Chapter

5.1

Electroacoustic Transducers

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5.1.1 Introduction

Conversion from electrical signals to acoustic signals ordinarily does not involve direct electroacoustic transformation; the electrical signal is transformed into mechanical vibration, which then is transformed into an acoustic signal.

The following transducers are used in the audio field generally as electromechanical transducers: electrodynamic transducers, electromagnetic transducers, electrostatic transducers, and piezoelectric transducers.

5.1.2 Basic Equations and Features of Dynamic Transducers

Among the various forms of transducers listed above, the electrodynamic type is the basis for the design of the majority of loudspeakers in use today. Invented by C. W. Rice and E. W. Kellogg in 1925, when combined with the vacuum-tube amplifier, it provided the means for the use of audio technology in applications far greater than the telephone, introduced 50 years earlier by Alexander Graham Bell. Figure 5.1.1 shows the principle of operation. A permanent magnet and magnetic-pole pieces form a uniform magnetic field in the gap. The coil vibrating direction is at right angles to the magnetic field so that the force acts on the coil in accordance with the Fleming rule. This relationship is expressed by

$$F_d = BlI \tag{5.1.1}$$

Where:
F_d = driving force, N
B = flux density, Wb/m^2
l = total length of coil, m
I = current flowing into coil, A

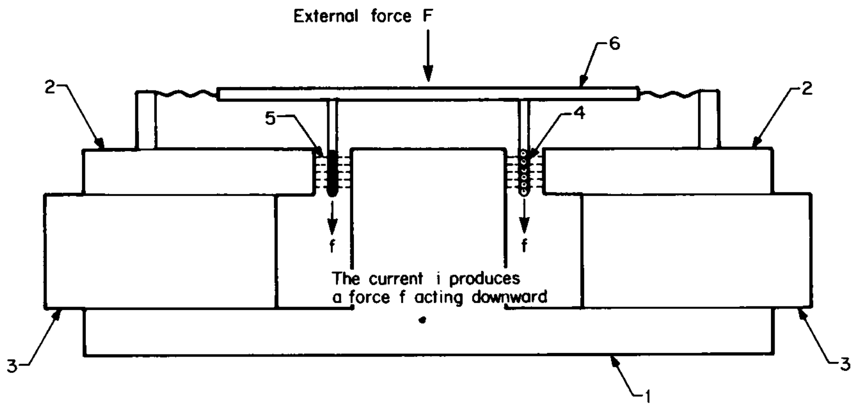


Figure 5.1.1 Simplified form of a moving-coil transducer consisting of a voice coil cutting a magnetic field of a flux density *B*. 1, 2 = pole pieces; 3 = permanent magnet; 4 = voice coil; 5 = magnetic flux; 6 = diaphragm.

Assuming the velocity at which a coil moves by means of driving force F_d to be v, the electromotive force E_d arising from this movement is in the opposite direction to the direction of current I. Therefore, E_d is determined by

$$E_d = -Blv \tag{5.1.2}$$

Where:
E_d = counterelectromotive force (V)
v = moving-coil velocity (m/s)

BI in Equations (5.1.1) and (5.1.2) is called the *power coefficient A*, which shows the conversion efficiency of a dynamic transducer. Assuming the mechanical impedance of the vibrating system as viewed from the coil side to be Z_m, the force acting on the coil corresponds to a summation of external forces F and driving forces F_d, which is balanced with drag $Z_m v$.

$$F + F_d = Z_m v \tag{5.1.3}$$

Where:
F = external force, N
F_d = driving force, N
Z_m = mechanical impedance of the vibrating system, mechanical ohms

By substituting Equation (5.1.1), F is found as follows

$$F = Z_m v - AI \tag{5.1.4}$$

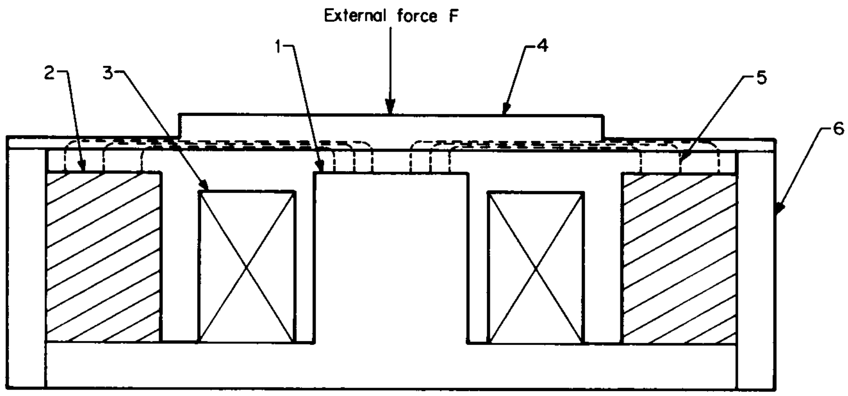


Figure 5.1.2 Simplified form of an electromagnetic transducer. 1 = pole piece; 2 = permanent magnet; 3 = drive coil; 4 = diaphragm; 5 = magnet flux; 6 = frame.

In the electrical system, assuming the electrical impedance of the driving coil to be Z_e the total voltage at the coil terminals corresponds to a summation of E and E_d, whereby the following equation is obtained

$$E + E_d = Z_e I \tag{5.1.5}$$

Where:
E = voltage applied across coil terminals (V)
Z_e = electrical impedance of coil (Ω)

When Equation (5.1.2) is substituted, E is determined by

$$E = Z_e I + Av \tag{5.1.6}$$

Thus, Equations (5.1.4) and (5.1.6) are basic equations of the dynamic mechanical-electrical systems.

5.1.2a Basic Equations and Features of Electromagnetic Transducers

For an electromagnetic transducer, a magnetic diaphragm placed in a static magnetic field, in which a permanent magnet supplies the steady magnetic flux, is vibrated in an ac magnetic field formed by signal current flowing into a coil, thus generating a sound. This principle is shown in Figure 5.1.2. In this figure, assume that the diaphragm is subjected to attraction force F_m by the static magnetic field and the external force F. At this time, the diaphragm vibrates from a summation of static displacement ξ_s by the attraction force in the static magnetic field and by the dynamic displacement generated by an ac magnetic field and external force F. Assuming this to be ξ, ξ is expressed by

$$\xi = \xi_s + \xi_d \tag{5.1.7}$$

Where:
ξ = total displacement, m
ξ_s = static displacement, m
ξ_d = dynamic displacement, m

Assuming the equivalent circuit of the mechanical system of the diaphragm to be a single-resonance circuit with the number of degrees of freedom equal to 1, it may be regarded as being composed of the lumped constant of equivalent mass, the mechanical resistance, and the stiffness s. Therefore, from force-balanced conditions, the following is established

$$F + F_m = m\frac{\partial^2\xi}{\partial t^2} + r\frac{\partial\xi}{\partial t} + s\xi \tag{5.1.8}$$

Where:
F = external force, N
F_m = attraction force by static magnetic field, N
m = equivalent mass, kg
r = mechanical resistance, N/m
s = stiffness, Ns/m

If the resistance is ignored, since it is quite negligible compared with magnetic resistance in the air space, the following relation is obtained

$$Z_m = r + j\omega m - j(s - s_n)/\omega \tag{5.1.9}$$

$$A = \mu_0 s_n U_0/g_0^2 \tag{5.1.10}$$

$$s_n = \mu_0 S U_0^2/g_0^2 \tag{5.1.11}$$

$$Z_e = Z_c + j\omega L_m \tag{5.1.12}$$

$$L_m = \mu_0 n^2 S/g_0 \tag{5.1.13}$$

$$F = Z_{m^v} - AI \tag{5.1.14}$$

$$E = Z_e I + Av \tag{5.1.15}$$

Where:
Z_m = mechanical impedance of the vibrating system, mechanical ohms
ω = angular frequency, rad/s
A = force factor, N/A
s_n = negative stiffness, Ns/m
L_m = inductance, H
Φ= total magnetic flux in space, Wb
B = flux density, Wb/m^2
μ_0 = magnetic permeability in space, H/m
U_m = magnetic motive force of magnet, A/m
S = magnetic-pole area, m^2
g_0 = quiescent space length in magnetic-force-free conditions, m
n = number of coil windings, turns
I = current flowing into coil, A
Z_c = coil electrical impedance, Ω

The difference between this transducer and the magnetic or dynamic transducer, in addition to the gap, is that negative stiffness in Equation (5.1.11) is generated. This stable condition is as follows

$$s < U_0/2\mu_0 S_0 g_0 R_{\text{air}}^{2} \tag{5.1.16}$$

where $R_{\text{air}}{}^2$ = magnetic resistance out of the air space, A/m. This relationship is shown in Figure 5.1.3. Other differences are that because the coil is fixed, reliability is high and construction is simple, and that if the frequency is high, the force factor becomes small because of the coil inductance, thereby reducing efficiency.

5.1.2b Basic Equations and Features of Electrostatic Transducers

In the electrostatic transducer, when voltage is applied to two opposite conductive electrodes, an electrostatic attraction force is generated between them, and the action of this force causes a conductive diaphragm to be vibrated, thereby emitting sound. Figure 5.1.4 shows the construction. Electrostatic attraction force F_s , when signal voltage E is applied to polarized E_0 , is

$$F = \frac{\varepsilon_0 S(E_0 + E)^2}{2(g_0 - \xi_0 - \xi_d)^2} \tag{5.1.17}$$

Where:
F = static attraction force, N
ε_0 = dielectric constant, F/m
S = electrode area, m^2
E_0 = polarized voltage, V
E = signal voltage, V
g_0 = interelectrode distance, m

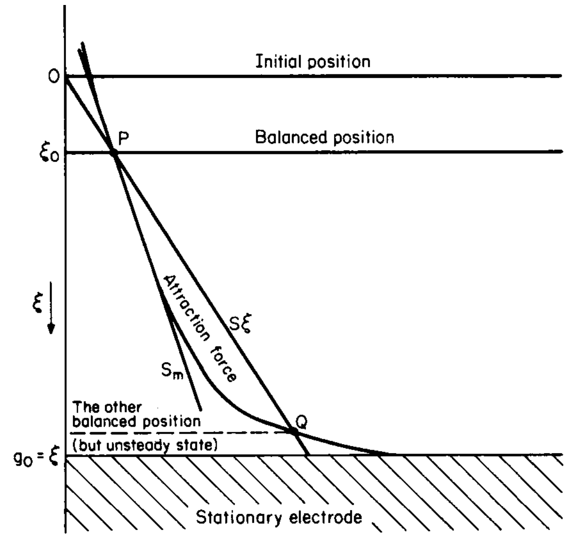


Figure 5.1.3 Static displacement shows balancing the attraction and the recover force.

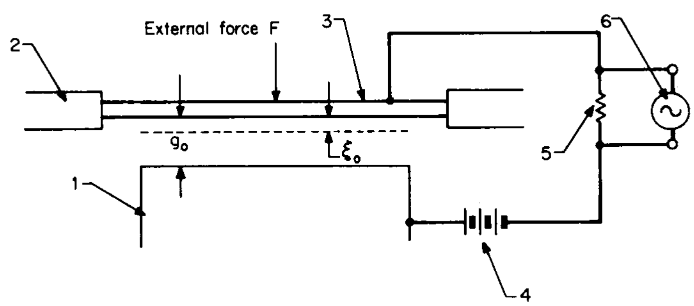


Figure 5.1.4 Cross-sectional view of an electroacoustic transducer. 1 = back electrode; 2 = clamping ring; 3 = diaphragm with electrode; 4 = polarizing power supply; 5 = polarizing electrical resistance; 6 = signal source.

ξ_s = static displacement, m
ξ_d = signal displacement, m

Considering the correspondence between electromagnetic and electrostatic types, Equation (5.1.17) is as shown in Table 5.1.1.

The basic equations of the electrostatic type are

$$s_n = \varepsilon_0 S E_0^2 / g_0^3 \tag{5.1.18}$$

Table 5.1.1 Correspondence Between Electromagnetic and Electrostatic Types

Electromagnetic	nI	U_0	μ_0	$F_m\Phi$
Electrostatic	E	E_0	ε_0	$F_s q$

$$A = \varepsilon_0 S E_0 / g_0^2 \tag{5.1.19}$$

$$Z_m = r + j\omega m - j(s - s_n)/\omega \tag{5.1.20}$$

$$Y_s = j\omega(\varepsilon_0 S / g_0) \tag{5.1.21}$$

$$F = Z_m v - AE \tag{5.1.22}$$

$$I = Y_s E + Av \tag{5.1.23}$$

Where:
Z_m = mechanical impedance of the vibrating system, mechanical ohms
Y_s = electrical admittance of electrostatic capacity before displacement
F = external force, N
I = current, A
s_n = negative stiffness, N/m
A = force factor, N/V
r = mechanical resistance, Ns/m
m = mass, kg
s = diaphragm stiffness, N/m
ω = angular frequency, rad/s

Equations (5.1.22) and (5.1.23) are basic equations of the electrostatic transducer. Sensitivity of this transducer can be obtained by increasing the polarized voltage and reducing the distance between electrodes. Since the electrostatic type, unlike the electromagnetic type, has nothing to restrict attraction force, the force of the diaphragm to stick to the electrode is infinite. Therefore, the diaphragm requires a very large stiffness. Electrical impedance decreases inversely proportionally to the frequency since it is quantitative. This type is simply constructed, and since it has relatively good characteristics, it is used for high-range speakers and headphones.

5.1.2c Basic Equations and Features of Piezoelectric Transducers

If a crystal section is distorted with a force applied in one direction, positive and negative charges appear on the opposite surfaces of the crystal. This is called the piezoelectric direct effect. When a field is applied to the crystal section from the outside, a mechanically distorted force is generated. This is a piezoelectric counter effect. Ferrodielectric substances, which exhibit such a phenomenon, are polarized. These include crystal, piezoelectric crystals such as Rochelle salts, titanium oxide, and lead zirconate titanate (PZT). In general, PZT, having high

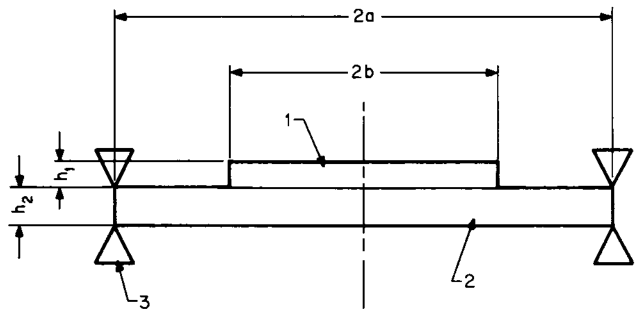


Figure 5.1.5 Simplified form of a monomorphic piezoelectric transducer. 1 = piezoelectric element (E_1, P_1 μ_1); 2 = metal plate (E_2, P_2 μ_2); 3 = supporting ring.

reliability and a reasonable price, is used as the piezoelectric element for the speaker. By using a configuration such as shown in Figure 5.1.5, the output sound level and resonance frequency can be determined. Power sensitivity q, when radian frequency $\omega \to 0$ is calculated by

$$q_0 = 20\log\left|\frac{K_1 U_0 Z_e}{E_0}\right| \tag{5.1.24}$$

Where:
q_0= power sensitivity
U_0 = volume velocity, m^3/s
Z_e = electrical impedance of piezoelectric element, Ω
E_0 = input voltage, V
K_1 = constant

Assuming displacement at the piezoelectric element and laminated metal sheet to be ξ' and displacement at the peripheral metal part to be ξ, U_0 is found as follows

$$U_0 = \int_0^b 2\pi r\xi' \; dr + \int_b^a 2\pi r\xi \, dr \tag{5.1.25}$$

Z, which is mainly a qualitative component, is determined by

$$Z = \frac{K_2}{\pi\omega\varepsilon_{33}T} \times \frac{h_1}{a^2\eta^2} \tag{5.1.26}$$

Where:
ε = dielectric constant of piezoelectric element
$\eta = b/a$

K_2 = constant

To find the optimum condition of η if $\mu_1 = \mu_2 = \mu$ with radius a, material thickness $h = h_1 + h_2$, and the piezoelectric constant d_{31}, $\varepsilon^T{}_{33}$ constant, the following is obtained

$$\frac{U_0\sqrt{Z}}{E} \propto \frac{\alpha(1+\beta)\sqrt{\beta}}{1+\alpha\beta} \times \frac{\mu[3+\mu-\eta(1+\mu)]}{(1+\mu)C+\eta^2\left[(1-\mu)C+2(1-\mu^2)\left(1-\frac{3}{2}\zeta+\frac{3}{4}\zeta^2\right)\right]} \tag{5.1.27}$$

$$C = (1-\mu_2)\left(\beta_2+\frac{3}{2}\beta\zeta+\frac{3}{4}\zeta^2\right)\alpha\beta+2\mu(1-\mu)\left(1-\frac{3}{2}\zeta+\frac{3}{4}\zeta^2\right) \tag{5.1.28}$$

$$\zeta = (1-\alpha\beta^2)(1+\alpha\beta) \tag{5.1.29}$$

Where:
$\alpha = Q_1/Q_2$
μ = Poisson ratio, defined as the charge density at any point divided by the absolute capacitivity of the medium

From the above, it is found that η = 0.5 to 0.8 is better. β is dependent on α, but when the relative sensitivity of various metals is compared, 0.2 < β < 1.0; therefore, aluminum is the best. The primary resonance frequency of the vibrator is

$$f_1 = \frac{2.22^2 h_1}{2\pi a^2\beta}\sqrt{\frac{Q^2}{3p_2(1-\mu_2)}\left(1-\frac{3}{2}\zeta+\frac{3}{4}\zeta\right)} \tag{5.1.30}$$

Where:
f_1 = primary resonance frequency, Hz
Q_2 = Young's modulus, N/m^2
P_2 = density, kg/m^3

Assuming radius α, thickness h, and Poisson's ratio to be constant, C is determined by

$$f_1 \propto C\sqrt{\left(1-\frac{3}{2}\zeta+\frac{3}{4}\zeta^2\right)/\beta}$$

$$C = \sqrt{Q_2/P_2} \tag{5.1.31}$$

where C = sound velocity, m/s. Furthermore, the resonance frequency of the vibrator is expressed as

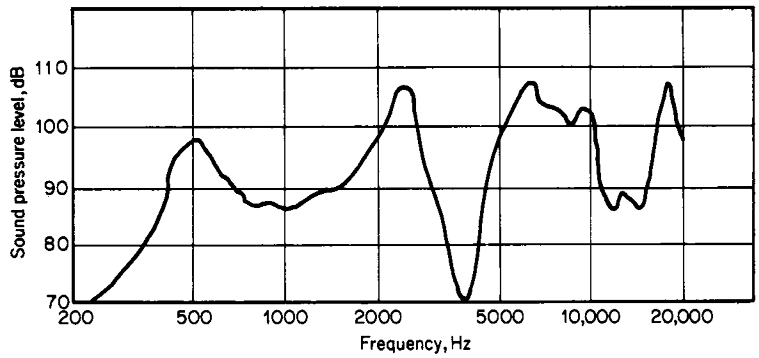


Figure 5.1.6 Frequency characteristics of a typical monomorphic piezoelectric transducer.

