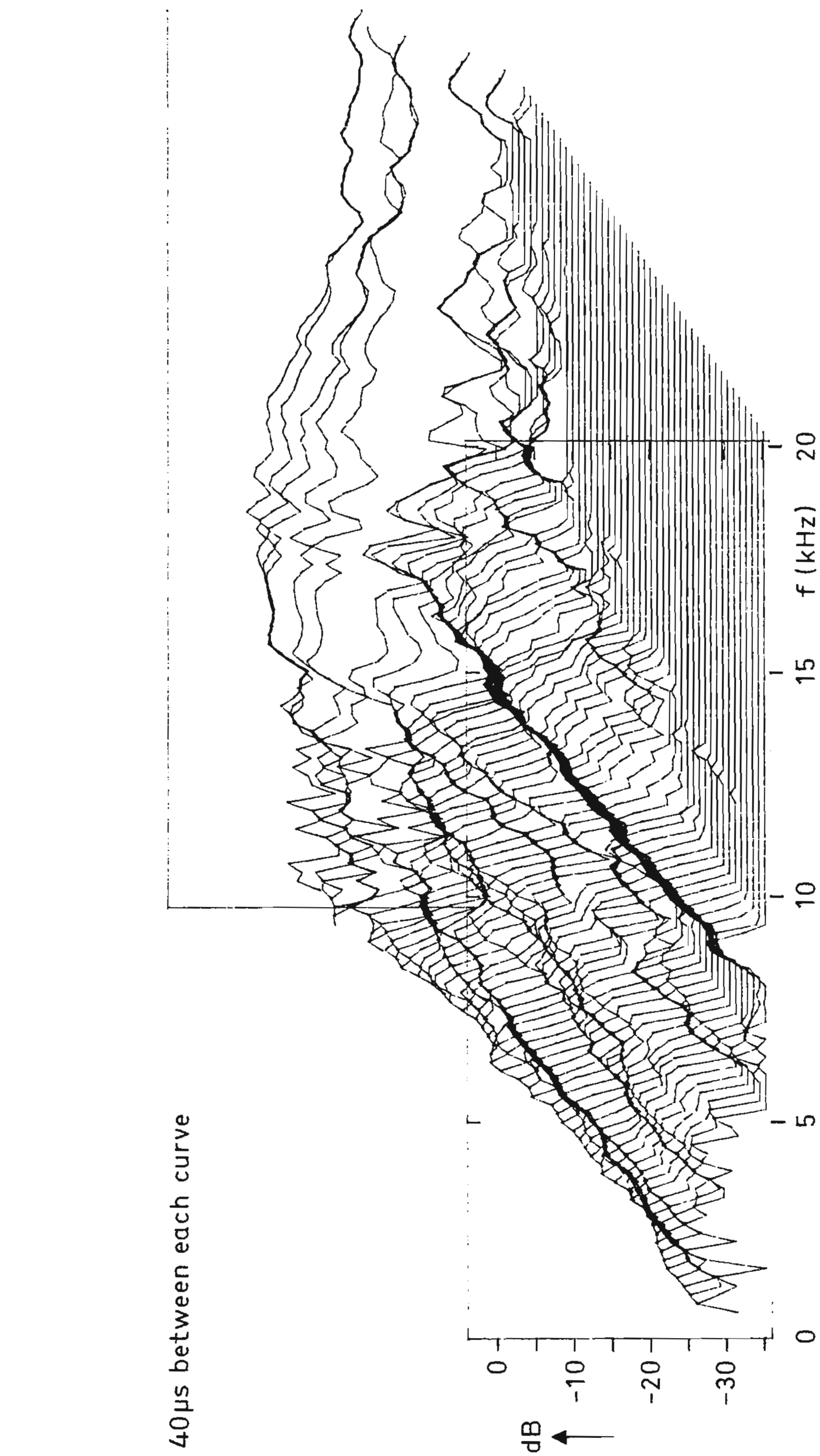
Using the signal sampling and analysing techniques already discussed a 3-dimensional 'landscape' of the acoustic pressure at the microphone can be constructed. Such characteristics are shown in Fig. 6.5 for a full-range speaker (a) and a dome tweeter (b).

Samples taken at regular intervals provide for complete analysis resulting in the 'landscape' characteristic, showing time contours of the decaying SPL across the complete frequency spectrum. Lines behind the contours are omitted as these would confuse the print-out.

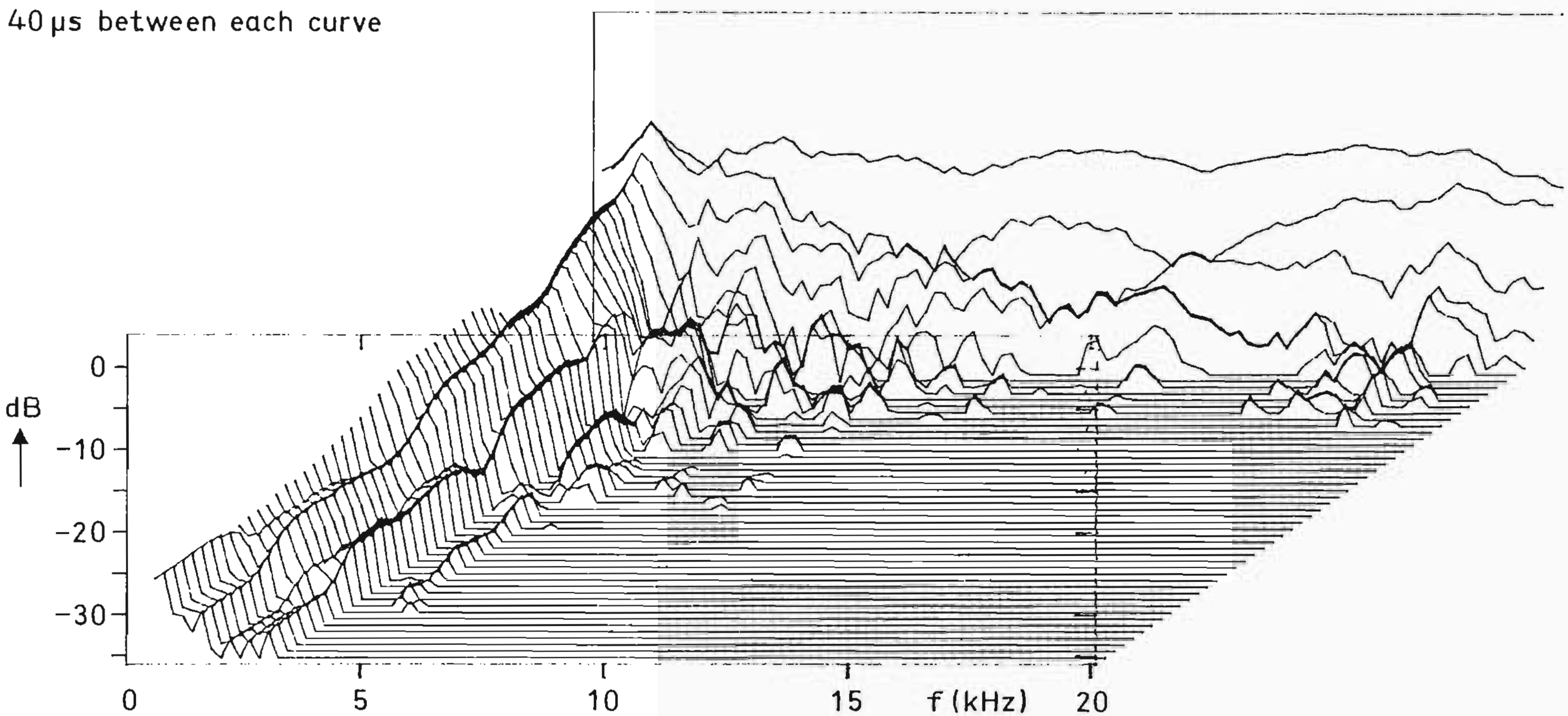
Resonances usually masked in normal sound curves, show up in the landscape as ridges of constant frequency, (parallel to the time axis) and reflections show up as ridges at regular time intervals parallel to the frequency axis. The landscape thus formed tells us a lot more about the loudspeaker (or system) character than the traditional methods of measurement and corresponds very well to the way in which our ears perceive sound.