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{s_o}{m_o}} \tag{5.1.32}$$

Where:
m_0 = vibrator mass, kg
s_0 = vibrator stiffness, N/m

However, to reduce mechanical Q, a small $s_0 \times m_0$ is preferable, and therefore aluminum is the best material. Figure 5.1.6 shows the sound-pressure-frequency characteristics of a speaker with this construction.

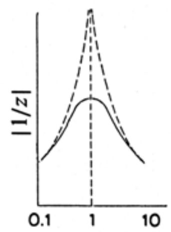
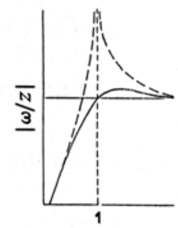
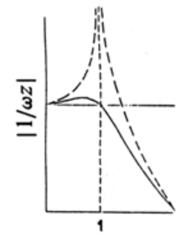
5.1.3 Control System and Its Acoustic Characteristics

For acoustic equipment, in the process of transforming electrical energy to acoustic energy, conversion from the electrical system to the mechanical system and from the mechanical system to the acoustic system is performed. The conversion process is expressed approximately by the equation

$$\frac{P}{E} = \frac{F}{E} \times \frac{V}{F} \times \frac{P}{V} \tag{5.1.33}$$

The left-hand term shows the ratio of electrical input to sound pressure, which should be kept constant regardless of frequency. However, the first term, the ratio of electrical input to driving force, and the third term, the ratio of diaphragm velocity V to sound pressure P on the right, are fixed by the conversion and radiation systems in the relationship with frequency. For example, the sound pressure of a direct-radiation type of speaker increases in proportion to frequency if the velocity V is constant. Consequently, if V/F decreases with frequency, the ratio is not related to frequency as a whole even when F/E is constant. This corresponds to a mass when the vibrat-

Table 5.1.2 Three Control Systems

	Resistance control	Mass control	Stiffness control
Z_m approximation	r	ωm	s/ω
$\upsilon v/F\|$	$1/r$	$1/\omega m$	ω/s
Characteristics			
Applications	Horn speaker	Direct radiant-type speaker	Headphone

ing system is regarded as a single resonance system, which is called *mass control*. Likewise, when *V*/*F* becomes unrelated to frequency, both the resistance control and the frequency increase; this is called *stiffness control*. Table 5.1.2 summarizes these characteristics.

5.1.4 Bibliography

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Chapter

5.2

Direct-Radiator Loudspeakers

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5.2.1 Introduction

The diameter of a speaker diaphragm normally ranges between a few centimeters and dozens of centimeters when high-amplitude sound must be produced. The following sections outline the basic principles involved in direct-radiator loudspeakers.

5.2.2 Piston Source in an Infinite-Plane Baffle

An actual diaphragm has many different oscillation modes, and its motion is complicated. On the assumption—for easier analysis—that the diaphragm is rigid, radiation impedance and directivity are considered for typical circular and rectangular shapes. As shown in Figure 5.2.1, part of a circular rigid wall is oscillating at a given velocity v exp ($j\omega t$). The upper part of this circular piston is subdivided into the micro area d_s, and when a micro part is oscillated by the piston, the total reaction force subjected from the medium side is calculated. Thus, the radiation impedance Z_R of the diaphragm is found from the ratio of this reaction force to the diaphragm's oscillating speed. This shows how effectively sound energy from the diaphragm is used. Radiation impedance in the circular diaphragm is shown in the following equation, and the results in Figure 5.2.2.

$$Z_R = (\pi a^2 pC)\left[1 - \left(\frac{J_1(2ka)}{ka}\right) + j\frac{S_1(2ka)}{ka}\right] \tag{5.2.1}$$

Where:
J_1 = Bessel function of the first order
S_1 = Struve function

Directional characteristics of the circular diaphragm are shown in the following equation, and the results in Figure 5.2.3.

$$D(\theta) = \left|\frac{2J_1(ka\sin\theta)}{ka\sin\theta}\right| \tag{5.2.2}$$

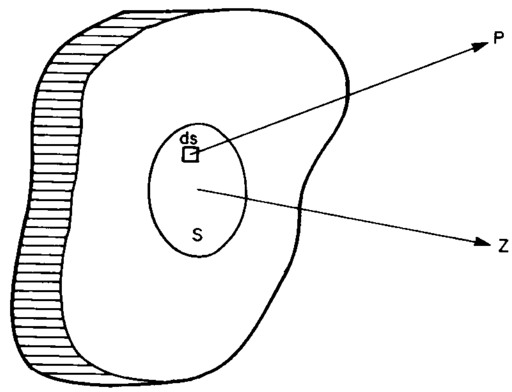

Figure 5.2.1 Piston on an infinite rigid wall.

Where:
$D(\theta)$ = ratio between sound pressures whose angles θ are in 0 and θ directions
θ = perpendicular on the surface center
k = number of waves
a = radius, m

Rectangular impedance is shown in Equation (5.2.3), directional characteristics in Equation (5.2.4), and the respective calculation results in Figure 5.2.4.

$$R(v,\sigma) = 1 - (2/\pi v^2)[1 + \cos(vq) + vq \sin(vq) - \cos(vp) - \cos(v/p)] + (2/\pi)[pI_1(v,\sigma)] + I_1(v,1/\sigma)/p$$

$$X(v,\sigma) = (2/\pi v^2)[\sin(vq) - vq \cos(vq) + v(p + 1/p) - \sin(vp) - \sin(v/p)] - (2/\pi)[pI_2(v,\sigma) + I_2(v,1/v)/p]$$

$$v = k\sqrt{S}$$

$$q = (\sigma + 1/\sigma) \tag{5.2.3}$$

Where:
v = nondimensional frequency
$p = \sqrt{\sigma}$

$$I_{1,2} = \int_{(\zeta - 1/2)}^{(\xi + 1/\xi)(1/2)} (1 - 1/\xi t^2)^{1/2} \begin{matrix}\cos \\ \sin\end{matrix}(vt)\,dt \tag{5.2.4a}$$

$$\xi = \sigma or - \sigma \tag{5.2.4b}$$

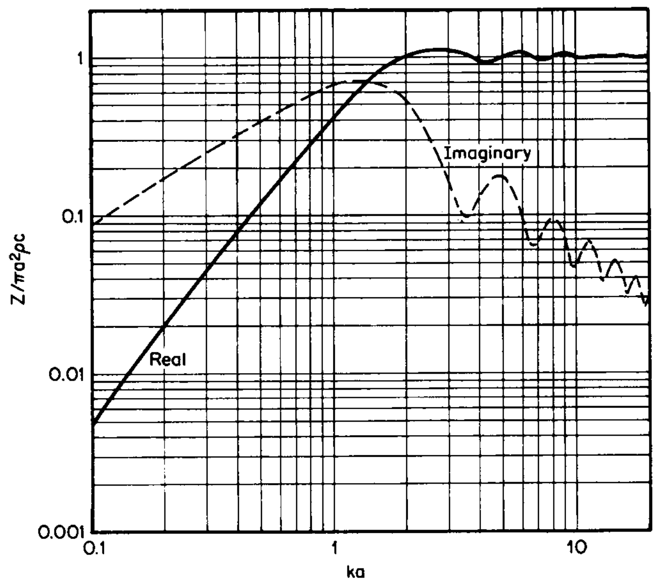

Figure 5.2.2 Radiation impedance for a rigid circular diaphragm in an infinite baffle as a function of $k\,a = 2\pi\,a/\lambda$.

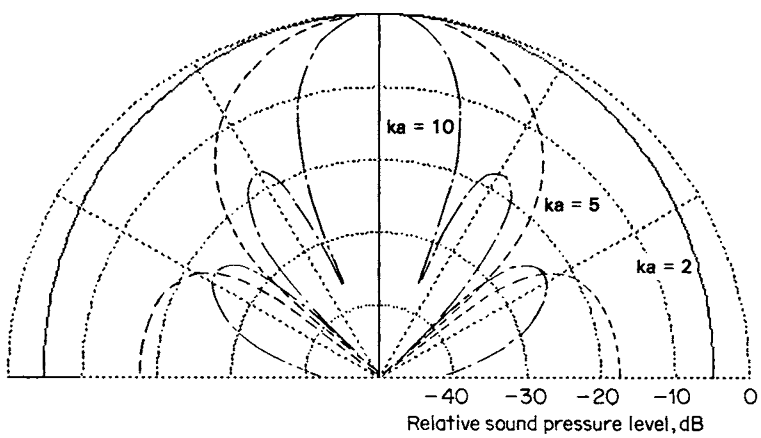

Figure 5.2.3 Directional characteristics of a circular diaphragm.

1, 2, subscripts of I, = cos for 1 and sin for 2.

$$D(\theta_1\theta_2) = \frac{\sin\,\phi_1}{\phi_1}\cdot\frac{\sin\phi_2}{\phi_2} \tag{5.2.4c}$$

$$\phi_{1,2} = \frac{\pi d_{1,2}}{\lambda}\sin\phi_{1,2} \tag{5.2.4d}$$

Where:
$D(\theta_1, \theta_2)$ = ratio between sound pressures in 0 and θ_1/θ_2 directions ($\theta_1 = \theta_2 = 0$ is a perpendicular of the center on the rectangular surface)
λ = wavelength, m
$d_{1,2}$ = length of each side of rectangle, m

Radiation impedance shows how effectively sound energy is radiated, while directional gain is used to show how expanding sound energy is radiated in space. The ratio of total acoustic energy W is found by integrating the sound strength from that on a spherical surface a distance r from the sound source with the sound strength that exists on the same point from the nondirectional sound source that emits the same energy. This is expressed in decibels:

$$W = \frac{r^2}{pC}\int_0^{2\pi}\int_0^{\pi}|\dot{P}(r,\theta,\phi)|^2\sin^2\theta\ \ (d\theta)(d\phi) \tag{5.2.5}$$

$$DI = 10\log\left(\frac{4\pi r^2}{W}\cdot\frac{|\dot{P}_{max}|^2}{pC}\right) \tag{5.2.6}$$

Where:
W = total acoustic energy, W
r = distance in the maximum sound pressure direction for standardization, m
$P_{\max}$ = sound pressure at distance r, N/m^2
DI = directivity index (directional gain), dB

5.2.2a Baffle Shape and Acoustic Characteristics

In the preceding section an infinite baffle was discussed, but such a baffle cannot be put to practical use. Consequently, it is necessary to precheck the types of characteristics that can be obtained when a definite baffle is installed in a speaker. Because the sounds radiated to the front baffle and reflected to the rear are opposite in phase, the difference in distance between the passes of sound through the rear and front baffles from a speaker is canceled by the front and rear sounds of one-half even multiples and added to each other by the sounds of odd multiples.

Therefore, high and low sound pressures occur. To avoid this, the speaker is installed off center, resulting in a baffle with a complicated shape. One side should be a few times longer than the wavelength of interest. However, this shape does not produce favorable characteristics, and this

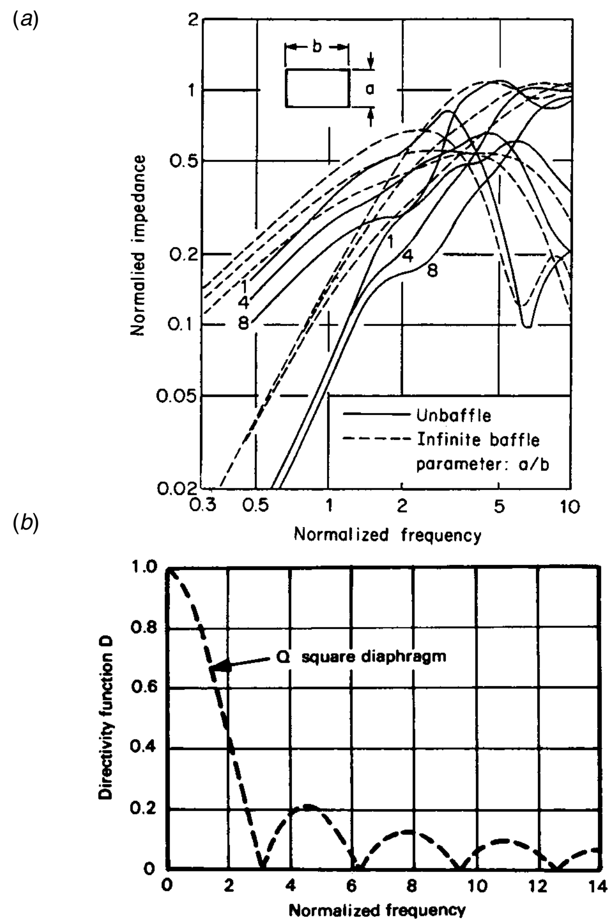

Figure 5.2.4 Diaphragm characteristics: (*a*) radiation impedance for a rigid rectangular diaphragm, (*b*) directivity function for a rigid square diaphragm. Note that in (*a*) solid lines, which have been calculated by using the finite element method (FEM), are instructive for practical designs.

type of baffle is not often used in practical applications. Typical baffle characteristics are shown in Figures 5.2.5 and 5.2.6.

5.2.2b Acoustic Characteristics of Rigid Disk with Constant-Force Drive

This section comments on the types of sound-pressure-frequency characteristics produced at a remote distance on the center axis of a diaphragm and the acoustic output obtained therefrom when a circular piston diaphragm is placed in an infinite rigid wall and driven at a given force. When a circular diaphragm with radius α is subjected to a constant force F moving in the axial direction, sound pressure P is determined by

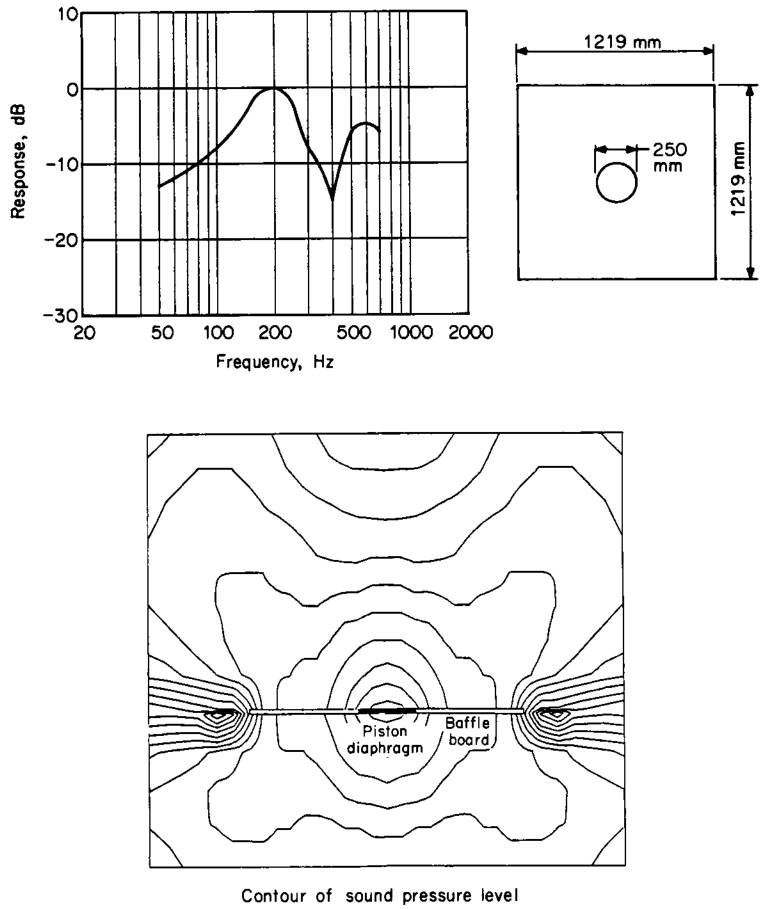

Figure 5.2.5 Pressure-response-frequency characteristics for a direct radiator installed in the center of a finite baffle, estimated by FEM.

$$\dot{P} = p_{\theta}\frac{\partial\phi}{\partial t} = j\frac{\omega p_{\theta} a_2}{2r}\exp(-jkr)\cdot\dot{v}$$

$$= j\frac{\omega p_{\theta} a^2}{2r}\exp(-jkr)\cdot\frac{\dot{F}}{\dot{Z}} \tag{5.2.7}$$

The absolute value $|\dot{P}|$ of sound pressure is shown in the equation

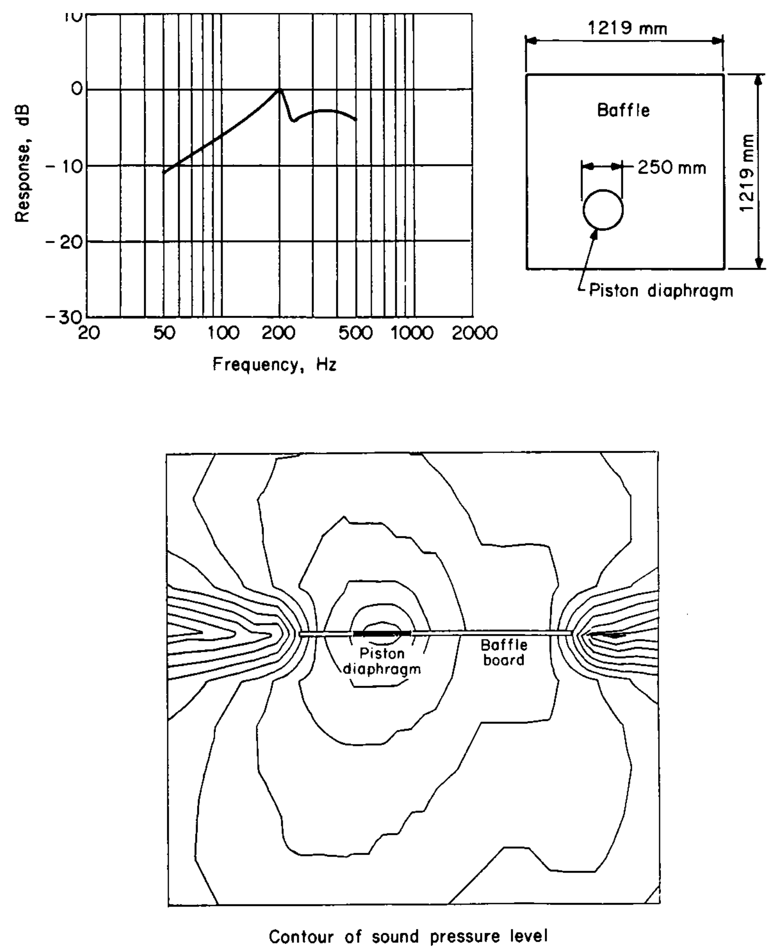

Figure 5.2.6 Pressure-response-frequency characteristics of a direct radiator installed off center, estimated by FEM.

$$|\dot{P}| = \frac{\omega p_\theta a^2}{2r}\left|\frac{\dot{F}}{\dot{Z}}\right| \tag{5.2.8}$$

Where:
P = sound pressure on the axis, N/m^2
p_θ = gas density, kg/m^3
θ = velocity potential
a = diaphragm radius, m
r = distance from diaphragm on the axis, m
$\dot{P}$ = driving force, N

ω = angular frequency, rad/s

When the oscillation system is regarded as a single resonance system, $\dot{Z}$ is obtained as follows

$$\dot{Z} = r_m + j\omega m + \frac{1}{j\omega C_m} \tag{5.2.9}$$

Where:
$\dot{Z}$ = mechanical impedance of oscillation system, mechanical ohms
r_m = mechanical resistance of oscillation system, N/m
C_m = oscillation-system compliance, m/N
m = mass of oscillation system, kg

Therefore, sound pressure $|\dot{P}|$ is determined by

$$|\dot{P}| = \frac{\omega^2 p_\theta a^2 C_m}{2r}|\dot{F}| \quad \omega < \omega_\theta \tag{5.2.10}$$

$$|\dot{P}| = \frac{p_\theta a^2}{2rm}|\dot{F}| \quad \omega > \omega_\theta \tag{5.2.11}$$

This is shown in Figure 5.2.7.

$$W_a = \frac{\pi p_\theta a^4 \omega^4 C_m}{2c_2}|\dot{F}|^2 \quad ka < 1 \quad \omega < \omega_\theta \tag{5.2.12}$$

$$W_a = \frac{\pi p a^4 \omega^2}{2cr^2}|\dot{F}|^2 \quad ka < 1 \quad \omega = \omega_\theta \tag{5.2.13}$$

$$W_a = \frac{\pi p a^4}{2cm^2}|\dot{F}|^2 \quad ka < 1 \quad \omega > \omega_\theta \tag{5.2.14}$$

$$W_a = \pi p a^2 c \omega^2 C_m^2 |\dot{F}|^2 \quad ka > 1 \quad \omega > \omega_\theta \tag{5.2.15}$$

$$W_a = \frac{\pi p a^2}{r^2}|\dot{F}|^2 \quad ka > 1 \quad \omega = \omega_\theta \tag{5.2.16}$$

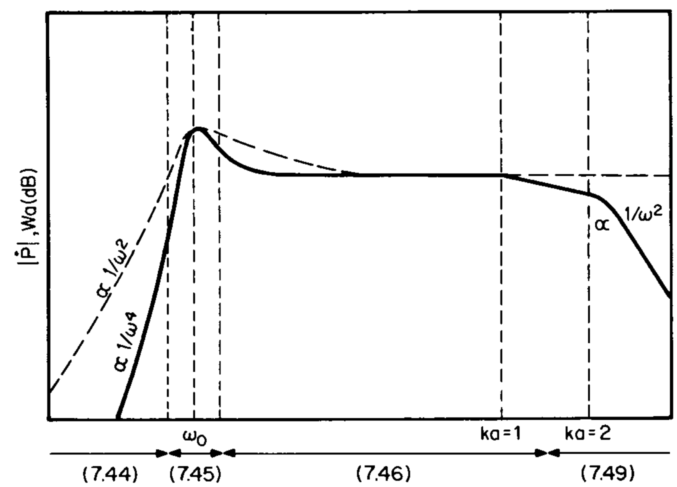

Figure 5.2.7 Acoustic power and pressure-response-frequency characteristics of a piston source in an infinite-plane baffle.

$$W_a = \frac{\pi \rho a^2 c}{\omega^2 m} |\dot{F}|^2 \quad ka > 1 \quad \omega > \omega_\theta \tag{5.2.17}$$

Where:
C = sound velocity, m/s
k = number of waves
ω = resonance angular frequency, rad/s

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Chapter

5.3

Dynamic-Type Direct-Radiation Speakers

Katsuaki Satoh

5.3.1 Introduction

The dynamic direct-radiation loudspeaker is divided broadly into the following components:

- Magnetic circuit
- Drive coil
- Diaphragm
- Support system
- Frame

Although the construction of each speaker design is unique, the underlying fundamentals are the same.

5.3.2 Operational Details

The typical configuration of a dynamic-type direct-radiation speaker is shown in Figure 5.3.1. Most magnetic circuits are of the external type, using a ferrite magnet designed to generate a magnetic-flux density of a few thousand to a few ten thousand G in an approximately 1- to 2-mm air gap formed by the north and south poles. To control distortion, the drive coil provided in the air gap is designed so that it does not move out of the uniform magnetic field formed by the magnetic pole because of vibration. Thus, the drive coil used has approximately 0.1-mm-diameter windings of several turns. The impedance normally is a multiple of 4 Ω. The diaphragm is available in a variety of shapes and materials, as described later. The dust cap is used to prevent dust from intruding into the magnetic air gap; when the cap must function as a damper, a permeable material is used. Thus, the centering suspension and cone suspension function to: 1) support these vibration systems, 2) hold the drive coil in the magnetic air gap, and 3) generate deemphasis in the axial direction.

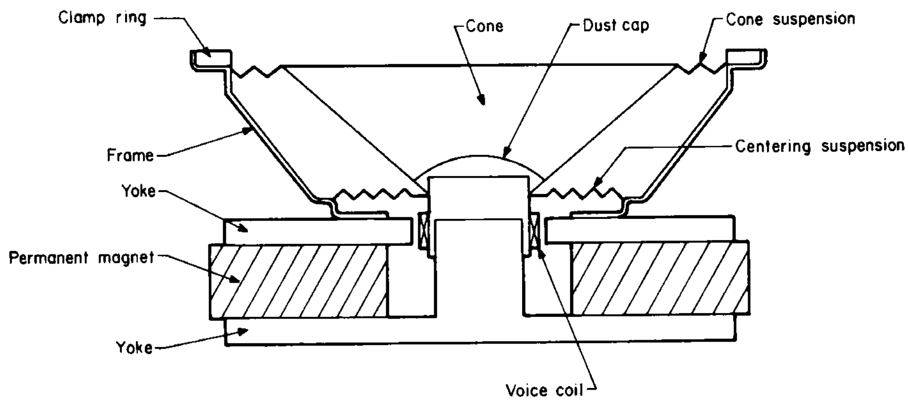


Figure 5.3.1 Structure of the dynamic direct-radiator loudspeaker.

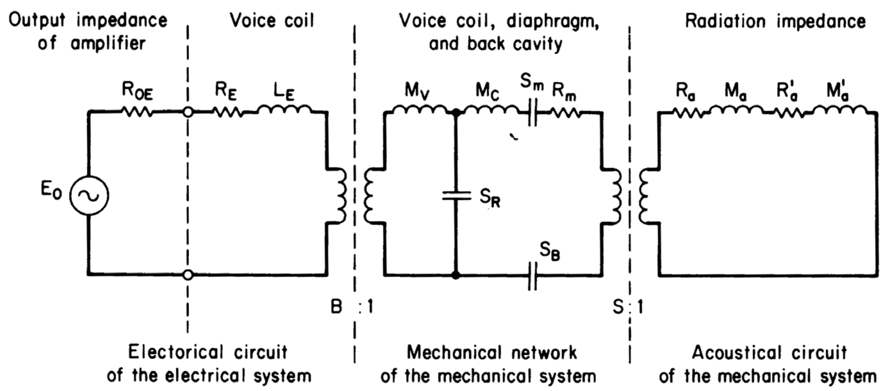


Figure 5.3.2 Electromechanical equivalent circuit. R_{OE} = output impedance of amplifier, Ω ; R_E = resistance of voice coil, Ω ; L_E = inductance of voice coil, H; M_V = mass of voice coil, kg; S_R = stiffness between cone and voice coil, N/m; M_C = mass of cone, kg; S_B = stiffness of back cavity, N/m; R_a, R_a' = radiation resistance of diaphragm, mechanical ohms; M_a , M_a' = radiation mass of diaphragm, kg; Bl = force factor; S = area of diaphragm, m^2.

5.3.2a Equivalent Circuit and Frequency Response

Figure 5.3.2 shows the equivalent circuit of a dynamic type of speaker. The sound-pressure-frequency characteristics of the equivalent circuit are shown in Figure 5.3.3. An examination of these characteristics divided by frequency bands follows.

In low ranges, the diaphragm and support system are free from split vibration, but they are considered to be a single resonance system. Thus, an equivalent circuit as shown in Figure 5.3.4*a* is produced. The velocity, amplitude characteristics, and sound pressure characteristics on the axis are as shown in Figure 5.3.4*b*. As can be seen from this figure, Q_0 determines sound pressure characteristics near the resonance frequency. If all element constants are found, Q_0 can be

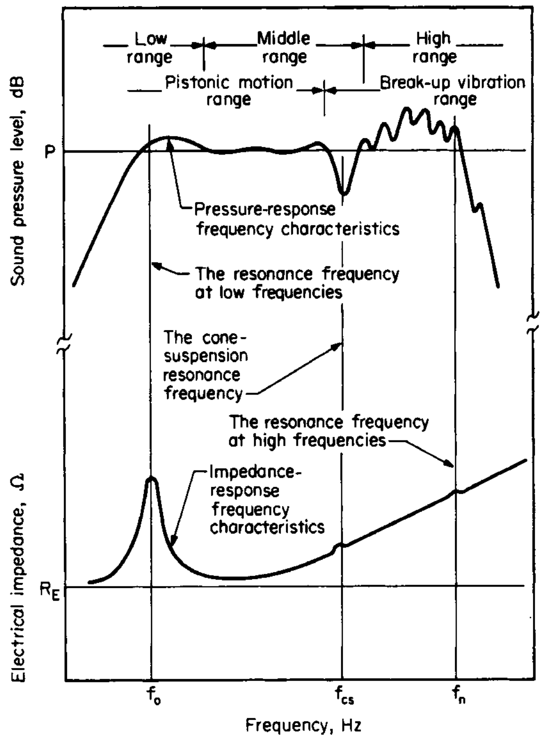


Figure 5.3.3 Frequency characteristics of the dynamic direct-radiator loudspeaker.

obtained by calculation, but these constants must often actually be found by measurement. Voice-coil impedance near the resonance frequency is expressed as a sum of electrical impedance and motion impedance. That is,

$$Z_e = Z_c + \frac{A^2}{Z_M} \tag{5.3.1}$$

When $R_c \gg \omega L$,

$$Z_e = R_c + \frac{A^2}{Z_M} = Re + \cfrac{1}{\cfrac{1}{\cfrac{A^2}{r_m}} + \cfrac{1}{\cfrac{A^2}{j\omega m}} + \cfrac{1}{\cfrac{A^2}{\cfrac{1}{j\omega C_m}}}} \tag{5.3.2}$$

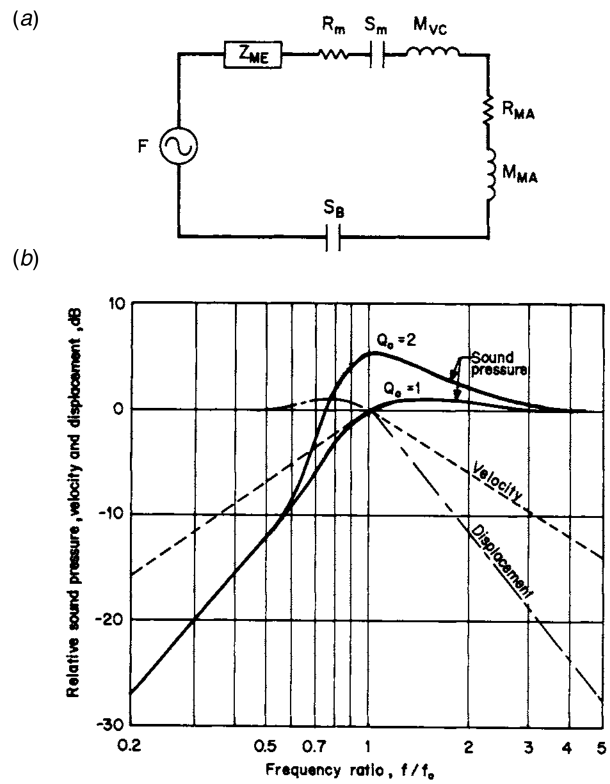


Figure 5.3.4 Loudspeaker characteristics: (*a*) mechanical equivalent circuit at a low-frequency range, (*b*) frequency characteristics of sound pressure, velocity, and displacement. Z_{ME} = motional impedance, mechanical ohms; R_m = resistance of vibrating system, mechanical ohms; S_m = stiffness of vibrating system, N/m; M_{VC} = mass of vibrating system, kg; R_{MA} = resistance of radiating system, mechanical ohms; M_{MA} = mass of radiating system, kg; S_B = stiffness of back cavity, N/m.

The vector impedance locus is shown in Figure 5.3.5. From these results, Figure 5.3.6 is obtained, and Q_0 can be found directly from electrical impedance. In midrange, cone suspension less rigid than the diaphragm produces a split vibration. This phenomenon appears typically near 1000 Hz with a speaker using a paper-cone diaphragm. The analytical results of this condition, using the finite-element method (FEM), are shown in Figure 5.3.7*a*. To eliminate this, damping material is coated and the shape is redesigned, thus controlling the resonance. For the diaphragm, specific resonance starts to appear, a peak and a dip in sound pressure response occur, and a strain may result. Regarding this shortcoming, the results of analysis by FEM are shown in Figure 5.3.7*b*. To control this specific resonance, materials with a larger internal loss are used, the shape of the diaphragm is changed from a simple cone to a paracurve, and corrugation is provided. Furthermore, when the frequency rises, elastic deformation concentrates at the junction between the drive coil and the diaphragm, and stiffness S_R appears there equivalently. Therefore,

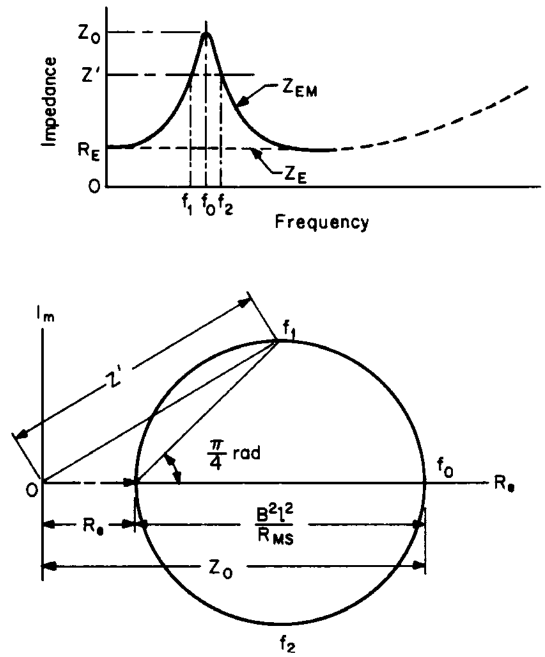


Figure 5.3.5 Loudspeaker voice-coil impedance and impedance locus. R_E = resistance of voice coil, Ω; $Z = Z_0 / \sqrt{2}$, Ω; B = magnetic-flux density in the gap, Wb/m^2; l = length of wire on voice-coil winding, m; R_{MS} = resistance of vibrating system, mechanical ohms; f_0 = resonance at low-frequency range, Hz; f = frequency at –3 dB, Hz.

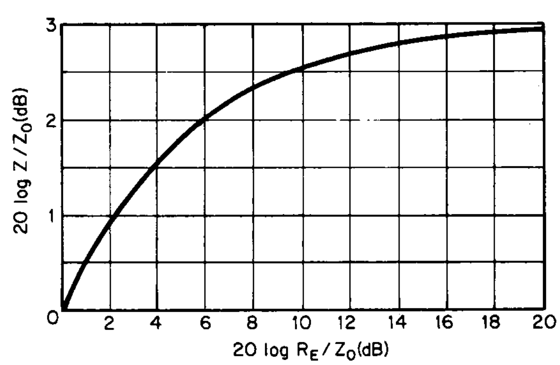


Figure 5.3.6 Relation between Z' and Z.

sound pressure is suddenly lowered at a higher level than the resonance frequency by S_R and the diaphragm mass, which actually presents the playback limit.

(a)

(b)

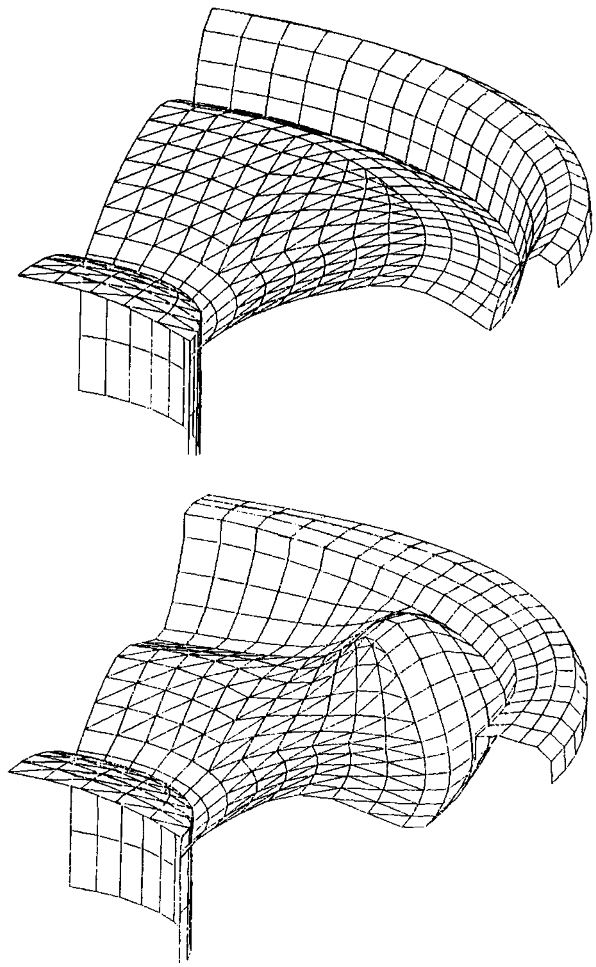


Figure 5.3.7 Breakup vibrating modes, estimated by FEM: (*a*) fundamental mode of the suspension, (*b*) axial mode of the cone.