Ideally the loudspeaker should exhibit a flat steady-state SPL with a fast fall off, uniform across the frequency spectrum, after switch off. Neither type of ridge should be present in the ideal case.

In constructing these characteristics signal processing techniques have been used to remove irrelevant data, thus presenting a clearer picture of the free-space loudspeaker characteristic.



(a)



(b)

Fig. 6.5. The decay characteristics of a full-range loudspeaker (a) and a dome tweeter (b). The peaks along the frequency axis appear as ridges parallel to the time axis and represent resonant frequencies. Ridges parallel to the frequency axis indicate reflections (against the enclosure walls).

7 Direct and reflected sound

Listening to sound from a point source in a perfectly quiet environment, the sound pressure level has a component which is due to direct radiation from the source and a component which is the sound reflected from the environment.

7.1 Direct sound

Consider a point source of acoustic power, W, radiating evenly in all directions under free space (zero reflection) conditions (Fig. 7.1). The energy density, E, at a distance r from the source can be derived from the expression

$$W\mathrm{d}t = E4\pi r^2 c\mathrm{d}t$$

where c is the velocity of sound (344 m/s).

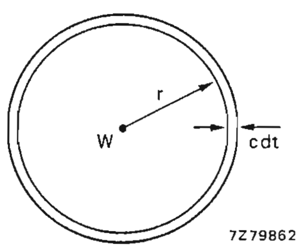


Fig. 7.1. Energy density at distance r from a point source = $W/4\pi r^2 c$.

Rewriting the expression, we get

$$E = \frac{W}{4\pi r^2 c}. \tag{7.1}$$

We can also define energy density as

$$E = \frac{p^2}{\varrho c^2} \tag{7.2}$$

where p is the alternating (sound) pressure in N/m^2 (or Pa) and ϱ is the density (normally about 1,18 kg/m^3).
From Section 1, eq. (1.15) we observe that the intensity of sound is given by the expression

$$I = \frac{p^2}{\varrho c}.$$

Thus, from eqs (7.1) and (7.2) we can write

$$I = \frac{W}{4\pi r^2}$$

or

$$I = \frac{WQ}{4\pi r^2} \tag{7.3}$$

where Q is the directivity factor of a directional source: that is, the ratio of the intensity of radiation in the considered direction from the directional source to that which would be observed at the same point if the source was of the same power but omnidirectional.

7.2 Reflected sound

Taking the same source considered in Section 7.1 and placing it in a natural environment will cause reflected sound to be propagated in the space surrounding the source. With the source radiating the same acoustic power, W, the energy density at any point will be the sum of that contributed by the direct sound ($W/4\pi r^2 c$) and that due to reflected sound.

If the source is switched off the sound energy density decays exponentially, i.e.

$$E = E_0 e^{-\alpha t}$$

where E_0 is the energy density at the instant of switch off.
The reverberation time, T, is the time it takes the sound to fall by 60 dB after the source has been switched off. Thus $e^{-\alpha T} = 10^{-6}$, and the reverberation coefficient, α, is $6 \ln 10/T$. The reverberation time for a particular room will vary with frequency so that for a particular sound it will depend on the frequency content of the sound.

The rate of energy absorption after switch-off is given by

$$\frac{\mathrm{d}E}{\mathrm{d}t} = -\alpha E_0 e^{-\alpha t}$$

and thus, at the moment of switch off ($t = 0$) the rate of absorption is αE_0. This corresponds to the rate of absorption while the source of the sound is radiating and thus,

$$W\,dt = \alpha E_0 V\,dt$$

where V is the volume of the room. Manipulating this expression and substituting $6 \ln 10/T$ for α, we get an expression for the energy density of reflected sound due to a sound source of known output W in a hall or room of known volume V and reverberation time T:

$$E = 0{,}072\,\frac{WT}{V}. \qquad (7.4)$$

Other expressions applicable to the reflected sound are as follows:

Average free path length

$$d = \frac{4V}{S}. \qquad (7.5)$$

(S = total surface area of the walls of the room).

Absorption

$$\begin{aligned} A &= S\{-\ln(1-\bar{\alpha})\} \\ &= S\bar{\alpha} \quad \text{where } \bar{\alpha} \ll 1 \end{aligned} \qquad (7.6)$$

($\bar{\alpha}$ is the average absorption coefficient of the room).

Reverberation time

$$T = 0{,}16\,\frac{V}{A} \qquad (7.7)$$

or

$$T = 0{,}16\,\frac{V}{A + 4Vm}$$

where V is the volume of the room in cubic metres, A is the absorption square metres, and m is the absorption of the air.

Room constant

$$R = \frac{S\bar{\alpha}}{1-\bar{\alpha}} \qquad (7.8)$$

If $\bar{\alpha} \ll 1$

$$R = A$$

Reflected energy intensity

$$I = \frac{p^2}{4\varrho c} = \frac{W}{R} \qquad (7.9)$$

7.3 Direct radiation field

The listener in a hall or room will receive both direct sound from a loudspeaker and reflected sound. The ratio of direct to reflected sound will depend on the placement of the loudspeaker and the listener's position relative to it. Close to the loudspeaker direct sound will swamp the reflected sound. Further away the ratio will diminish. The locus of points at which the direct and the reflected sound are equal defines boundary within which the direct sound is dominant and outside which the reflected sound is dominant.

The boundary for an *omnidirectional point source* of sound can be calculated by equating eqs (7.1) and (7.4). The radius r_0 of the area in which direct sound is dominant is thus given by

$$r_0 = 0{,}057\sqrt{\frac{V}{T}}. \qquad (7.10)$$

For a practical loudspeaker system a directivity factor must also be taken into account.

7.4 Total sound level

The total alternating sound pressure is the sum of the direct and the reflected sound. For a point at a distance r from the source, the sound pressure due the direct radiation can be obtained from eq. (7.3),

$$I = \frac{WQ}{4\pi r^2} = \frac{p^2}{\varrho c}.$$

Thus

$$p^2 = \frac{WQ\varrho c}{4\pi r^2}.$$

The sound pressure due to the reflected sound is derived from eq. (7.9),

$$I = \frac{p^2}{4\varrho c} = \frac{W}{R}.$$

Thus the total sound pressure (direct + reflected) is given by

$$p^2 = W\varrho c\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right). \qquad (7.11)$$

Equation (7.11) can be manipulated to obtain

$$\frac{p^2}{p^2_{\text{ref}}} = \frac{W}{W_{\text{ref}}}\,\frac{W_{\text{ref}}\,\varrho c}{p^2_{\text{ref}}}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right).$$

The sound pressure level (SPL) and power level (PWL), as defined in Section 1, are thus

$$\text{SPL} = 10\log_{10}\frac{W}{W_{\text{ref}}} + 10\log_{10}\frac{W_{\text{ref}}\,\varrho c}{p^2_{\text{ref}}} + 10\log_{10}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right).$$

$$\text{SPL} \approx \text{PWL} + 10\log_{10}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right). \qquad (7.12)$$

7.5 Energy measurement in a reverberant room

A reverberant room used for loudspeaker measurements is set up so that the indirect sound is greater than the direct sound, i.e.

$$\frac{4}{R} \gg \frac{Q}{4\pi r^2}.$$

As the distance r from the loudspeaker to the microphone is always more than 1 metre, and

$$R \approx A$$

eq. (7.12) can be rewritten,

$$\text{PWL} = \text{SPL} - 10\log\frac{4}{A},$$

or using eq. (7.7),

$$\text{PWL} = \text{SPL} - 10\log\frac{25T}{V}.$$

Thus, for example, if $V = 200\ \text{m}^3$,

$$\text{PWL} = \text{SPL} - 10\log\frac{T}{8}.$$

The last term is a correction factor enabling conversion between a sound pressure curve and the equivalent energy curve.

8 Listening room acoustics

The listening room will normally be the living room and, apart from the placement of carpets, hangings and furniture, there is little the listener can do to adjust its acoustics.

8.1 Resonances in small listening rooms

In a listening room sound can travel not only back and forth between opposite walls, it can also travel around the room by reflection from adjacent walls. At certain angles of reflection standing waves may be set up. As there are two pairs of "walls", plus the ceiling and floor, there are three groups of modes of oscillation.

The resonance frequencies of any rectangular enclosure are given by the Rayleigh equation:

$$f_n = \frac{c}{2}\left\{\left(\frac{n_L}{L}\right)^2 + \left(\frac{n_W}{W}\right)^2 + \left(\frac{n_H}{H}\right)^2\right\}^{\frac{1}{2}} \qquad (8.1)$$

where f_n = nth resonance frequency (Hz)
c = velocity of sound (344 m/s)
L = room length (m)
W = room width (m)
H = room height (m)
and n_L, n_W and n_H are the integers 0, 1, 2, 3 ....

The lowest resonance frequency produced, f_c, is dependent on the length of the room and is defined by the equation

$$f_c = \frac{c}{2L}. \qquad (8.2)$$

Putting $n_L = 1$, $n_W = 0$ and $n_H = 0$ in the Rayleigh equation gives the same result as eq. (8.2). A series of harmonics are also produced, and these are found by giving the term n_L the values 2, 3 ....

Table 8.1 gives the resonance frequencies of the first 24 modes of oscillation for three different rectangular listening rooms, the sizes of which are shown in Fig. 8.1. If the source of sound is located in the corner of one of these rooms, it will be possible to excite every mode of oscillation.

These standing waves greatly modify listening conditions, particularly at low frequencies. Their amplitudes depend on the absorption.

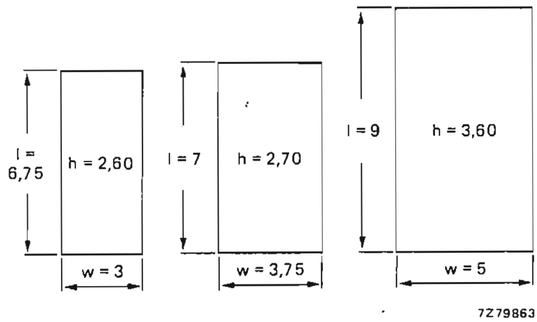


Fig. 8.1. Sizes of three listening rooms used for comparison.

Table 8.1 The first 24 modal frequencies in three typical listening rooms

Room = 6,75 × 3 × 2,6 m		Room = 7 × 3,75 × 2,7 m		Room = 9 × 5 × 3,6 m	
frequency (Hz)	mode	frequency (Hz)	mode	frequency (Hz)	mode
25,19	1.0.0	24,29	1.0.0	18,90	1.0.0
50.40	2.0.0	45,33	0.1.0	34,00	0.1.0
56,67	0.1.0	48,57	2.0.0	37,80	2.0.0
62,00	1.1.0	51,43	1.1.0	38,90	1.1.0
65,39	0.0.1	63,00	0.0.1	47,22	0.0.1
70,10	1.0.1	66,44	2.1.0	50,83	2.1.0
75,56	3.0.0	67,48	1.0.1	50,86	1.0.1
75,80	2.1.0	72,86	3.0.0	55,70	3.0.0
82,54	2.0.1	77,59	0.1.1	58,20	0.1.1
86,52	0.1.1	79,52	2.0.1	60,50	2.0.1
90,10	1.1.1	81,29	1.1.1	61,20	1.1.1
94,44	3.1.0	85,80	3.1.0	66,10	3.1.0
99,92	3.0.1	90,67	0.2.0	68,00	0.2.0
113,3	0.2.0	93,86	1.2.0	70,60	1.2.0
114,9	3.1.1	96,30	3.0.1	73,76	3.0.1
116,1	1.2.0	102,9	2.2.0	77,79	2.2.0
124,0	2.2.0	106,4	3.1.1	81,22	3.1.1
130,7	0.0.2	110,4	0.2.1	82,79	0.2.1
130,8	0.2.1	116,3	3.2.0	88,52	3.2.0
132,2	1.0.2	120,6	2.2.1	91,00	2.2.1
136,2	3.2.0	125,9	0.0.2	94,44	0.0.2
140,1	2.0.2	128,3	1.0.2	96,32	1.2.2
140,2	2.2.1	132,3	3.2.1	100,3	3.2.1
142,5	0.1.2	133,8	0.1.2	100,4	0.1.2

Each mode of oscillation has a different distribution of sound pressures, and its own damping constant depends on the absorption in the room and the room volume. Also, when the sound is cut off, the sound pressure in each mode decays exponentially at a rate depending on its own damping constant.

In a typical small room differences in sound pressure levels of as much as 25 dB have been measured. During the decay process, where two modes are close together, beats between the modal frequencies can occur. In a large room, the beat effect will be negligible since there is better diffusion, but in a small room, there are fewer frequencies below about 120 Hz (low limit for male speech) and the spectrum is discontinuous. Bass boom on speech occurs around this frequency and, because of the discontinuous spectrum, "large room sound" cannot be achieved in a small room.

The modal frequencies of the three rooms being considered can be drawn as 'spectra'; Fig. 8.2 shows how the large room has the smoothest distribution of its natural resonances, whilst the small room will be boomy on male speech. The first 50 frequencies are given.