5.3.2b Efficiency

Speaker efficiency is expressed in terms of the ratio of electrical input to acoustic output. The electrical input with due regard to only the real-number part in the equivalent circuit in Figure 5.3.2 is expressed by

$$W_e = R_c I^2 \tag{5.3.3}$$

Where:
W_e = electrical input, W

R_c = coil resistance, Ω
I = current flowing into the coil, A

Acoustic output W_a is determined by

$$W_a = r_R \left| \frac{F}{Z_m} \right| \tag{5.3.4}$$

Where:
W_a = acoustic output, W
r_R = acoustic radiation resistance, N/m
F = driving force, N
Z_m = vibration-system mechanical impedance, mechanical ohms

Consequently, efficiency η is found as follows

$$\eta = \frac{W_a}{W_e + W_a}$$

$$= \frac{1}{1 + W_e / W_a} \tag{5.3.5}$$

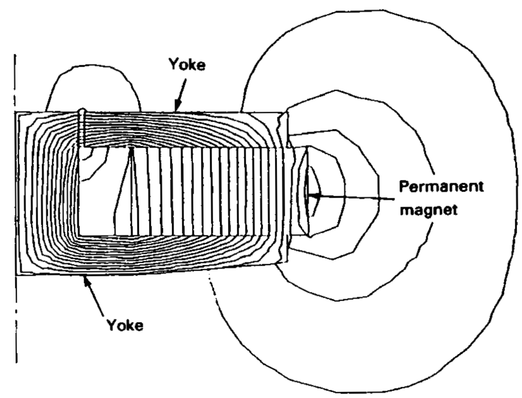
With the diaphragm considered as a stiff disk, if it is an infinite baffle board, W_a can employ the approach shown previously in this chapter. If the acoustic output is constant, η is determined by

$$\eta = \frac{1}{1 + \dfrac{2cm^2 R_c}{\pi p a^4 B^2 l^2}} \times 100 \tag{5.3.6}$$

Where:
η = conversion efficiency, percent
p = air density, kg/m^3
c = sound velocity, m/s
m = vibration-system mass, kg
a = effective radius of diaphragm, m
B = flux density, We/m^2
l = coil length, m
R_c = coil resistance, Ω

In Equation (5.3.6), the magnitude on the second term normally is approximately 50. The efficiency is only a few percentage points, which is very low.

(*a*)



(*b*)

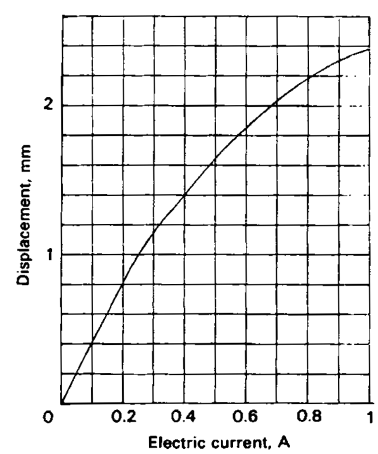


Figure 5.3.8 Flux distribution of the half-magnetic circuit: (*a*) typical flux lines in an air gap estimated by FEM, (*b*) relation between the electric current and the displacement.

5.3.2c Nonlinear Distortion

The strain that takes place in a dynamic speaker includes several types of distortion: 1) driving-force distortion, 2) support system distortion, 3) air distortion, and 4) frequency-modulation distortion.

Driving-Force Distortion

Driving-force distortion occurs mainly because a drive coil flows out from the uniform magnetic field as the amplitude varies, whereby the driving force ceases to be proportional to current. Figure 5.3.8*a* shows the magnetic-flux distribution and magnetic-flux-density distribution near the magnetic pole in a typical magnetic circuit. Figure 5.3.8*b* shows the relationship between the power coefficient generated by a coil located in such a magnetic circuit and the coil displacement. Consequently, the following nonlinear differential equation must be solved. Here, a study

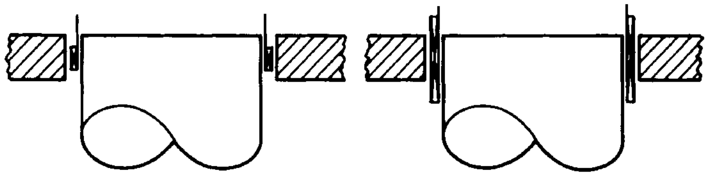


Figure 5.3.9 Relation between the voice coil and the magnetic circuit for reducing distortion: (*a*) voice coil shorter than the air gap, (*b*) voice coil longer than the air gap.

may be made at an ultralow frequency with a large amplitude. Therefore, radiation impedance can be approximated with radiation mass and vibration-system impedance with stiffness. The basic equations are as follows, assuming the stiffness to be linear:

$$M_{MA}\frac{d^2\xi}{dt^2} = m\frac{d^2\xi}{dt^2} + r\frac{d\xi}{dt} + s_n - A(\xi)I(\xi) \tag{5.3.7}$$

$$E_0 \sin\omega t = RI(\xi) + A(\xi)\frac{d\xi}{dt} \tag{5.3.8}$$

Where:
$A(\zeta)$ = ζ function force factor, N/A
M_{MA} = radiation mass, kg
s_n = vibration-system stiffness, N/m
$E_0 \sin\omega t$ = applied voltage, V
R = coil resistance, Ω
$I(\zeta)$ = current of ζ function, A
ζ = displacement, m

To reduce this distortion, it is preferable to adopt a method of decreasing the coil-winding width as shown in Figure 5.3.9*a* so that it is not off the magnetic field or sufficiently increasing the coil-winding width as shown in Figure 5.3.9*b*. Because driving-force distortion develops as current distortion, this distortion can be reduced by detecting current flowing into a coil with a microresistance and feeding this current back into the input terminal of the amplifier. This is shown in Figure 5.3.10. Other driving-force distortions include a strain generated by hysteresis of the magnetic-circuit yoke. This can be substantially improved by using a silicon steel plate for a yoke and a magnetic material of small conductivity. This technique is shown in Figure 5.3.11.

Support-System Distortion

Support-system distortion is such that since elasticity of suspension is nonlinear, force and displacement cease to be proportional to each other. The force-versus-displacement characteristics

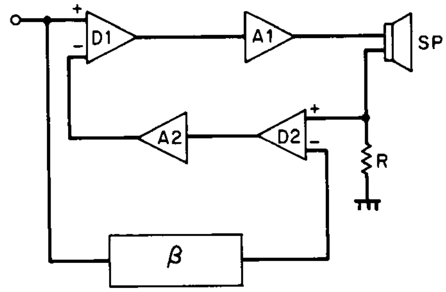


Figure 5.3.10 System for reducing current distortion. *D*1, *D*2 = differential amplifiers; *A*1, *A*2 = amplifiers; β = feedback circuit; *SP* = loudspeaker; *R* = resistor for detecting current distortion.

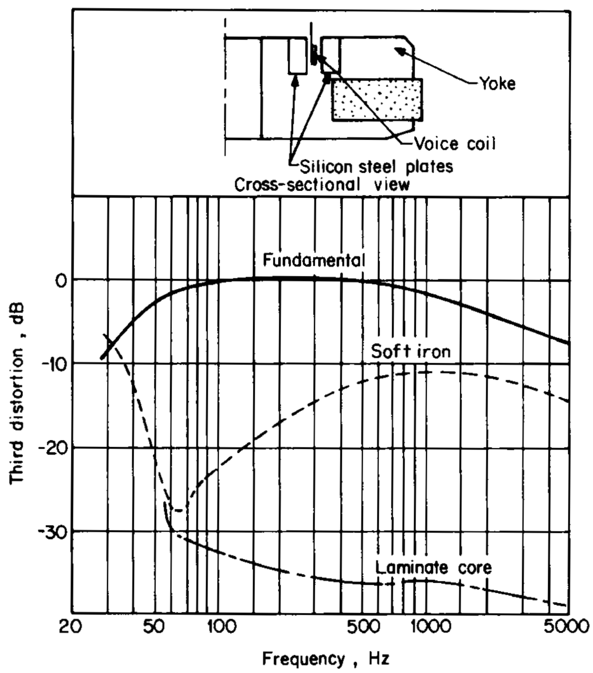


Figure 5.3.11 Comparison of the third-harmonic distortion between soft iron and silicon plates (solid line = fundamental current level, dashed line = soft-iron-type yoke, dash-dot line = laminate-core-type yoke).

of a general support system are shown in Figure 5.3.12. The function showing such a curve is expressed by

$$F(\xi) = \alpha\xi + \beta\xi^2 + \Upsilon\xi^3 \tag{5.3.9}$$

Where:
$F(\zeta)$ =force at displacement V, N

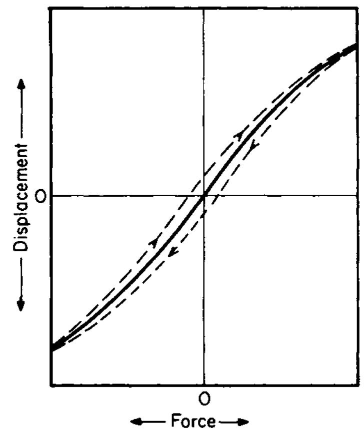


Figure 5.3.12 Relation between force and displacement in a typical suspension.

ζ = displacement, m
α, β, γ = constants

Consequently, stiffness ζ is found by

$$s(\xi) = \alpha + \beta\xi + \Upsilon\xi^2 \tag{5.3.10}$$

Assuming ω to be an ultralow frequency with this stiffness function substituted for the basic equation, Equations (5.3.7) and (5.3.8) are as follows.

$$M_{MA}\frac{d^2\xi}{dt^2} = m\frac{d^2\xi}{dt^2} + r\frac{d\xi}{dt} + s_n(\xi)\xi - A(\xi)I(\xi) \tag{5.3.11}$$

$$E_0 \sin \omega t = RI(\xi) + A(\xi)\frac{d\xi}{dt} \tag{5.3.12}$$

There are several methods of solving this equation. The calculation results on the assumption that the current is constant, using the indefinite-coefficient method and sample measurements, are shown in Figure 5.3.13. The point to be considered in the support system in particular is that, for a large amplitude, suspension elasticity is suddenly lost, forming a cropped wave and leading to rupture. Because the support system is nonlinear, not only does distortion occur, but a so-called jumping phenomenon is found. As shown in Figure 5.3.14, amplitude suddenly changes discontinuously for frequency and current. To prevent this, as large a suspension as possible is used, and such materials and construction are selected that the center-holding capacity is not lowered. Cone suspension uses a soft material wherever applicable, and corrugation and ribbing are provided to avoid edge resonance.

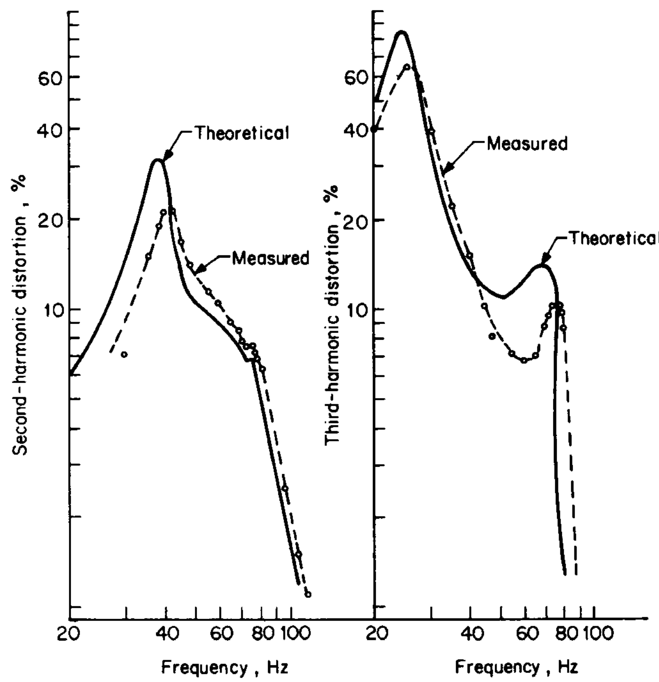


Figure 5.3.13 Distortion characteristics of a driving force, calculated from Equations (5.3.11) and (5.3.12).

Air Distortion

Generally, on the assumption that changes in volume are very small when the sound-surge equation is solved, the secondary or more terms are ignored. However, the smallest distortion cannot be ignored, and the high-order term cannot be ignored when sound pressure is large. Equation (5.3.12) shows the degree of second-harmonic wave due to nonlinearity on this high-order term:

$$p_2 = \frac{(\Upsilon + 1)\omega}{2\sqrt{2\Upsilon p_0}c} p_r^2 r \tag{5.3.12}$$

Where:

p_2 = second-harmonic distorted sound pressure generated by plane waves at distance r, N/m^2
p_r = fundamental wave sound pressure of plane wave at distance r, N/m^2
p_0 = atmospheric pressure, N/m
γ = ratio between constant-pressure specific heat and constant-volume specific heat (air, 1:4)
ω = angular frequency, rad
c = sound velocity, m/s
r = distance, m

The calculation results of Equation (5.3.12) are shown in Figure 5.3.15.

(a)

(b)

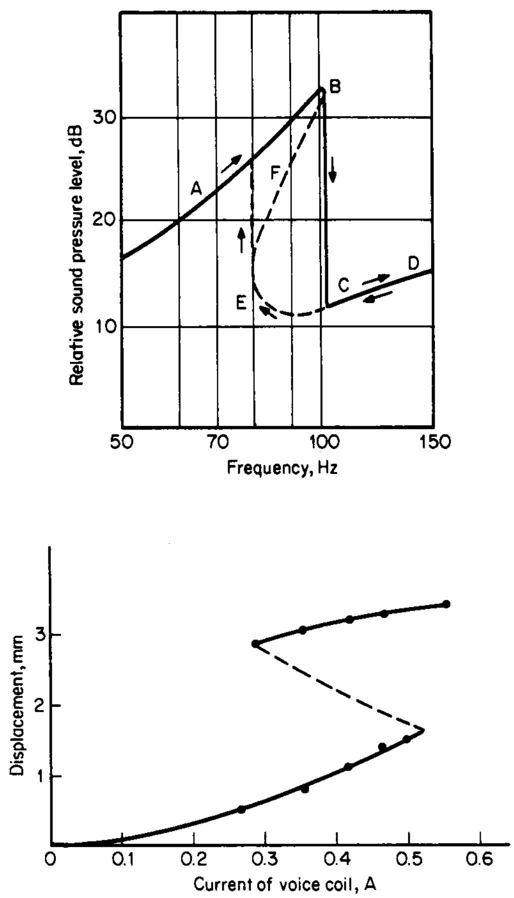


Figure 5.3.14 Nonlinear suspension system: (*a*) the unstable portion of the response frequency characteristic, indicated by the dashed line; (*b*) the unstable portion of the response current characteristic, indicated by the dashed line.

Frequency-Modulation Distortion

Signals input to a speaker have various frequency spectra. When low- and high-frequency sounds are radiated from a diaphragm at the same time, high-frequency sound is subjected to modulation because the diaphragm is moving forward and backward significantly according to low-frequency signals. The frequency-modulated wave generated thereby is expressed by a carrier and an unlimited number of sideband waves. The mean-square value of the ratio of sideband-wave energy to all energies of the sound wave is expressed in percentage as follows:

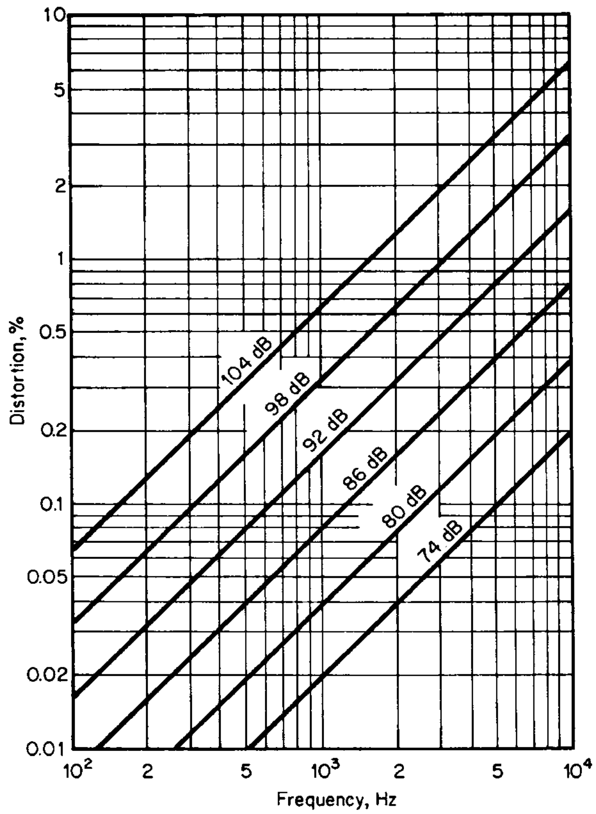


Figure 5.3.15 The distortion generated in the air gap between the cone and the listening-point distance in a direct-radiator speaker with a cone diameter of 20 cm, measured at a distance of 3 m.

$$D = 2900 \frac{f_2 \sqrt{p_1}}{f_1^2 d^2} \tag{5.3.13}$$

Where:
D = distortion, percent
f_2 = modulated wave (high-frequency), Hz
f_1 = modulated wave (low-frequency), Hz
p_1 = acoustic output of f_1, W
d = cone diameter, m

One of the methods for reducing this distortion is to use a multiway speaker system.

5.3.2d Diaphragm and Support System

It is no exaggeration to say that the diaphragm and support system nearly determine speaker acoustic characteristics. A typical shape and features for the diaphragm and support system are described below.

Diaphragm

Diaphragms are classified by shape into cone, plane, and dome diaphragms. The cone diaphragm is one of the most frequently used types. Figure 5.3.16 shows some typical shapes. Any of these types is directed to widening the piston-motion area to enhance a high-range playback limit frequency and also to reduce distortion. For this purpose, it is important to know the vibrating conditions of the cone diaphragm, but it is very difficult to find them analytically.

In the *dome diaphragm*, a thin metallic foil, resin-impregnated cloth, or paper is formed into a sphere, and the periphery of the diagram is driven. A diaphragm with a smaller-aperture diameter is easy to realize because of circumferential drive, split vibration can be controlled up to a high frequency, and favorable directional characteristics are also obtained. Materials used in this diaphragm include the following:

- Sulfite cellulose
- Sulfate pulp
- Paper mixed with highly elastic fiber such as silicon carbide whiskers, carbon fiber, and alamido fiber
- Metal foil such as aluminum, titanium, and beryllium
- High-polymer film such as polyethylene telephthalate or highly elastic materials reinforced by deposition such as carbon, boron, and beryllium
- Composite materials using honeycomb and foamed urethane as a core

Support System

The support system is divided broadly into a cone suspension system and a center holder. The cone suspension system is required to absorb reflection from the frame as a termination of the diaphragm to control edge resonance and also to prevent an acoustic short circuit which would occur before and after the diaphragm along with a baffle board. This system must be constructed so that it is easy to move in the vibrating-axis direction of the diaphragm and difficult to move in the lateral direction along with the center holder. The principal construction features of the cone suspension system are shown in Figure 5.3.17. Materials having proper mechanical resistance are preferable.