In the case of a very large room, the first resonance frequencies are infrasonic. That they have no effect is important, but what is more significant is that the amplitude of the higher-order modes which can be heard are obviously much reduced. Naturalness can be achieved only in large rooms; coloration is unavoidable in small rooms.

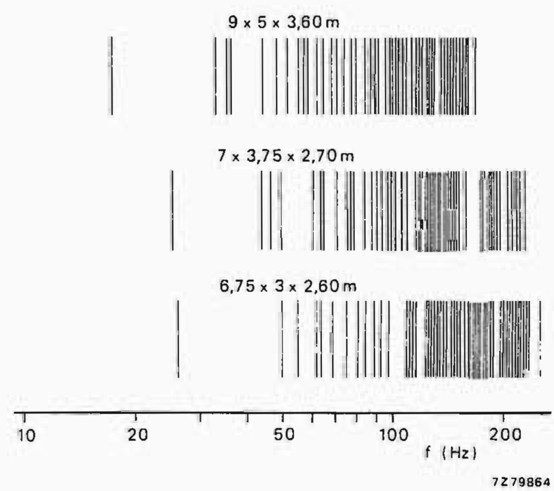


Fig. 8.2. The first 50 modal frequencies of the three listening rooms shown in Fig. 8.1.

8.2 Listening room properties

Since a spectrum in which the resonances are uniformly distributed is preferable to one in which they are grouped together in narrow bands separated by large gaps, it follows that the worst shape for a room would be a cube. This is because, since all dimensions are the same, resonances will occur at the same frequencies. Fortunately, cubical rooms are seldom found.

A distribution of resonances of minimum 10 per 1/3 octave provides a good room characteristic. It can be calculated that the best distribution of resonance for a small room is obtained with the dimensions of the length, width and height in the ratio of 1,6 : 1,25 : 1; for an average room 2,5 : 1,6 : 1, and for a large room 3,2 : 1,25 : 1.

The three room sizes indicated in Fig. 8.1 may be compared. The smallest room has the proportions 2,6 : 1,2 : 1, whilst the other two have proportions of roughly 2,5 : 1,4 : 1. As we have seen, both the larger rooms will have a more uniformly distributed spectrum and there will be less coloration of the sound.

A standing wave at mains frequency is to be avoided as, otherwise, hum present in the loudspeaker output will be considerably magnified. A room with one or more dimensions of 3,44 metres is thus not ideal where the mains frequency is 50 Hz.

8.3 Reverberation and absorption

The reverberation time of a listening room is defined as the time taken for a sound to die away to one-millionth of its original intensity. Such a value represents a decay from a comfortable listening level to the threshold of audibility. Somewhere between the deadness of a short reverberation time and the uncertainty of a long reverberation time lies an optimum value for listening.

Ideal acoustic conditions are those that lend a degree of 'liveness' to the sound without detracting from the clarity, and here we find that the criteria vary with application. If slow organ music is being played, acoustic properties which reinforce the musical characteristic by allowing the pipes to 'resound' through the listening area may be seen as desirable but the same characteristics may completely mar the articulation of rapid speech. If the echoes of one syllable have not been absorbed before the next is enunciated the resulting sound is indistinct and hard for the listener to understand.

The optimum reverberation time thus depends on the 'programme' being delivered and, in fact, on room size and frequency. Where intelligibility of speech is the only consideration, there is no limit to the absorption which can be tolerated if sufficient acoustic power is available from the source.

In considering the effects of room acoustics, it should be remembered that the recording engineer and, in the broadcasting context, the control room engineer, have at their disposal, control of reverberation. Artificial reverberation can be used to good effect and, although the reverberation time of a music studio is nearer that of a concert hall than a small living room, skill on the part of the recording engineer results in surprisingly good reproduction of music in listening rooms whose dimensions would lead one to expect the contrary. The same recording, played in very large rooms, with longer reverberation times will not necessarily sound natural. A reverberation time of 0,5 seconds at 1000 Hz in a room with a volume of about 100 m^3 has been suggested, falling further as the volume is reduced. Many broadcasting authorities and recording companies adopt a reverberation time of 0,4 seconds for the room in which the monitor loudspeakers are heard. This is considered to simulate average conditions in the majority of listening rooms in which the programme material will be reproduced.

Absorption at low frequencies is considered to be due to mechanical resonances in the building structure, where relatively large masses are moved by the resonance effect (e.g. windows, panels, etc.). To balance the low frequency absorption, heavy curtains in generous folds will provide mid and high-frequency absorption. A balance must be obtained. A building of modern construction is less likely to suffer from structural resonances, and if the use of curtain material is overdone there will be a distinct lack of mid-range frequencies in the reproduced sound. A luxuriously furnished large modern appartment need not necessarily provide the best listening conditions. Loss of mid-range frequencies can also be the result of too great an area of fitted carpet, whereas the use of separate rugs might be acoustically preferable. Loss of mid-range frequencies can be partially compensated by using the presence control on the amplifier.

8.4 Source positioning and checking

A corner position for a loudspeaker in the home has three advantages. Firstly, it is the position of least domestic inconvenience; secondly, the maximum number of room resonances can be excited and a smoother distribution obtained; thirdly, and probably of greatest significance, is the enhancement in bass response.

When a speaker is mounted close to a wall the effective radiating area is theoretically doubled at low frequencies, owing to the reflection of the wall. A room corner has three mutually perpendicular surfaces and thus the effective radiating area for a corner speaker is doubled three times. A gain of 8 would, however, only be possible if the surface had a reflectance of 100%. Allowing for the absorption of the walls, together with that of fitted carpet, adjacent curtains, etc., which reduce the effect, a reflectance of 50% would be more likely, thus leading to a gain of 3 to 6 dB (2 to 4 times).

Another effect of placing a loudspeaker close to a wall or in a corner is to increase radiation impedance, particularly at low frequencies, which contributes to better efficiency of low note reproduction.

In a small room the comparative isolation of low resonance frequencies may cause trouble; powerful peaks occur at these frequencies which may be objectionable. If this is the case, an alternative position for the loudspeaker must be found and the positions near to the centre of the long wall of the room should be tried; their best positions can only be found by experiment.

The correct phase connection is important, of course, for stereo reproduction. The correct connections are usually indicated clearly by the manufacturer. Where this is not clear, if the two speakers or speaker systems are placed side-by-side and fed with a common signal, they will exhibit attenuated low-note reproduction due to acoustic short-circuiting if they are connected in antiphase. Reversing one of the loudspeaker connections will connect them in phase and dramatically improve the sound.

8.5 Power requirements in a living room

The power requirements depend on the maximum loudness level required in the listening room. Factors determining the maximum loudness level include:
- the dynamic range of reproduction required,
- the background noise level,
- the volume of the listening room,
- the reverberation time of the listening room.

8.5.1 DYNAMIC RANGE

A dynamic range of about 70 dB could be expected from a large orchestra performing in a concert hall, but the dynamic range of recorded music is less than this. High quality tape recordings played on professional machines have a dynamic range of about 60 dB at the most, and domestic tape recorders provide considerably less. The same applies to disc recordings, where a dynamic range of 55 to 60 dB is possible. Both AM and FM radio transmissions have amplitude limiting and volume compression applied, so it is unlikely that the listener will experience a dynamic range greater than 60 dB. Modern digital techniques (Philips Compact Disc, for example) may provide substantial improvements in the future.