Requirements for centering suspension include the following:

- Provide proper stiffness in order to maintain a restoration force
- Hold a voice coil in the center of the gap formed by the magnetic circuit in order to smooth movement in the axial direction
- Maintain favorable linearity of driving-force-to-displacement characteristics even when the diaphragm is given large amplitude

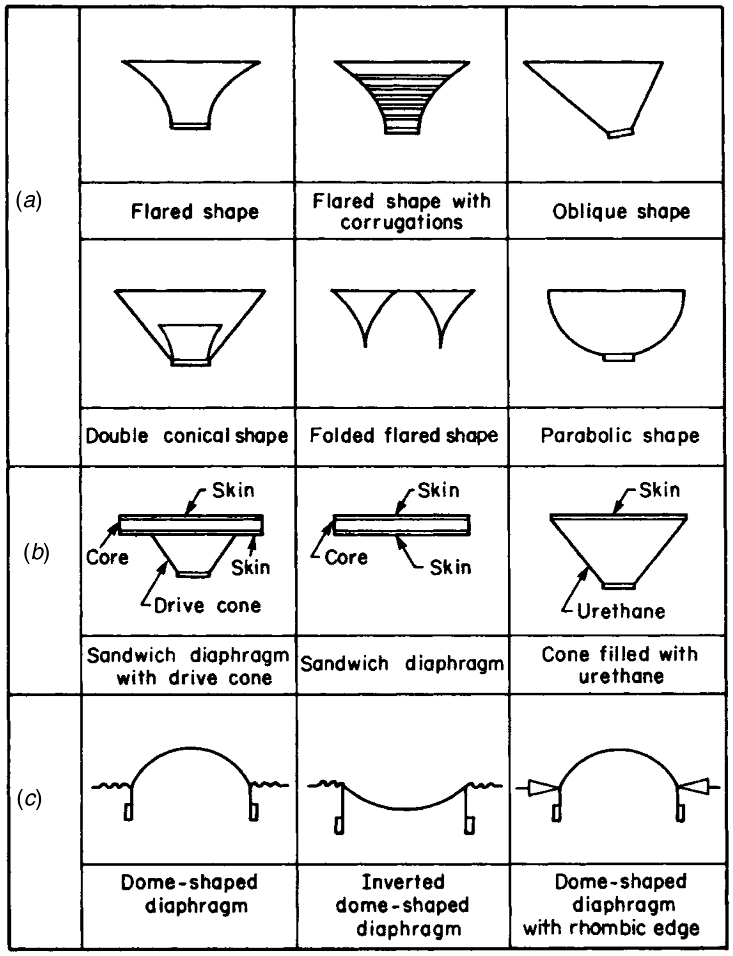


Figure 5.3.16 Sectional views of various diaphragm shapes: (*a*) diaphragm of the cone type extends to the reproducing band by changing the shape of the curved surface, (*b*) diaphragm of the plane type removes the cavity effect by using a flat radiation surface, (*c*) diaphragm of the dome type improves bending elasticity by forming thin plates into a domelike shape.

- Provide a light-weight assembly

As shown in Figure 5.3.18, most shapes of the support system are corrugated.

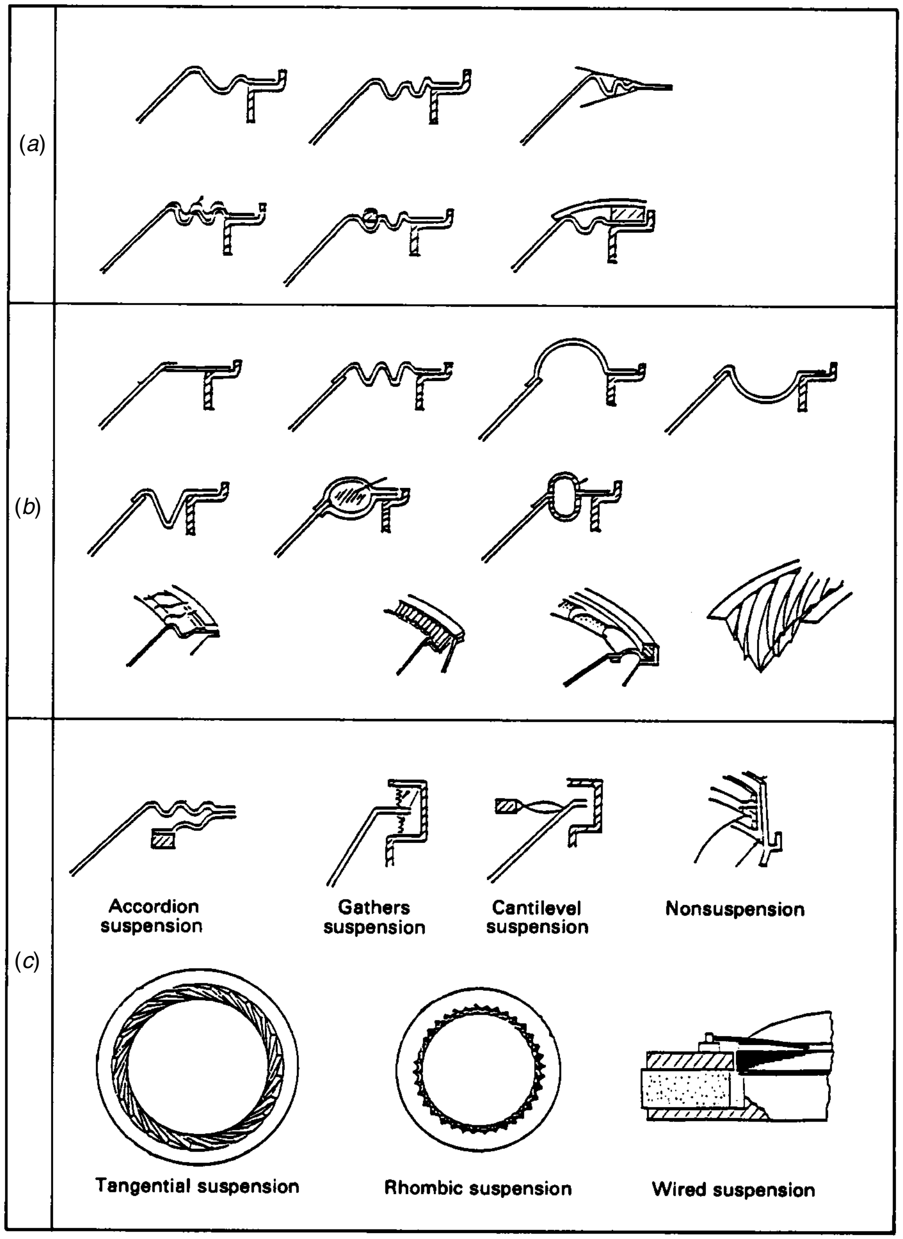


Figure 5.3.17 Sectional views of cone suspension systems: (*a*) the thinned edge of a diaphragm fulfills the function of the cone suspension, (*b*) material different from that of a diaphragm is used to fulfill the function of cone suspension, (*c*) exceptional cone suspensions.

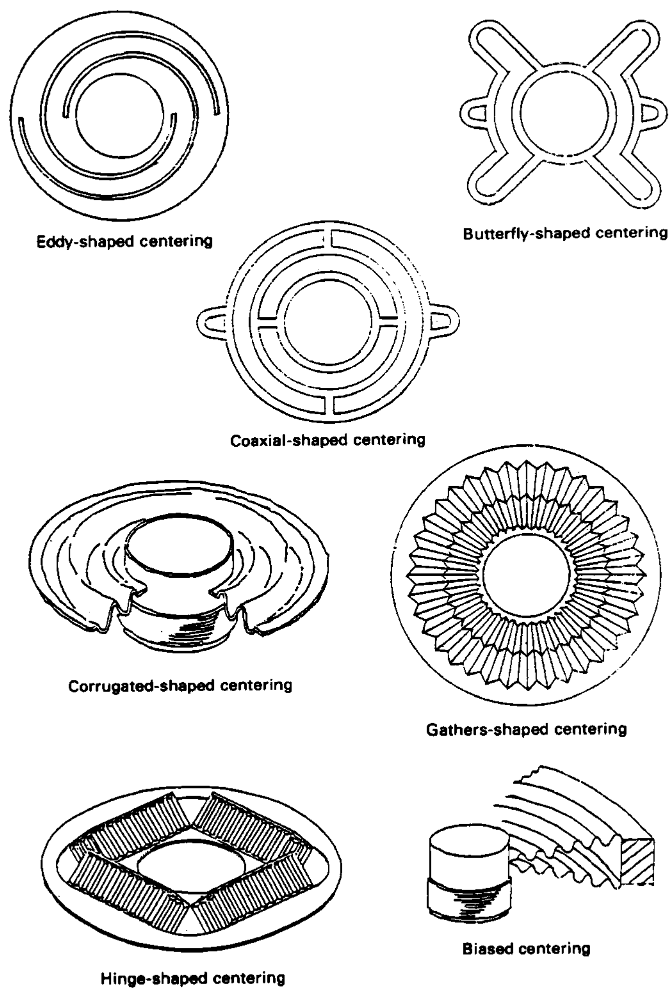


Figure 5.3.18 Various shapes of centering systems.

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Chapter

5.4

Loudspeaker Systems

Katsuaki Satoh

5.4.1 Introduction

Loudspeaker radiation efficiency lessens considerably at low frequencies with a simple plane baffle board. A baffle of 1 m or more per side is necessary for reproduction to nearly 100 Hz. As a means of increasing the baffle effect without an increase in size, a cabinet with an open rear can be used.

5.4.2 Baffle Design

Figure 5.4.1 illustrates an open-rear loudspeaker cabinet. Sound waves radiated from the rear of the speaker are of opposite phase from those radiated from the front and travel a longer distance around the sides of the cabinet to the reception point *P*. Consequently, the sound-pressure-frequency-response characteristics at the reception point vary in amplitude with frequency because of the phase difference between the front and back waves resulting from the path difference, $l + l_0$. The amplitude is at a maximum at odd multiples of $\lambda/2$ and a minimum at even multiples when cancellation occurs.

Therefore, the practical low-frequency response limit f_L is lower (generally considered to be one-half) than that corresponding to the wavelength $\lambda/2$. This frequency is found by

$$f_L = \frac{C}{4l} \text{ Hz} \tag{5.4.1}$$

where C = velocity of sound in air.

With this type of cabinet a standing wave occurs at a frequency corresponding to $\lambda/2$ in interval length between the upper and lower surfaces of the interior cabinet and both surfaces. One end of the cabinet functions as an open pipe, and a standing wave occurs at a frequency whose length corresponds to $\lambda/4$. To prevent this standing wave, a sound-absorbing material layer must be provided at the internal side of the cabinet. This type of cabinet is used for television and radio sets but seldom for high-fidelity systems.

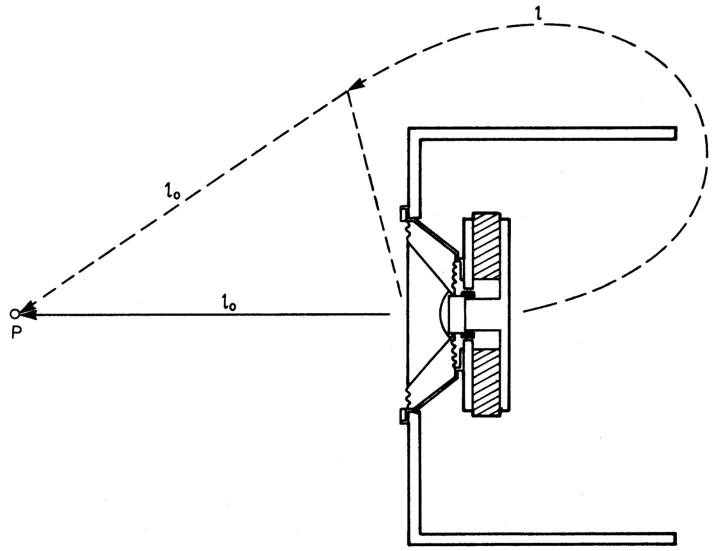


Figure 5.4.1 Sectional view of an open-rear cabinet. *P* = sound reception point; l_0 = distance between *P* and front surface of speaker; *l* = differential difference.

5.4.3 Enclosed Cabinet

With the enclosed type of open-rear cabinet, the rear side is closed to eliminate the influence of sound from the rear of the speaker and to prevent lowering radiation efficiency in low-frequency ranges (Figure 5.4.2*a*). However, because the rear is enclosed, the enclosed space comes to present air stiffness, and the minimum resonance frequency of the speaker system is determined according to the volume.

Figure 5.4.2*b* shows the equivalent circuit of this type of mechanical system. Stiffness S_{MB} in the space at the rear cabinet is

$$S_{MB} = \frac{p_0 C^2 S_d^2}{V_B} \tag{5.4.2}$$

Where:
p_0 = air density, kg/m^3
C = sound propagation speed in air, m/s
S_d = speaker diaphragm area, m^2
V_B = cabinet volume, m^3

The lowest resonance frequency f_{0B} of the speaker system is determined by

(*a*)

(*b*)

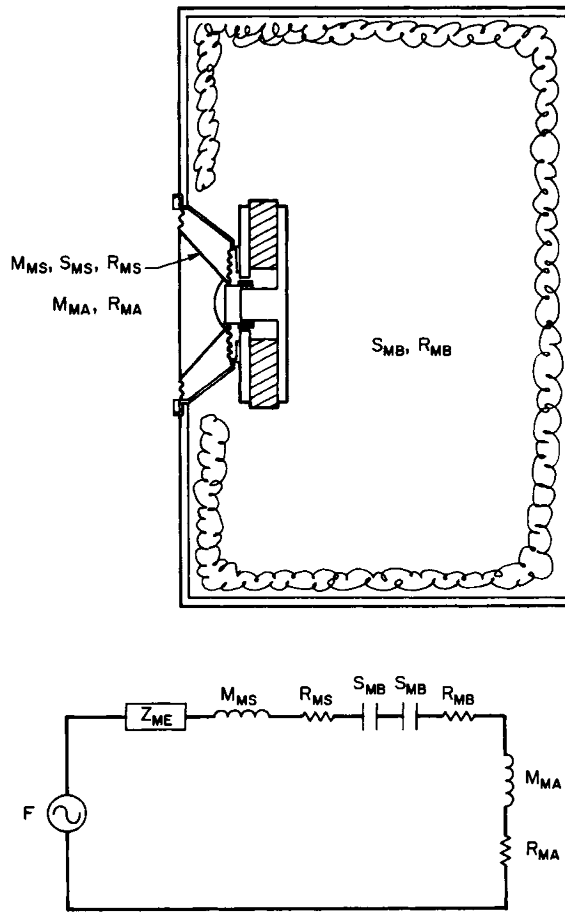


Figure 5.4.2 Enclosed cabinet: (*a*) sectional view, (*b*) equivalent circuit of the enclosed cabinet. Variables: F = driving force, N; Z_{ME} = motional impedance, mechanical ohms; M_{MS} = mass of vibrating system, kg; R_{MS} = mechanical resistance of vibrating system, mechanical ohms; S_{MS} = stiffness of vibrating system, N/m; S_{MB} = stiffness of cabinet, N/m; R_{MB} = mechanical resistance of cabinet, mechanical ohms; M_{MA} = radiation mass, kg; R_{MA} = radiation resistance, mechanical ohms.

$$f_{0B} = \frac{1}{2\pi}\sqrt{\frac{S_{MS}+S_{MB}}{M_{MS}+M_{MA}}} \tag{5.4.3}$$

Where:
S_{MS} = vibrating-system stiffness of speaker unit, N/m
M_{MS} = vibrating-system mass of speaker unit, kg

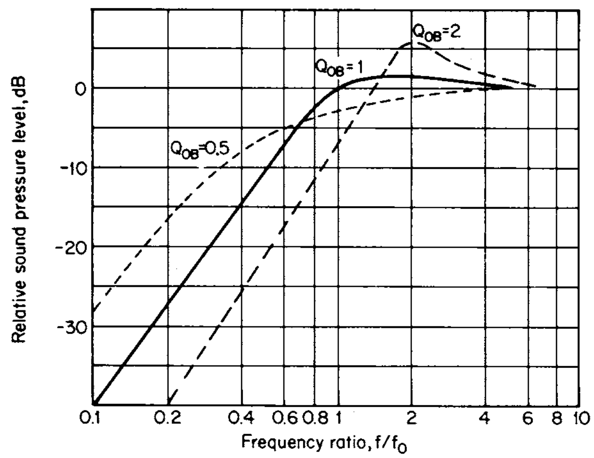


Figure 5.4.3 Relation between cabinet volume and frequency response.

M_{MA} = air-radiation mass of diaphragm, kg

As can be seen from Equations (5.4.2) and (5.4.3), when the cabinet volume is small and the diaphragm area is larger than the cabinet volume, the cabinet stiffness is larger than the vibrating-system stiffness of the speaker system ($S_{MB} \gg S_{MS}$). The low-range playback boundary is determined by the stiffness presented by the cabinet even when the minimum resonance frequency of the speaker unit is low, and low-range playback becomes difficult. Figure 5.4.3 shows the relationship of cabinet volume to sound pressure characteristics: when the volume becomes small, the lowest resonance frequency f_{0B} of the speaker system and the sharpness Q_0 of the resonance increase. When sound pressure P_r at a point $r(m)$ away from the cabinet axis, sharpness Q_0, and cabinet volume V_B are determined, and the lowest resonance frequency of the speaker unit is very low, and $S_{MB} \gg S_{MS}$ conditions are satisfied, the lowest resonance frequency f_{0B} of the cabinet is

$$f_{0B} = \frac{1}{2\pi}\sqrt{\frac{4C^2\pi^2r^2P_r^2Q_0}{p_0W_EV_B}} \tag{5.4.4}$$

W_E is an electric input (power) supplied to the speaker unit from the outside, which is applied as a principle. For the enclosed cabinet, to eliminate the influence of the reflection of sound waves by diffraction and to prevent irregularities in sound-pressure-frequency characteristics, it is necessary to make the installation position of the speaker asymmetrical and the cabinet corner round (Figure 5.4.4). A standing wave occurs at a frequency corresponding to $\lambda/2$ of the length between opposing surfaces within the cabinet, and it affects the vibration of the speaker-unit diaphragm. This is the same situation as with the open-rear

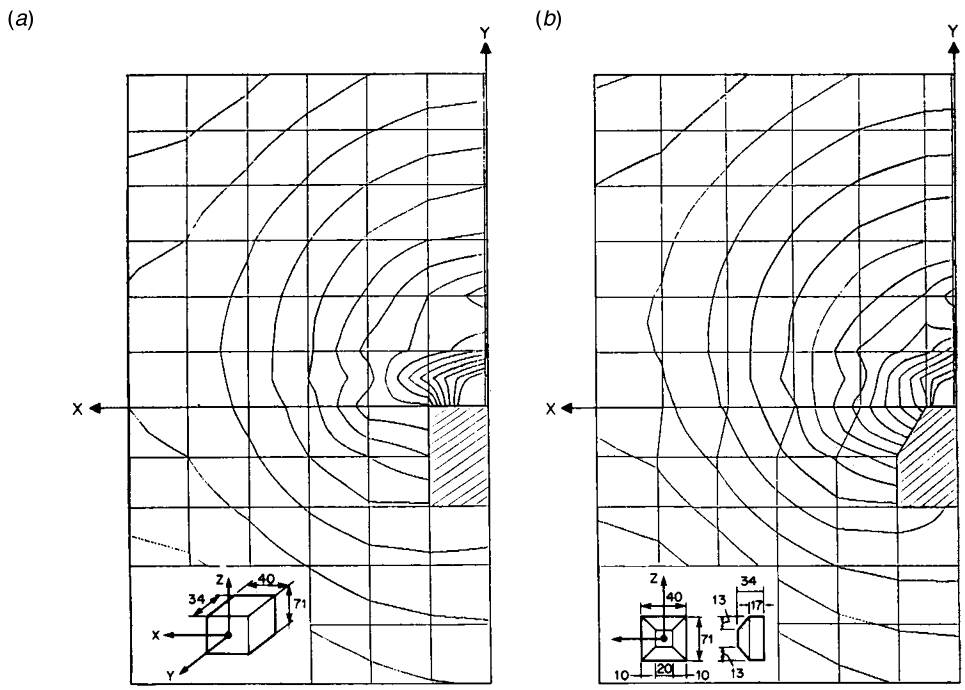


Figure 5.4.4 Influence of diffraction from a 2-cm diameter loudspeaker diaphragm at the corner of a cabinet shown as *x*-*y* contours of sound pressure levels: (*a*) rectangular cabinet; (*b*) tapered cabinet, estimated by FEM.