8.5.2 BACKGROUND NOISE LEVEL

The background noise level depends upon the location of the listening room, country surroundings obviously being quieter than tov
In addition to noise arriving from sources outside the room, the movement of persons, shuffling of feet, breathing and so on, contribute to the noise level. Table 8.2 gives typical noise levels for various backgrounds.

Table 8.2 Loudness levels of various backgrounds

noise source	phons
reference threshold at 1000 Hz	0
threshold of people with very good hearing	5
quiet house, country, midnight	20
quiet house, country, early evening	25
recording studio	30
quiet living room	30*
average living room	40
quiet conversation	50

* The lower the background noise, the more enjoyable will be the reproduced music, because the full dynamic range of 60 dB can be obtained without exceeding the preferred maximum listening level of 90 dB.

8.5.3 CALCULATION OF NECESSARY ACOUSTIC POWER

The acoustic power necessary to provide an SPL of 2×10^{-5} N/m² (Pa) - the hearing threshold at 1 kHz - can be calculated from equations given in Section 7. From eq. (7.9) we obtain,

$$\frac{p^2}{4\varrho c} = \frac{W}{R}$$

and as

$$p = 2 \times 10^{-5} \text{ N/m}^2 \text{ (Pa)}$$
$$\varrho \approx 1{,}18 \text{ kg/m}^3$$
$$c = 344 \text{ m/s}$$

and generally

$$R \approx A,$$

then we can substitute eq. (7.7) to get

$$W_{\text{threshold}} = 4 \times 10^{-14} \frac{V}{T} \text{ watts.} \qquad (8.3)$$

Equation (8.3) establishes the threshold level from which the final power is to be derived. First let us consider the reproduction of music in the listening room. If there is no background noise and a natural pause in the music occurs, the only audible sound will be electrical noise from the reproduction system. If the system has a satisfactory signal-to-noise ratio, there should be no objectionable noise, but inferior equipment may cause trouble at high volume settings. Needle scratch or tape hiss may also be present. Add to this the background noise of the room, which will usually mask the electrical noise, especially if good equipment is used. The programme must be satisfactorily reproduced above this total level (estimated at 30 dB). The maximum acoustic power level required, therefore, will be

$$W'_{\text{max}} = W'_{\text{threshold}} + W_{\text{background}} + W'_{\text{programme}}$$

where $W_{\text{background}}$ is the maximum background noise power and $W_{\text{programme}}$ is the power required to cover the full dynamic range of the reproduced programme, as shown in Fig. 8.3. If the background level is 30 phons and the dynamic range of the programme is 60 dB, the peak level will be a total of 90 dB above the threshold given by eq. (8.3). The maximum acoustic power required can be obtained from the general expression

$$W'_{\text{max}} = 4 \frac{V}{T} \times 10^{n-14} \text{ acoustic watts} \qquad (8.5)$$

where n is the peak sound level in bels above threshold, equal to background level plus dynamic range.

Consider the listening room 7 m long, 3,75 m wide and 2,7 m high, previously discussed. Its volume is 70 m³ and, if the room is comfortably furnished with a preponderance of soft furnishings, a reverberation time of 0,4 second can be assumed. The acoustic power required for reproducing a programme with a dynamic range of 60 dB over a background of 30 phons will be

$$W_{\text{max}} = \frac{4 \times 70 \times 10^{9-14}}{0{,}4} = 7 \times 10^{-3} \text{ acoustic watts.}$$

8.5.4 ESTIMATING THE POWER HANDLING CAPACITY

Having established the acoustic power required, we can now determine the loudspeaker power handling capacity. Taking the efficiency of the speaker system as 1%, the electrical power input will be

$$100 \times 7 \text{ mW} = 0{,}7 \text{ W}.$$

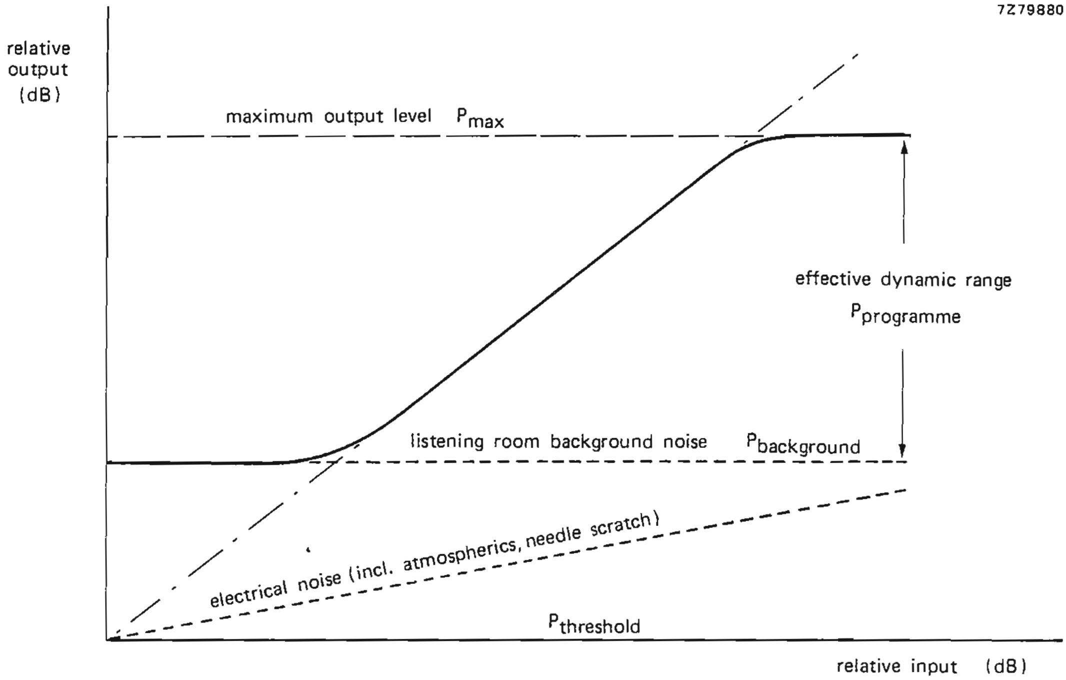


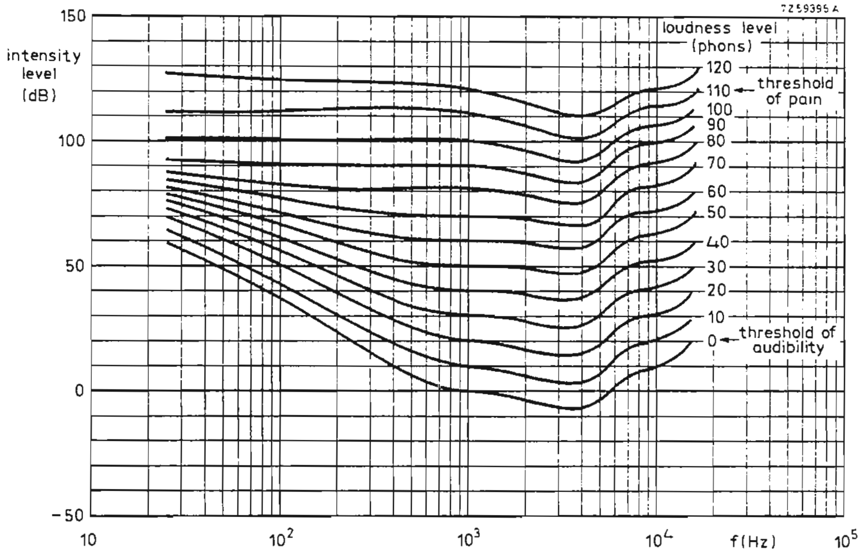
Fig. 8.3. Factors determining maximum acoustic power.