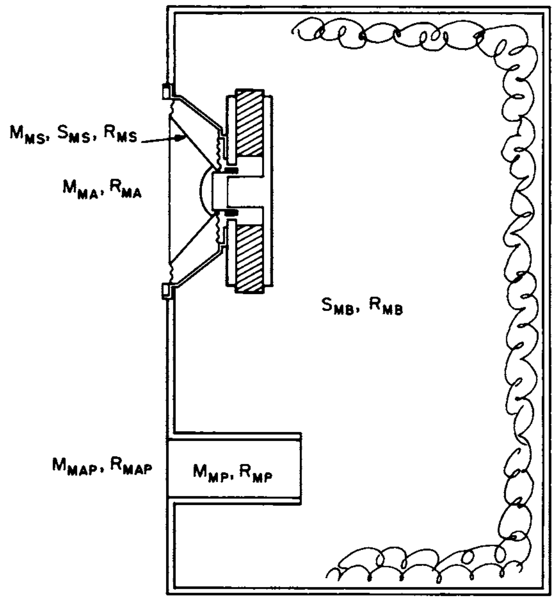
cabinet. Consequently, to prevent this standing wave a sound-absorbing material must be provided at the internal side.

5.4.3a Phase-Inverting Cabinet

This type of cabinet inverts the phase of sound radiated from the rear of the speaker in order to widen a low-range playback boundary. Figures 5.4.5*a* and 5.4.5*b* show the configuration and the equivalent circuit of the mechanical system. This cabinet is provided with a tube called a *port* that opens along with the speaker opening in front of the enclosed cabinet. The vibration of the diaphragm causes volume changes in the air within the cabinet, thereby radiating sound from the port. It produces a double sound source from which sound with the opposite phase is radiated at a very low frequency similarly to an open-rear type.

When the optimum constants of the speaker unit are used and the cabinet including the port have been properly selected, a low-range playback band can be made nearly 30 percent wider than the minimum resonance frequency of an enclosed cabinet with the same volume. However,

(*a*)



(*b*)

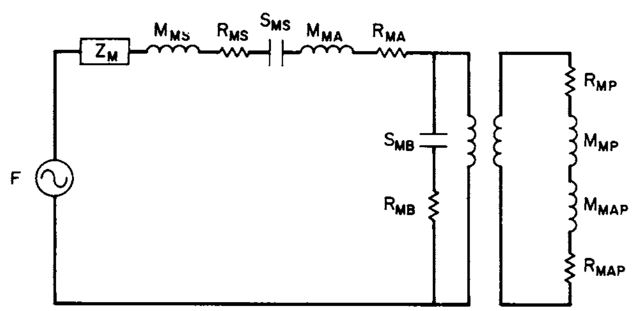


Figure 5.4.5 Acoustical phase-inverting type of cabinet: (*a*) sectional view, (*b*) equivalent circuit.

this resonance makes it impossible to invert the phase at a low frequency, and attenuation in sound pressure characteristics is noticeable (Figure 5.4.6).

Because the amplitude of the diaphragm becomes very small near the counter-resonance frequency of the cabinet, harmonic distortion is lowered, while the amplitude becomes large at a lower frequency, thus increasing harmonic distortion. This is one of the drawbacks of this type of cabinet. When sound pressure P_r at a point $r(m)$ away from the cabinet axis, sharpness Q_0, and cabinet volume have been determined, sound pressure P_r and the lowest resonance frequency f_{0B} of the cabinet are found.

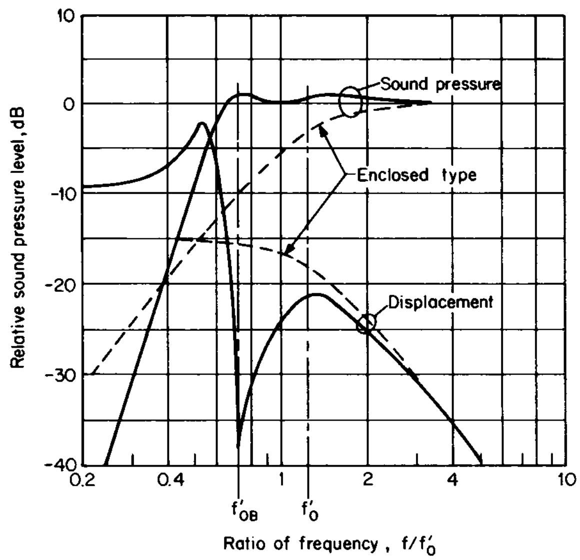


Figure 5.4.6 Comparison of the sound pressure level and displacement versus frequency for the acoustical phase-inverter (solid lines) and enclosed (dashed lines) types of cabinets.

$$f''_0 = \frac{1}{2\pi}\sqrt{\frac{S_{MS}}{M_{MS}}} \tag{5.4.5}$$

$$Q''_0 = \frac{2\pi f''_o\ M_{MS}}{R_{ME}+R_{MS}} \tag{5.4.6}$$

$$f'_{0B} = \frac{1}{2\pi}\sqrt{\frac{S_{MB}}{S_{MP}}} \tag{5.4.7}$$

$$\alpha = \frac{M_{MP}}{M_{MS}} \qquad \beta = \frac{S_{MB}}{S_{MS}} \tag{5.4.8}$$

$$A = \frac{1}{Q''_0}\left(1-\frac{\beta}{\alpha}\right) \quad B = \left(\frac{f}{f''_0}-\frac{f''_0}{f}\right)\left(1-\frac{\beta}{\alpha}\right)-\beta\frac{f''_0}{f} \tag{5.4.9}$$

$$\frac{f'_{0B}}{f''_0} = \sqrt{\frac{\beta}{\alpha}} \qquad \chi = \frac{f}{f''_0} \tag{5.4.10}$$

$$|p_r| = \frac{p_0 a^2 F}{2rM_{MS}} \times \frac{\chi}{\sqrt{\frac{1}{Q_0''^2}\left\{1-\left(\frac{\beta}{\alpha}\right)\frac{1}{\chi^2}\right\}^2 + \left[\left(\chi-\frac{1}{\chi}\right)\left\{1-\left(\frac{\beta}{\alpha}\right)\frac{1}{\chi^2}\right\}-\frac{\beta}{\chi}\right]^2}} \tag{5.4.11}$$

$$f_{0B} = \frac{1}{2\pi}\sqrt[3]{\frac{4C^2\pi^2 r^2 Pr^2 Q_0}{\sqrt{2}p_0 W_E V_B}} \tag{5.4.12}$$

The optimum conditions for the above speaker unit and cabinet constants are

$$S_{MB} = 0.5 S_{MS}$$

$$M_{MS} = \left(\frac{S_d}{S_p}\right)^2 M_{MP}$$

$$Q'_0 = \frac{1}{\sqrt{3}}$$

$$\frac{f'_{0B}}{f'_0} = \frac{1}{\sqrt{2}}$$

f'_{0B} and f'_0 are defined by

$$f'_{OB} = \frac{1}{2\pi}\sqrt{\frac{S_{MB}}{\left(\frac{S_d}{S_p}\right)^2 M_{MP}}} \tag{5.4.13}$$

$$f'_o = \frac{1}{2\pi}\sqrt{\frac{S_{MS}}{M_{MS}}} \tag{5.4.14}$$

Where:
S_{MB} = stiffness in cabinet rear space
S_{MS} = vibrating-system stiffness of speaker unit
M_{MS} = vibrating-system mass of speaker unit
M_{MP} = air mass by port
Q'_0 = resonance sharpness of speaker unit
S_d = speaker-unit diaphragm area
S_p = port-opening area

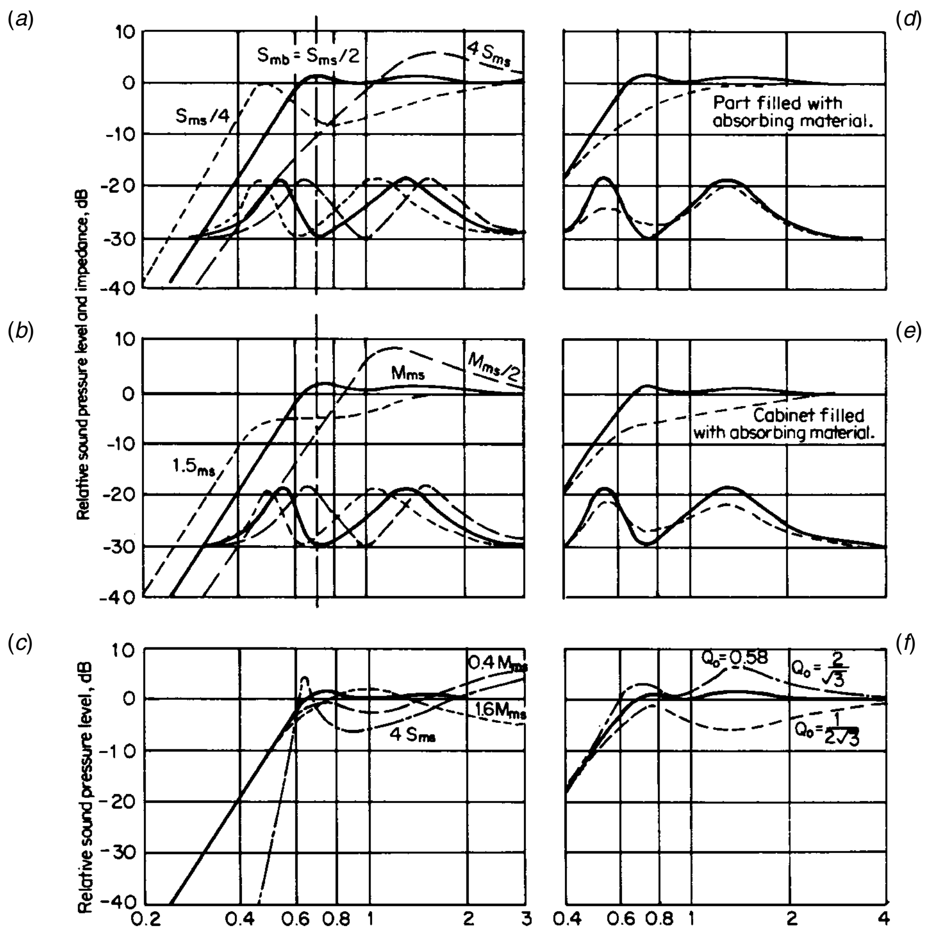


Figure 5.4.7 Pressure-response-frequency characteristic of the acoustical phase-inverter loudspeaker. The changes in the pressure-response-frequency characteristics are shown with: (*a*) cabinet volume, (*b*) length of port, (*c*) stiffness of loudspeaker mass, (*d*) absorbing material in the port, (*e*) absorbing material in the cabinet, (*f*) *Q* of the loudspeaker.

S_d/S_p = transformation ratio

In actual design, however, it is difficult to select these constants in optimum conditions, and it is common practice to use them slightly off the above conditions. Figure 5.4.7 shows changes in the characteristics under a variety of conditions. Consideration for the standing wave within the cabinet is the same as with the enclosed type. However, it is desirable to reduce sound-absorbing material to a minimum in order to prevent a standing wave and not increase resistance R_{MB} and port resistance P_{MP} within the cabinet (Figure 5.4.7*d* and *e*).

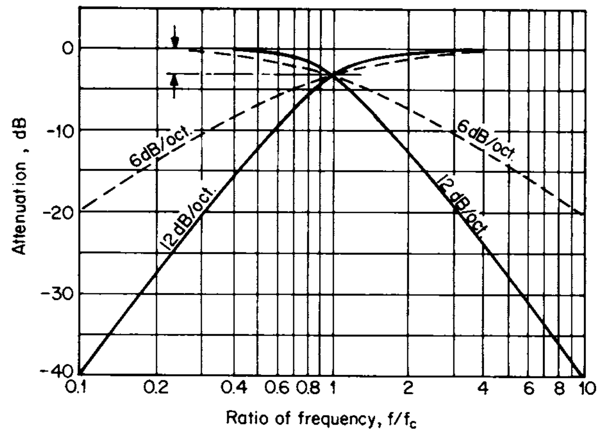
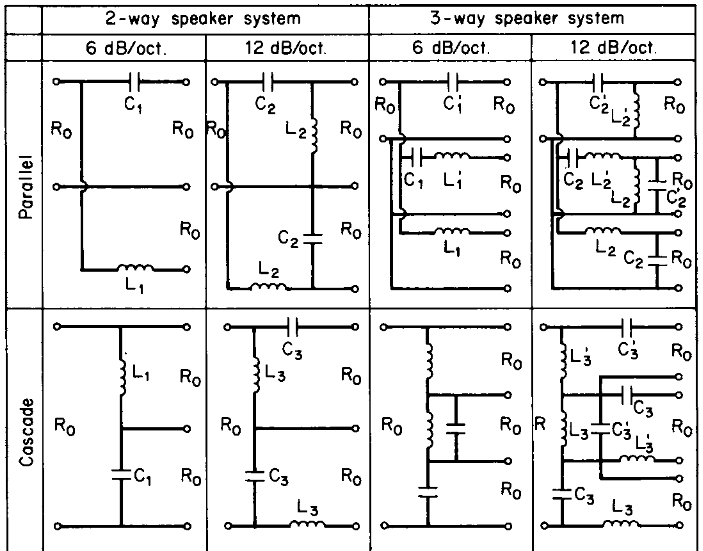


Figure 5.4.8 Attenuation-frequency characteristics of the dividing network.

5.4.3b Frequency Bands and Crossover Frequency

It is desirable to reproduce the full range of human audio frequencies—that is, 20 to 20,000 Hz—uniformly for the playback frequency band of the speaker system, but it is difficult for one speaker unit to reproduce such a wide range of frequencies. Therefore, it is common practice to adopt a system of dividing the playback band and using individual speakers for high, middle, and low ranges. The dividing network divides and applies electric input to each speaker. The frequency at the boundary for dividing the electric input is called the *crossover frequency.* The dividing network is normally configured by combination with high-pass and low-pass filters, divided into –6 dB per octave, –12 dB per octave, –18 dB per octave, and so on, depending on attenuation (Figures 5.4.8 and 5.4.9).

To determine crossover frequency and attenuation, it is necessary to consider various parameters including sound pressure characteristics, directional characteristics, and the harmonic distortion of the speaker used. Generally, the assigned lower frequency limit is less than the lowest resonance frequency of the speaker unit, and the amplitude of the vibrating system becomes large, thus increasing harmonic distortion, so that it should be more than 2 times this frequency. For the upper frequency limit, priority is given to directional characteristics, and in 30° characteristics this frequency is selected at a lower level than the –3-dB frequency below the axis. The speaker unit has available a wide variety of combinations such as a cone type for low frequencies and a horn type for middle and high frequencies. Therefore, efficiency is rarely constant. Consequently, the entire playback level must be adjusted to the speaker whose efficiency is the lowest of the system (in most cases, the speaker for lows). For this purpose, a volume-type attenuator and a constant-resistance-type attenuator are used (Figure 5.4.10, also the nomograph shown in Figure 5.4.11).



$C_1 = \frac{1}{2\pi f R_0}$, $L_1 = \frac{R_0}{2\pi f}$, $f_2 = \frac{1}{2\sqrt{2}\pi f R_0}$, $L_2 = \frac{R_0}{\sqrt{2}\pi f}$,

$C_3 = \frac{1}{\sqrt{2}\pi f R_0}$, $L_3 = \frac{R_0}{2\sqrt{2}\pi f}$

$C_1' = \frac{1}{2\pi f' R_0}$, $L_1' = \frac{R_0}{2\pi f'}$, $C_2' = \frac{1}{2\sqrt{2}\pi f' R_0}$, $L_2' = \frac{R_0}{\sqrt{2}\pi f'}$,

$C_3' = \frac{1}{\sqrt{2}\pi f' R_0}$, $L_3' = \frac{R_0}{2\sqrt{2}\pi f'}$

f = cut frequency of the woofer and the midrange

f' = cut frequency of the midrange and the tweeter

Figure 5.4.9 Dividing-network systems.

5.4.3c Cabinet-System Variations

A number of variations on the basic cabinet system design have been developed to address specific implementation issues and objectives. Some of the more common types are discussed in the following sections.

Drone-Cone Type

The *drone-cone*, a phase-inverting type, is structured so that only a unit of the diaphragm without the driving system is installed in the section corresponding to the bass-reflex port (Figure 5.4.12). In the schematic of the equivalent circuit of the mechanical system, shown in Figure 5.4.13, the mass of the diaphragm corresponds to the bass-reflex-port mass, and it forms another version to which the stiffness S_{Md} of the support system of the diaphragm is added. Therefore, this type has a disadvantage in that if bass characteristics are affected by the support-system stiffness S_{Md} and the support is too strong, thereby increasing stiffness, bass becomes harder to pro-

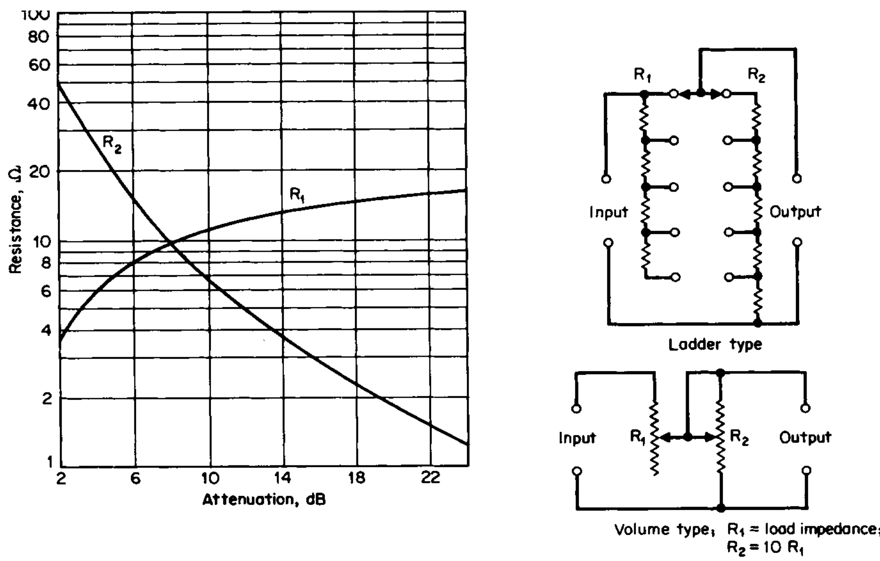


Figure 5.4.10 Constant-impedance attenuators.

duce than in a bass-reflex type. The basic operation is almost the same as with a bass-reflex system.