The following discussion is based on the desire to obtain the greatest realism in the reproduced sound and applies to only high fidelity installations. It is realized that there are very many applications where the specifications may be relaxed, but it is important to remember that the power handling capacity of the speaker must always be adequate, and preferably with a small margin, to accept the peak output power of the amplifier without distortion.

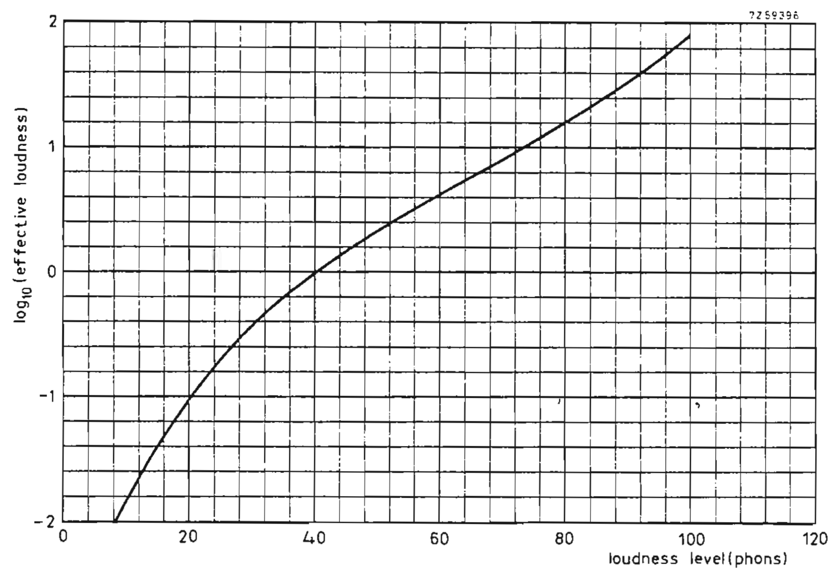
It can be seen from the Fletcher-Munson contours in Fig. 8.4(a) that the 90 phon curve is substantially flat over the bass region and so the relatively low electrical power of 0,7 W in the above example will be adequate to reproduce the signal at a level of 90 dB above the 1000 Hz threshold over the whole band.

However, at 40 Hz the threshold is nearly 60 dB above the threshold at 1000 Hz and at low volume levels over the bass region the sensitivity of the reproducing system should be increased, otherwise the signal may be inaudible. The physiological volume control (contour) has this property; the principles are illustrated in Fig. 8.4. It is a good solution, but it is more usual to employ bass boost using the normal tone control to overcome this bass deficiency. A bass boost of around 20 dB maximum is normally provided but, unless a contour control is employed, the increase in output can occur at all signal levels, not only those of low volume. To prevent distortion under bass boost conditions, the power handling capabilities of the system should be increased. In the example quoted, the peak power requirement becomes $0{,}7 \times 10^2 = 70$ W.

The ear is more tolerant of high volume levels where stereo reproduction is concerned. If the calculated power requirement of 70 W is applied to each channel, the power handling capacity of each speaker system and the power output of each of the reproducing amplifiers under bass boost conditions should therefore be 70 W peak and the acoustic power requirements will be

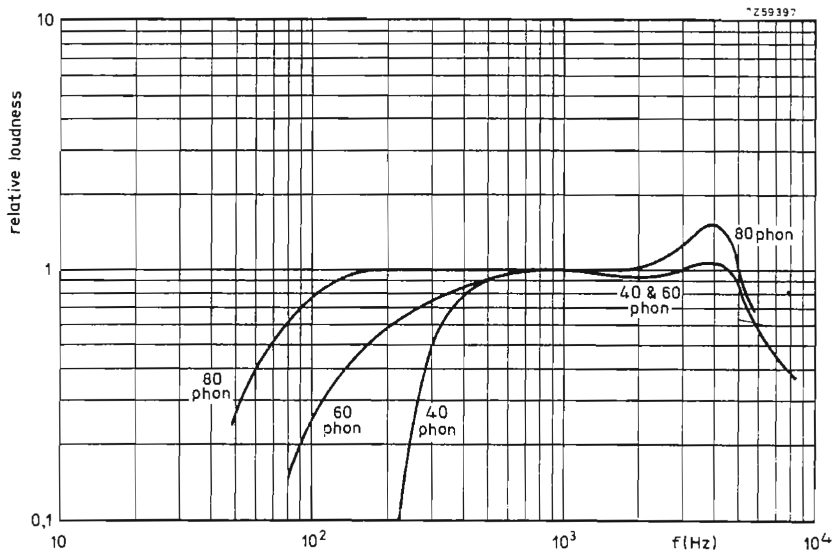


(a)

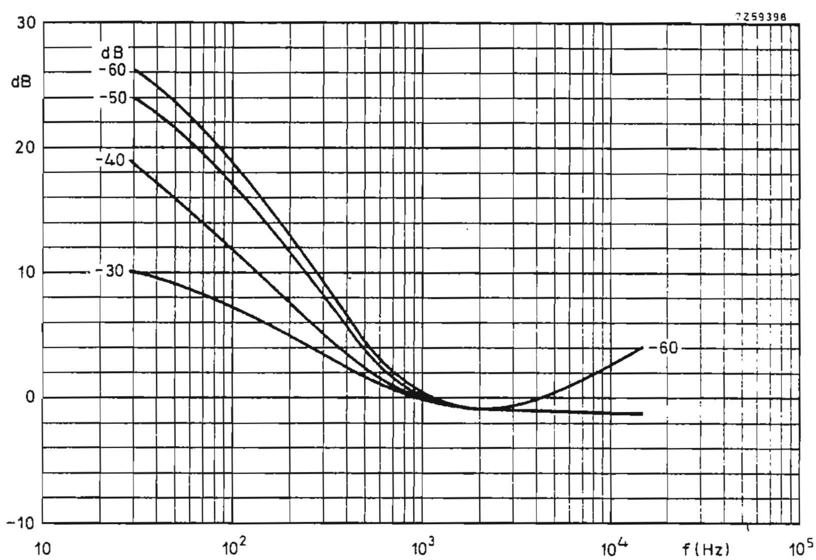


(b)

Fig. 8.4. Principle of a physiological volume (contour) control. From (a), the effective loudness is substantially logarithmic above about 40 phons and by taking the loudness at various frequencies for a given intensity and correcting for the modified logarithmic response of the ear, as shown in (b), the relative effective loudness as a function of frequency can be determined as shown in (c). Curve (c) clearly illustrates how a reduction in loudness causes loss of bass. The response of the contour control (d) counteracts this.



(c)



(d)

correctly satisfied for each channel individually. This does not mean that the full 140 W will ever be required at one time, since the total signal is divided between the two channels, but only that if all the signal is required on one channel there will be adequate output to meet the acoustic power requirements. Such is the ideal case, but in practice the power requirements can normally be reduced. By how much depends on the positioning of the speakers. For a corner mounted loudspeaker, a gain of from 2 to 4 times (3 dB to 6 dB) can be expected, as explained in Section 8.5. The power requirements of the system can thus be reduced by this amount from 70 W peak (music power) to, say, 35 W. In practice, the system would have a sinewave rating of 25 W r.m.s. and will be operating at a very much lower level than even this most of the time. Individual cases vary, however, and the speaker positioning and room absorption should be carefully considered before the calculated power requirements are modified.

The mono case is slightly different. An intensity level of 90 phons when reproduced by a mono system sounds very much harder and more irritating that when stereo is employed, or when it is produced by a live orchestra, since the directional discrimination of the ears is of no importance if the sound is emitted from one small source. This leads to a preference for reduced levels of loudness and smaller frequency range with mono reproduction. Something less than the calculated power requirement of 35 W peak will be required and in practice a 15 W installation would probably be considered satisfactory, if a corner speaker is used.

8.5.5 GENERAL RECOMMENDATIONS

From the foregoing it will be realized that there is no instant method of determining the electrical power requirements for a listening room. But a realistic estimate can be made. Figure 8.5 gives the recommended power handling capacities for each enclosure in a stereo installation, assuming corner mounting and a conversion efficiency of 1%.

It can be seen from Fig. 8.5 that a two-to-one variation in power is given. This is intended to cover the different furnishing conditions and absorptive properties of the listening room. In using this graph, special attention should be paid to the speaker efficiency.

The power given will be sufficient to reproduce a programme with a full dynamic range of 60 dB over a background of 30 dB with 20 dB bass boost. Individual cases can be calculated if required, but what the graph recommends should provide the listener with complete satisfaction in terms of listening quality, and enable him to get the very best performance from his equipment.

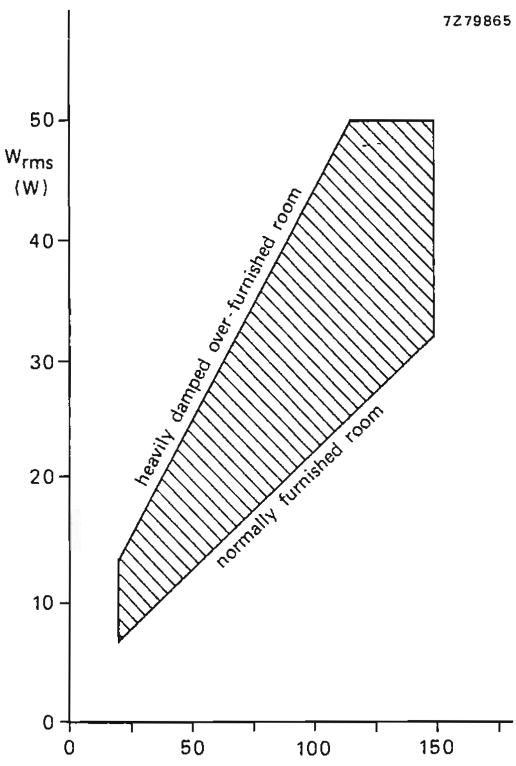


Fig. 8.5. Recommended power handling capacity (W_{rms}) for each corner-mounted enclosure of a stereo installation for different room volumes assuming 1% efficiency.

9 Hall acoustics

As with a small listening room, the acoustics of a large concert hall or theatre can have an appreciable effect on the output of a loudspeaker. A hall which exhibits all sorts of booming resonances and a long reverberation time is the nightmare of the public speaker, whilst the lifeless response of a hall with few resonant qualities may stifle a musical performance. Loudspeaker systems, and indeed complete public address systems can be built to favour a particular hall or a particular type of hall.

9.1 Hall characteristics

Consider a loudspeaker system placed to radiate sound into the auditorium of a hall and a microphone placed outside the field of direct radiation of the loudspeaker system, e.g. 'on stage' in the position an entertainer or speaker would normally occupy. If the loudspeaker system were fed with a pure sinewave varied smoothly in frequency through the audio spectrum, the sound picked up by the microphone would exhibit a very capricious characteristic compared with the loudspeaker output. The many signal paths of the reflected sound which the microphone picks up remain the same as frequency varies, but the relative phasing of the signals arriving at the microphone varies strongly with frequency as does the *en route* absorption of these signals.

The result of such a frequency plot will show a peak surround pressure level of about 14 dB above the average level. Although this figure is merely a typical observation and not absolute, it is a useful parameter for avoiding acoustic feedback. If the average sound pressure level at the microphone is 14 dB below the threshold for acoustic feedback this is generally sufficient to avoid acoustic feedback at any point during the performance.

9.2 Intelligibility

The point at which sound becomes unintelligible due to reverberation and other disturbing influences is when the coherent sound from a source is at a level, at the point of observation, which is less than the non-coherent sound. The coherent sound in this context is the direct sound from the source and the reverberant sound with a transit time of less than 50 ms. The non-coherent sound comprises reverberation due to the source which is delayed by more than 50 ms, plus general background noise received either directly or as reflections from other sources.

It should be noted that the reverberant sound is the sum of all the reflected sound from the source. The transit delay of 50 ms may well evoke the thought that at 50 ms the resultant sound pressure level will depend on path length and frequency of the reflected sound. It does; but what we are talking about here is the general sound level as a result of the many phase differences, cancellations and reinforcements that prevail. Although it is difficult to be precise about the point at which sound becomes unintelligible (because this partly depends on the type of sound being considered) reverberant sound which is delayed by more than 50 ms with respect to directly radiated sound is considered to have a destructive effect on the sound.

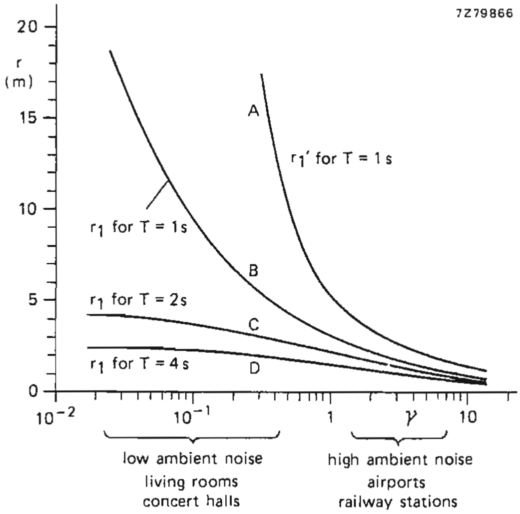


Fig. 9.1. Curves for discernability in a 4900 m³ hall. Curve A - coherent sound/incoherent sound = 0,5; curves B, C and D - coherent sound/incoherent sound = 1.