Horn-Loaded Type

The *horn-loaded* type includes a front-loaded system with a horn loaded in front of the speaker, a back-loaded system with a horn loaded at the back of speaker, and a combination system with the first two types combined. All of these types are designed to enhance transformation efficiency in low ranges by means of the horn load. However, to produce a horn effect in bass, the horn becomes longer so that it is customary to adopt a return-horn type partitioned within the cabinet, except for large systems intended for theater and concert public address (PA) applications (Figure 5.4.14).

Acoustic-Labyrinth Type

In the *acoustic-labyrinth* type, a partition panel is used within the cabinet to form an acoustic pipe (Figure 5.4.15). The fundamental principle is that when the pipe length is selected to produce a wavelength one-half of the bass frequency to be reproduced, the phase is the same as that of the sound wave of the front speaker unit at the opening, thereby increasing the level. To reproduce bass, as with a horn-loaded type, a very long acoustic pipe is required, which is not suitable for a small-size system. Because sound waves radiated from the opening are opposite in phase at less than the resonance frequency of acoustic pipe—as is the case with a phase-inverting type—attenuation of radiation efficiency becomes large.

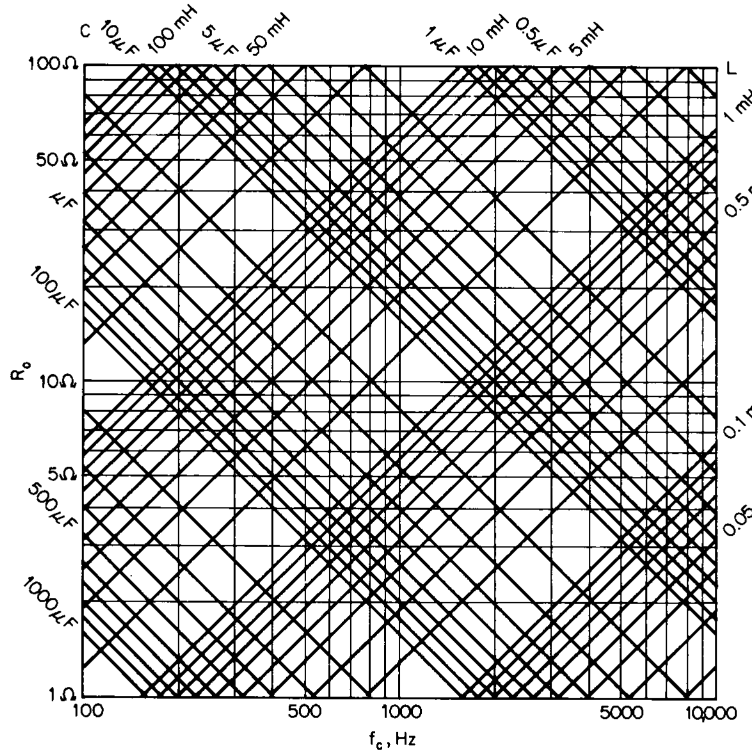


Figure 5.4.11 Constant-impedance nomograph.

Double-Port Type

In the *double-port* type, a vacant chamber of the bass-reflex type of cabinet is partitioned into two parts, and ports are provided (Figure 5.4.16). This type is designed to expand the low ranges with port resonance frequencies preset.

Acoustic-Coupler Type

In a small-size system, the lowest resonance frequency is determined by cabinet stiffness. To avoid this limitation, the speaker aperture diameter can be made small while the diaphragm area is reduced, thus lowering radiation efficiency. To overcome this problem, an *acoustic-coupler* type can be used. This technique employs a small speaker for drive, and a large-aperture-diameter diaphragm for radiation is provided in front of the speaker (Figure 5.4.17).

Double-Drive Type

The acoustic-coupler type incorporates a radiation unit consisting only of a diaphragm without the drive system. The *double-drive* type uses a large-aperture-diameter speaker in this part for drive with the same phase (Figure 5.4.18).

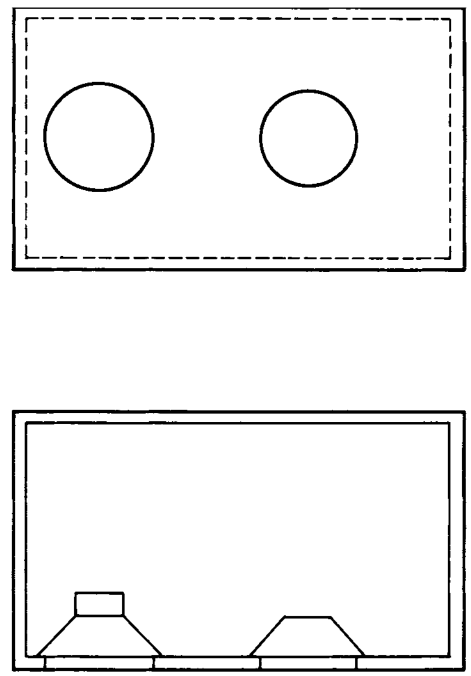


Figure 5.4.12 Sectional view of the drone-cone type of cabinet.

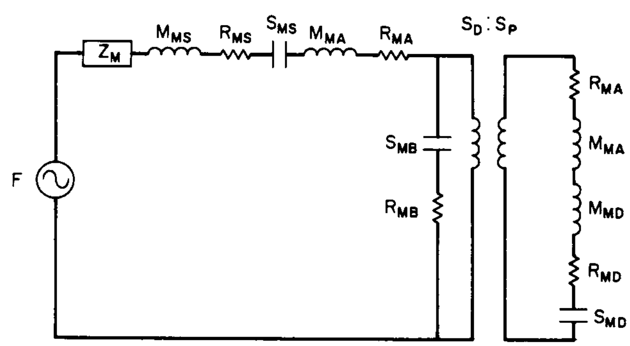


Figure 5.4.13 Equivalent circuit of the drone-cone type of cabinet. $\dot{F}$ = driving force, N; Z_{ME} = motional impedance, mechanical ohms; M_{MS} = mass of vibrating system, kg; R_{HS} = mechanical resistance of vibrating system, mechanical ohms; S_{MS} = stiffness of vibrating system, N/m; S_{MB} = stiffness of cabinet, N/m; R_{HB} = mechanical resistance of cabinet, mechanical ohms; S_D = area of diaphragm, m^2; S_P = area of drone, m^2; R_{MD} = mechanical resistance of drone, mechanical ohms; M_{MD} = mass of drone with the mutual mass, kg; R_{MA} = mechanical resistance of vibrating system, mechanical ohms.

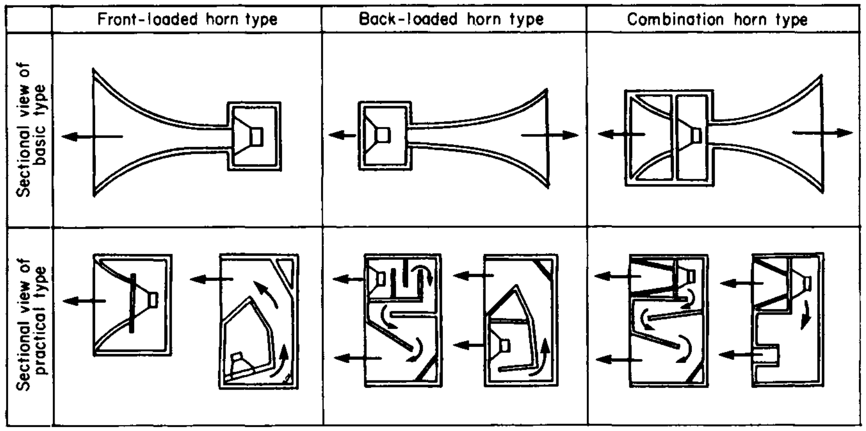


Figure 5.4.14 Sectional view of horn-loaded type.

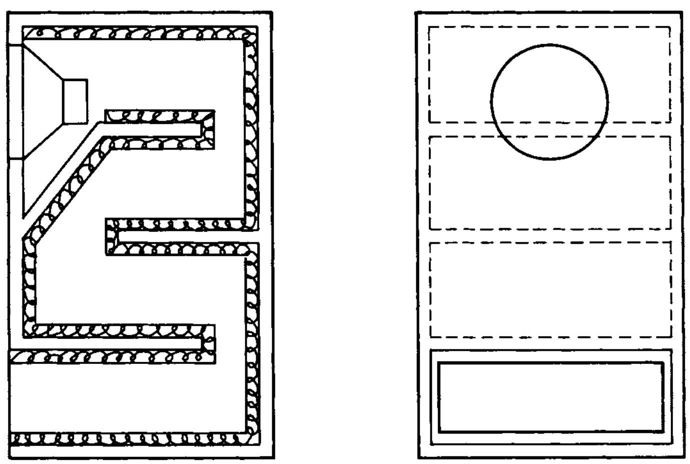


Figure 5.4.15 Sectional view of an acoustic-labyrinth type.

Other Types

Other examples are shown in Figure 5.4.19. They include the *Karlson* type, the *R-J cabinet* type, and the *acoustic-matrix* type.

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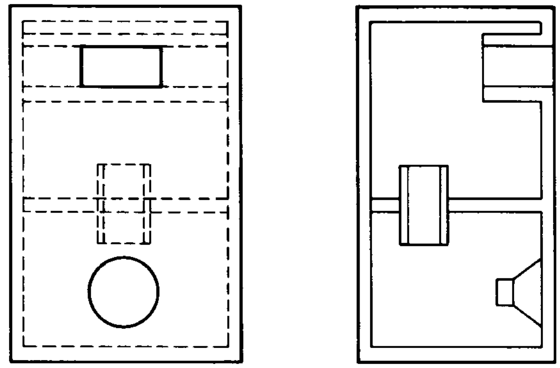

Figure 5.4.16 Sectional view of double-port type.

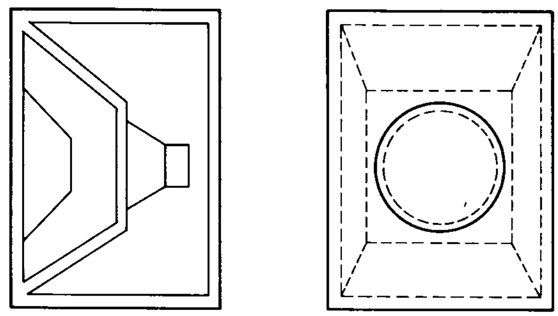

Figure 5.4.17 Sectional view of acoustic-coupler type.

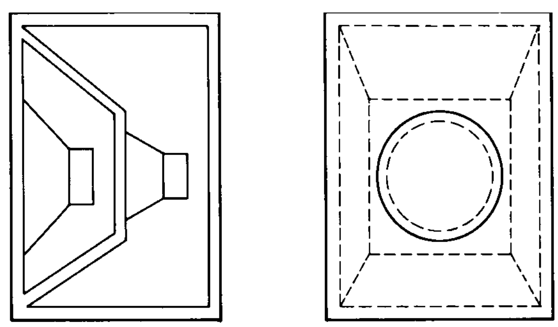

Figure 5.4.18 Sectional view of double-drive type.

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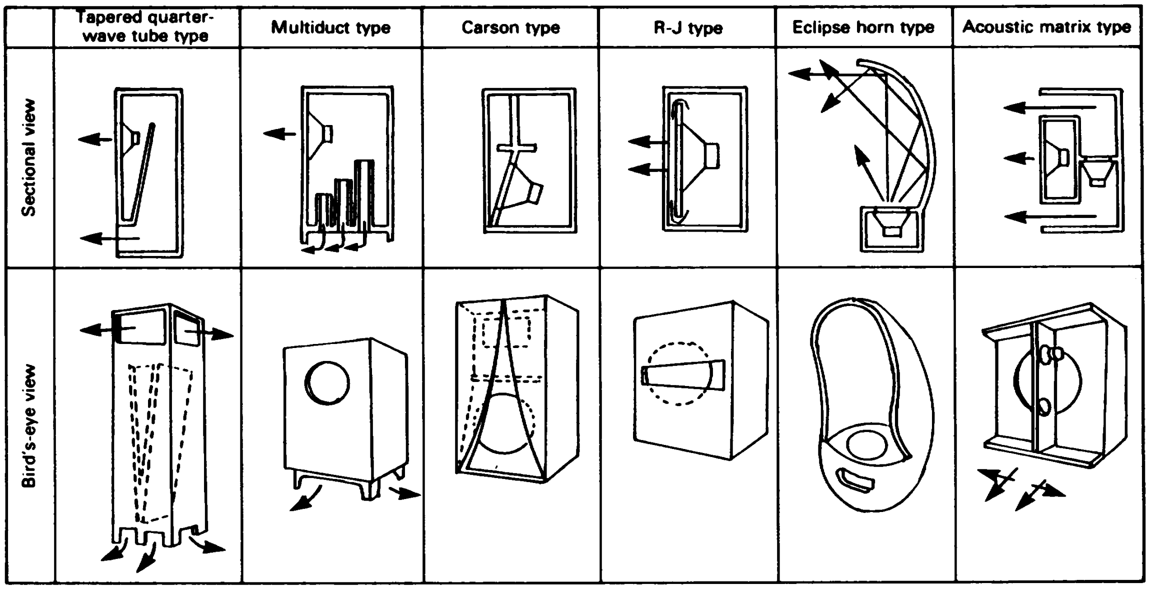

Figure 5.4.19 Other loudspeaker systems.

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Chapter

5.6

Public Address System Loudspeakers

Katsuaki Satoh

5.6.1 Introduction

The problems encountered with speaker systems intended for use in large areas, such as stadiums and concert halls, vary according to the size and type of facility. Generally speaking, however, potential problems include the following:

- Unequal sound pressure distribution in the service area
- Echo generation
- Areas of inferior articulation
- Acoustical feedback

If these problems can be predicted in advance, speaker arrangement and component selection are simplified, and overall performance is enhanced.

Two primary types of speaker placement have emerged for public address applications: central-cluster systems and distributed speaker systems.

5.6.2 Central-Cluster System and Distributed System

The central-cluster system, which uses high-efficiency midrange to high-range speakers with sharp vertical and horizontal directivity and high-power bass speakers, has the following advantages compared with the distributed system:

- Echoes seldom occur.
- The system can be used for various purposes, such as improving sound-directional localization, producing special effects, and so on.
- Material, construction, and maintenance costs are low.

On the other hand, there are drawbacks. Because the speakers are concentrated in one or two locations in this system, volume may be excessive near each speaker, thus increasing the level difference between the nearest and farthest listening positions.

Table 5.6.1 Comparison between Distributed and Central-Cluster Systems Employed in a Stadium

No.	Item	Central-cluster system	Distributor system
1	Speaker system	Multispeaker system employed relatively easily	Large speakers easily employed
2	Echo generation	Seldom	Often
3	Sound pressure level	Great level difference between near and distant positions	Relatively uniform
4	Workability	Labor for speaker house preparation, piping, and wiring saved	Speaker tower and pole installation, piping, and wiring executed separately
5	Speaker line transmission loss	Amplifier installed in or around speaker house, saving labor	Low-impedance drive impossible owing to long distance
6	Adjustment maintenance	Relatively easy owing to concentrated system	Sometimes difficult according to number of installation places and speaker structure
7	System price	Reasonable	More expensive

The distributed system functions satisfactorily for public address (PA) and announcing. However, since speakers are distributed over an area, an echo is liable to occur owing to the distance between speakers, and interference may be caused by reflected sound. In particular, when a full bass tone is required for music reproduction, a booming effect frequently will be encountered. When speakers are located around the rear seats, the sound is localized behind the audience. Also, when speakers are installed around a field to cover seats, the volume becomes too low in the field, and large speakers cannot be used because they obstruct the view of spectators. Accordingly, there is a limit to the sound quality and sound pressure distribution realizable by this system.

Table 5.6.1 compares the distributed and central-cluster systems. These advantages and drawbacks do not apply to all cases. Rather, they vary depending on the complexity and cost of the facility, the kinds of entertainments, methods of operation, and ambient environmental conditions. Therefore, the speaker-system location, direction, and extent must be determined to meet field conditions.

5.6.3 Theater and Auditorium Systems

Acoustically required items for a PA system vary according to application. However, general requirements may be roughly outlined and defined as follows:

- Volume is satisfactory: *volume*
- Sound can be heard clearly: *clearness*
- Sound is pleasant: *reverberation*

These requirements cannot be fulfilled solely by adjusting the PA system; technical architectural conditions should also be met.

Typical entertainment-related applications and their acoustic requirements are listed in Table 5.6.2. An electrical sound system for a hall is shown in Figure 5.6.1 and a sound control system is shown in Figure 5.6.2. An example loudspeaker system is illustrated in Figure 5.6.3.

The size of the loudspeaker system is determined by the sound-pressure-level and frequency characteristics required for the hall. The attenuation curve of the sound pressure level changes with interior conditions, which are quite different from free-field conditions. Therefore, the loudspeaker system is designed after the architectural and interior design has been completed. The average absorption a coefficient is given by

$$\bar{\alpha} = \frac{\sum(\alpha_i - S_i)}{\sum S_i} \tag{5.6.1}$$

Where:
S_i = individual boundary surface area, m^2
α_i = absorption coefficient of area S_i

The room constant R is shown by

$$R = \frac{\bar{\alpha} S}{1 - \bar{\alpha}} \tag{5.6.2}$$

where S = total surface area, m^2.