The interfering background noise will usually be uniformly distributed throughout the hall and the total background noise power can be given as a factor, γ, of the transmitted sound power from the source. Background noise obviously has an effect on intelligibility, as is borne out in Fig. 9.1, where the distance from the radiating source for intelligible sound, r_1, and that for unintelligible sound, r_1', are plotted against γ for various values of reverberation time, T, for a 4900 m^3 hall. Intelligible sound (at distance r_1) is where the ratio of coherent to non-coherent sound is 1. Unintelligible sound, in this context, is taken as sound where the coherent sound is at half the level of the non-coherent sound.

Increasing reverberation time has already been identified as reducing the intelligibility of sound and Fig. 9.1 bears this out. A good concert hall with a relatively long reverberation time (2 seconds for example) is thus most unsuited for use as a theatre, for which a reverberation time of 0,7 to 0,8 seconds is more appropriate.

9.3 Acoustic and electro-acoustic techniques

A number of general points are very important in connection with the acoustical properties of a hall. As they have already been discussed they are presented here as a summary of important considerations.

9.3.1 THE POSITION OF THE SOURCE OF THE SOUND

The source of the sound in a hall, whether it be an orchestra or the loudspeakers of a public address system, is best positioned so that the complete audience has an unimpeded view. Where the source is obscured from view it is likely that attenuation of the mid, and more especially, the high-range frequencies, will occur.

9.3.2 REFLECTED SOUND

Reverberation time should generally be kept fairly short in accordance with the sort of figures already discussed. Coherent sound should be maintained at a high level with general background sound and reverberant sound of greater than 50 ms stifled as much as possible. Positioning loudspeakers on walls or in corners generally improves coherence by increasing radiation impedance, and thus efficiency, and increasing directivity and gain in the direction of radiation.

9.3.3 HALL FURNISHINGS

If the seats in the auditorium are well upholstered to present a fair degree of acoustic damping, then the acoustics in the hall will not vary so much with the size of attendant audience.

9.4 Sound amplification

Where a public address system is used to amplify voice(s) and/or instruments in a hall, the microphone is, of course, placed well out of the field of direct radiation of the loudspeakers in order to avoid acoustic feedback. The microphone will pick up some sound indirectly. If the sound which is to be amplified has an energy density of E_o at the microphone, and the loudspeaker output due to this input is an acoustic power of W_a, then we can say,

$$W_a = \eta \frac{U^2}{R_L} = \eta \frac{1}{R_L} (\mu s \alpha \sqrt{E_o})^2 = \beta E_o \qquad (9.1)$$

where R_L = loudspeaker resistance
η = loudspeaker efficiency
μ = amplification
s = microphone sensitivity
α = conversion factor between alternating sound pressure p and the square root of energy density $\sqrt{E_o}$.

The indirect field of the loudspeaker is obtained from eq. (7.4):

$$E_i = 0{,}072\ W_a \cdot \frac{T}{V}$$

$$= 0{,}072\ \beta E_o \cdot \frac{T}{V}.$$

As the energy density at the microphone, E_o, must be 14 dB higher than that of the indirect field E_i at the microphone, we can say

$$\frac{E_i}{E_o} \leq 0{,}04$$

thus

$$0{,}072\ \beta \frac{T}{V} \leq 0{,}04$$

and

$$\beta \leq 0{,}556 \frac{V}{T}.$$

This becomes

$$\beta \leq 0{,}556 \frac{V}{T} \cdot \frac{Q_m}{Q_1^{*}} \qquad (9.2)$$

for a microphone and loudspeakers with directivities of Q_m and Q_1 respectively. The term Q_1^{*} is in fact a function of Q_1 which has to do with the absorption of the area of hall into which the loudspeaker is directed. This in turn is a function of the size of the audience and several other parameters but generally Q_1^{*} can be equated to Q_1 without much loss of accuracy.

The microphone must be placed in the direct field of the source otherwise the useful sound pressure is dramatically reduced and the ratio between this and background noise is diminished. With the microphone in the direct field the intensity at the microphone for an omnidirectional source is as defined in eq. (7.1):

$$E_s = \frac{W}{4\pi r^2 c}\,.$$

The resulting useful acoustic power radiated by the loudspeaker is then given by eqs (9.1) and (9.2):

$$\begin{aligned} W_a &= \beta E_s \\ &= \frac{0{,}556\ WV}{4\pi r^2 c T} \cdot \frac{Q_m}{Q_1} \\ &= 0{,}044\ \frac{W}{cr^2} \cdot \frac{V}{T} \cdot \frac{Q_m}{Q_1}\,. \end{aligned} \tag{9.3}$$

The public address system necessary for a certain hall can thus be specified using the expressions given here. The microphone(s) must be placed in the indirect field of the loudspeaker(s) and the possibility of acoustic feedback avoided. The volume of the hall, V, and the reverberation time, T, are easily obtained. The energy density of the sound source at the microphone, E_o, must be known and this involves knowledge of the sound source itself and the distance from the microphone. A normal speaking voice delivers an acoustic power of about 20 µW and can be raised to 1 mW when shouting.

Typically, in a 5000 m^3 hall with a 100 W amplifier and a loudspeaker efficiency of 1% a singer would need to hold the microphone about 2,5 cm from his or her mouth to obtain the best from the public address system and exclude acoustic feedback. The distanced may vary and calculations can be made to derive the necessary system parameters (microphone sensitivity, amplification and so forth) for a much larger separation between the microphone and user.

It is preferable that as few microphones as possible are used, as each one will pick up some background noise and thus add to the level of non-useful sound amplified by the system.

The quality of the overall sound is obviously dependent on the quality of the system as well as the hall characteristics. Both calculation and experiment can enable the optimum deployment of loudspeakers to be realized and suitable sound levels to be used but, clearly, good quality is not going to be achieved without a suitable loudspeaker system. Conversely, a good quality loudspeaker can make the best of a bad environment.

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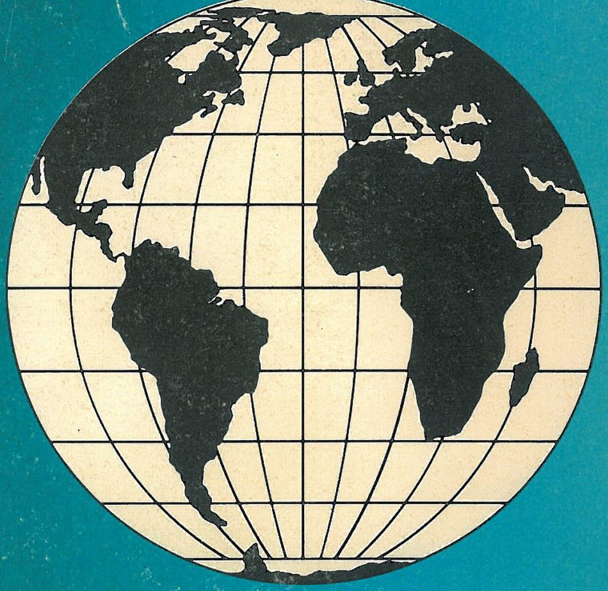

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