The attenuation equation is

$$SPL = PWL + 10\log\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) \tag{5.6.3}$$

Where:
SPL = sound pressure level at point from sound source, dB
PWL = power level of sound source, dB
Q = directivity index of sound source
r = distance from sound source, m
R = room constant

The first term of Equation (5.6.3), 10 log ($Q/4\pi\ r$), shows inverse-square-law attenuation. The second term, 10 log (4 / r), shows the reverberant component. The relationship of attenuation and distance from the source is shown in Figure 5.6.4. Examination of this figure makes the following points clear:

Table 5.6.2 Entertainment-Related Applications and Acoustic Requirements

Entertainment items	Acoustic requirements (overall evaluation standard)
Meetings, ceremonies, training courses, lectures, etc.	As speech is mainly transmitted, sound clearness is most important. As for electronics, howling margin should be increased to prevent howling. It is desirable that interiors be finished for sound absorption.
Popular music (broadcast via TV, etc.), popular songs, jazz	Ample reverberation and sufficient sound volume (sound pressure) are required. As for architecture, a finish to improve sound reflection is required. As for electronics, a large-capacity amplifier and a high-performance speaker system are required.
Traditional Japanese music, kabuki performances	As a rule, electrical acoustic equipment, if used, is employed only as an auxiliary to transmit speech and accompaniments. Since kotos and shamisens generate impact sound, an appropriate microphone is difficult to find. Interiors should be treated for sound absorption.
Opera, dance, plays	As a rule, electrical acoustic equipment is employed to produce special effects and not to transmit speech and songs. Accordingly, a comparatively large-capacity system is required. As for architecture, special precautions should be taken for ample reverberation.
Classical music	Electrical acoustic equipment is not used except for transmitting explanations and calling. As for architecture, special steps should be taken to secure a longer reverberation time.
Plays, folksongs	As in popular music, ample reverberation and sufficient volume (sound pressure) are required. Clearness is also required for plays.
Motion pictures	An electrical sound system for motion-picture acoustics is required. The power amplifier unit can be shared, and a special speaker system is required. Speakers for special effects may be installed on walls, ceilings, etc.

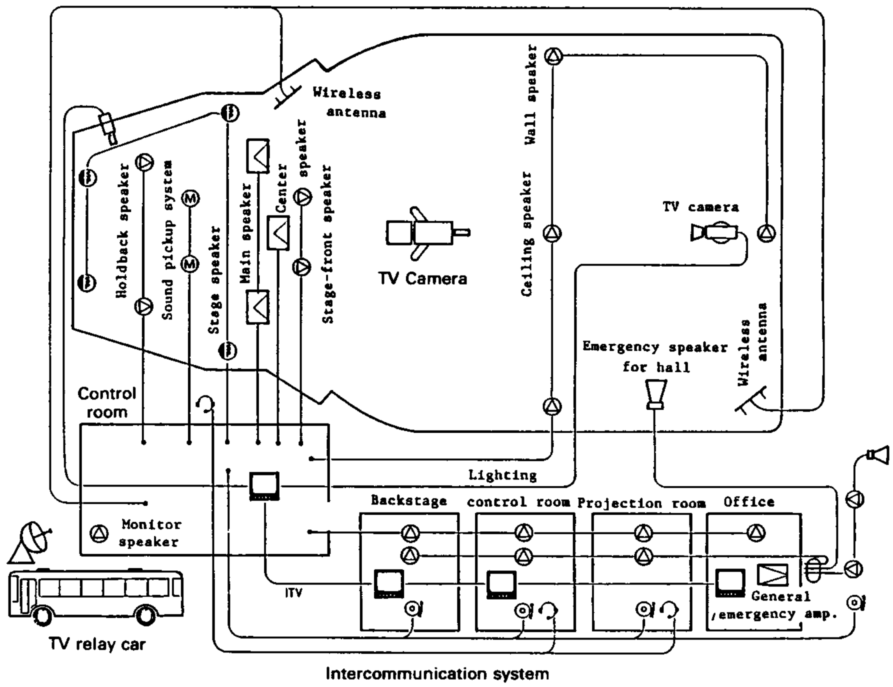


Figure 5.6.1 Electrical sound system for an auditorium.

- Sound pressure deviation is 5 dB between the front seats and the rear seats without a loudspeaker system.
- If a sound pressure level of 112 dB is generated at a point 1 m from the source, a sound pressure level of 90 dB can be obtained at the farthest seats.
- The power amplifier must apply 15.8 W to the loudspeaker, thus increasing the sound pressure level by 12 dB at the farthest seats.

The peaking factor of a speech source is 10 dB, and the peaking factor of a music source is 20 dB. Therefore, the power amplifier capacities are 158 W and 1580 W with the two sources, respectively.

5.6.3a PA System Using a Computer for a Multipurpose Sports Arena

Multipurpose sports arenas used for such events as music concerts, memorial ceremonies, lectures, exhibitions, and fashion shows require two or three types of stage setting according to use. When used for a sports contest or an exhibition, an arena will be utilized in a variety of ways. For example, an entire arena may be used as a stage. In this case, it may accommodate an audience of tens of thousands. Consequently, the speaker system will include center-cluster speakers (octahe-

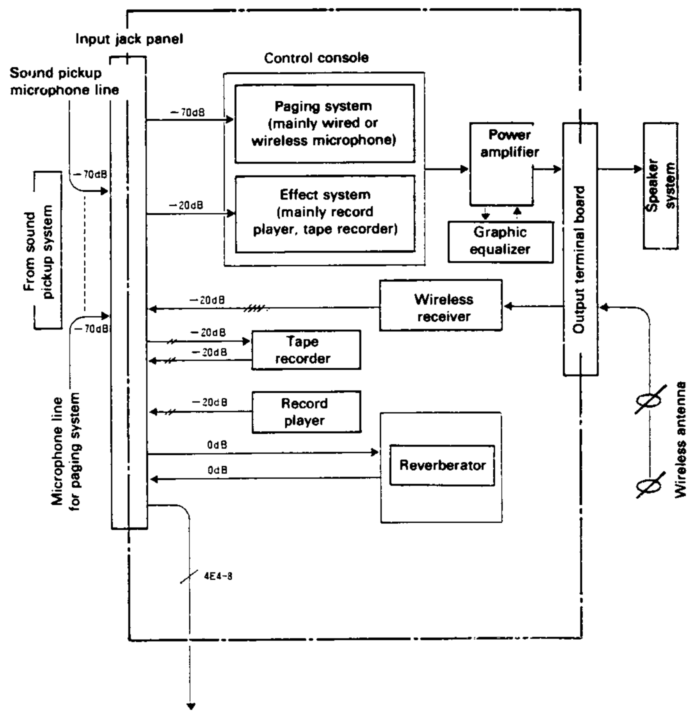


Figure 5.6.2 Sound control system.

dron assembly directed downward) and distributed units installed at the upper part of the seats. When a temporary stage is required for concerts, gatherings, and so on, speakers must be installed at both sides of the stage.

Adjustment of a PA system for such a multipurpose hall usually entails high labor costs. Speakers are selected according to the type of entertainment, delay time with distributed speakers must be adjusted, the sound field must be adjusted by using a graphic equalizer, the volume level among speakers must be balanced for two- or three-way speaker systems, etc. (see Table 5.6.3). Moreover, because a few hundred input-output lines are used, the patching operation associated with the audio control console is time-consuming. This problem is solved by computerized methods of connection and patching of input lines to the control console and of setting the speaker system according to the type of entertainment, as described and shown in Figures 5.6.4 and 5.6.5.

5.6.3b Stadium PA Systems

Stadiums are used for various purposes: baseball and football games, track meets, expositions, and open-air shows (see Table 5.6.5). Possible applications of PA systems include:

- Opening and closing ceremonies

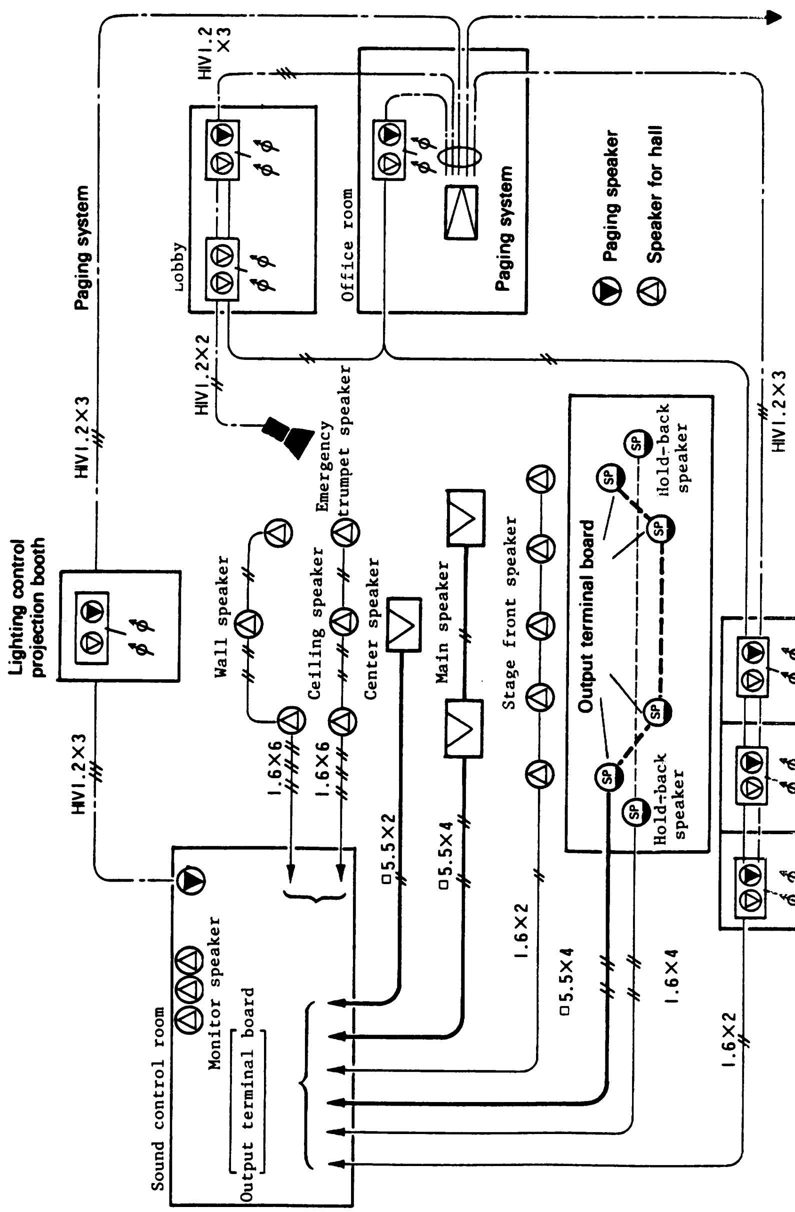


Figure 5.6.3 Example loudspeaker system.

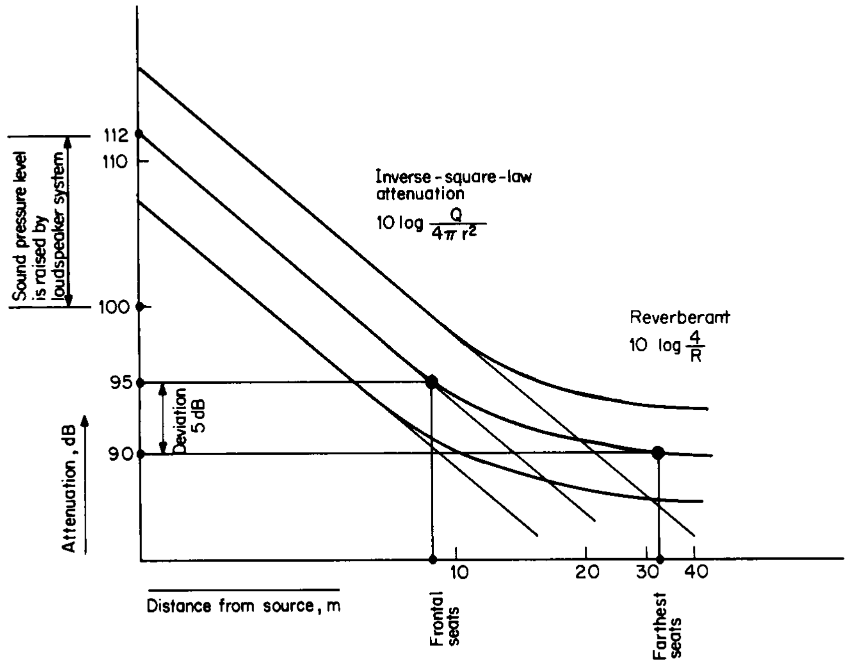


Figure 5.6.4 Sound pressure attenuation with increasing distance in an enclosed space.

- Background music (when contestants enter or leave the stadium)
- Musical accompaniment for vocal presentations
- Musical accompaniment for action
- Guidance by announcer
- Background music for exhibitions
- Transmission for radio and TV broadcasting
- Auxiliary use for concerts

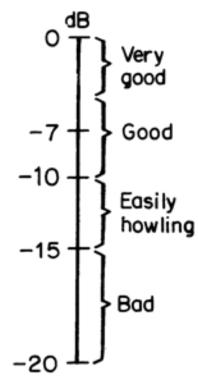
Speaker-system design requirements include the following:

- Speaker-unit performance and speaker-system scale, including the installation method, which secures the pressure required over the entire area between the main sound-source speaker and the farthest seat position
- Weather resistance and durability of the speaker unit (for low ranges in particular), as stadium facilities are normally built for outdoor use
- Directivity at midrange and high-range areas
- Sustained feedback (howling) threshold and frequency characteristic

Table 5.6.3 Electronics for Theater and Auditorium Systems

Technical conditions	Suggested measurement method	Suggested value
Maximum sound pressure level	1. Transmit pink noise with the electrical sound system and main speaker. 2. Measure sound pressure behind seats on the first floor, using the sound-level meter.	Continuously 95 dB or more.
Sound-pressure-distribution deviation within seats	1. Transmit pink noise with the electrical sound system, main speaker, etc. Measure sound pressure level at specified locations (10 or more) using a sound-level meter. 2. Determine deviation at each location against the center of seats on the first floor. 3. Measure sound pressure at the center at 80 to 85 dB. 4. For analysis at each frequency, measure at 125 and 500 Hz and 1 and 4 kHz.	±3 dB maximum.
Transmission-frequency characteristics	1. Transmit pink noise with the electrical sound system and main speaker, and adjust the sound pressure level at the center of seats on the first floor to 80 to 85 dB. 2. Record the change in sound pressure level for each frequency via one-third-octave filter, using a level recorder. 3. Make measurements at 5 to 10 locations inside seats.	The transmission-frequency characteristics should be uniform and have no irregularities. The multiway speaker should have no extreme irregularities in characteristics near the crossover frequency.
Howling-frequency characteristics	1. Set a microphone to be used to the position of center microphone on the stage, and insert one-third-octave filter and attenuator (ATT) into the volume-intensifying loop. 2. Adjust the ATT so as to exceed the howling-limit point for every frequency; then enter ATT reading on graph paper.	The frequency characteristics should be uniform over the entire range. (In particular, no peak should be allowed.) When a peak is found, adjust so as to eliminate the peak by using a graphic equalizer. It is desirable that the level be low over the entire range.

Table 5.6.4 Electronics for Theater and Auditorium Systems (continued)

Technical conditions	Suggested measurement method	Suggested value
Howling margin	1. Set a microphone to be used to the position of center microphone on the stage and install speaker 50 cm above the microphone axis as primary sound source. 2. Insert ATT into the volume-intensifying loop, and adjust frequency to 6 dB below the howling-limit point via ATT. 3. Transmit pink noise with the primary-source speaker, collect it with a microphone, and transmit with main speaker, increasing volume. 4. Measure the sound pressure level applied to the microphone by the primary-sound-source speaker, using a sound-level meter to obtain 80 dB. (*A*) 5. At that time, measure the sound pressure level at a representative seat location, using a sound-level meter. (*B*) 6. Determine the level difference (*B*) − (*A*) as the howling margin.	

Computers have been actively utilized for these purposes, enabling sound pressure distribution and level to be calculated in advance.

Speaker-Unit Performance and Sound Pressure Level

Assuming that the distance between the sound source and the farthest seat position is 300 m, that noise from the audience and other distractions is 85 dB, and that the signal-to-noise ratio which can produce intelligible sound is 5 dB, 90 dB (sound pressure level required at farthest seat position) + 50db (attenuation is 300–m free space) = 140 dB . Accordingly, to cover the farthest seat position a sound pressure of 140 dB is required at a position 1 m from the speaker. This sound pressure must be generated by applying sufficient power and employing an appropriate number of speakers.

Assuming that sound pressure rises by 10 dB by applying a power of 100 W and by 8 dB by employing six additional speakers, and that a system installed on both left and right sides can cover the farthest seat position,

$$140\text{ dB} - 20\text{ dB} - 8\text{ dB} - 8\text{ dB} = 104\text{ dB} \qquad (5.6.4)$$

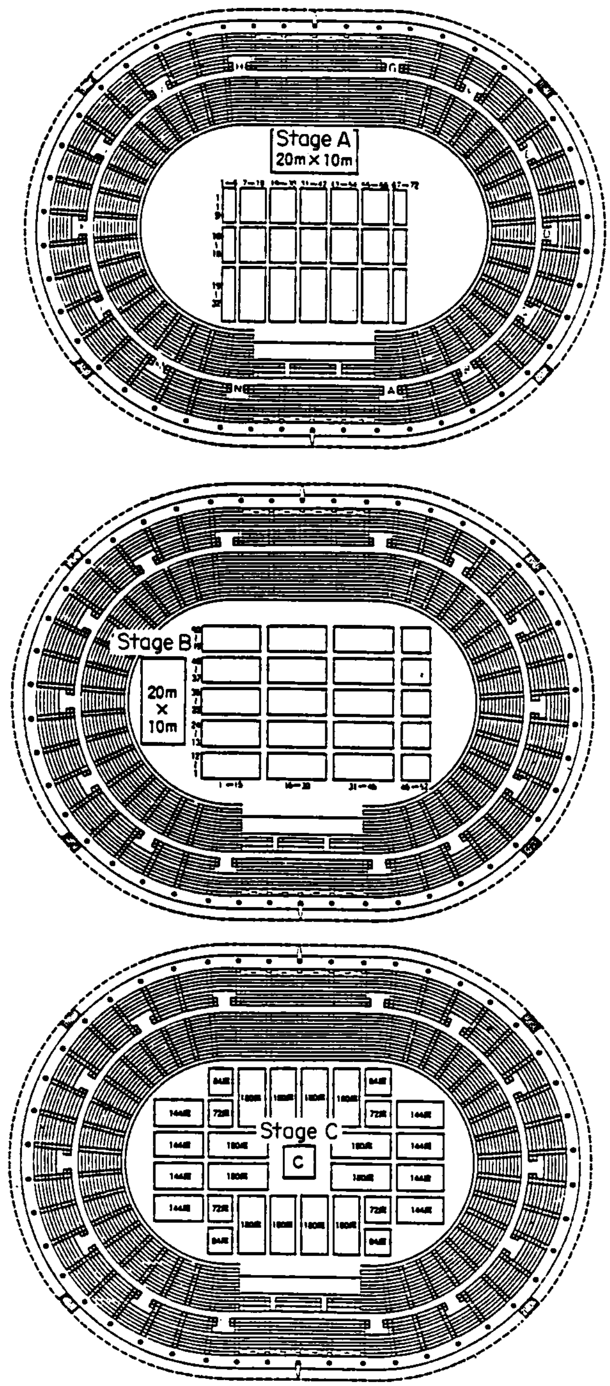


Figure 5.6.5 Example of stadium staging.

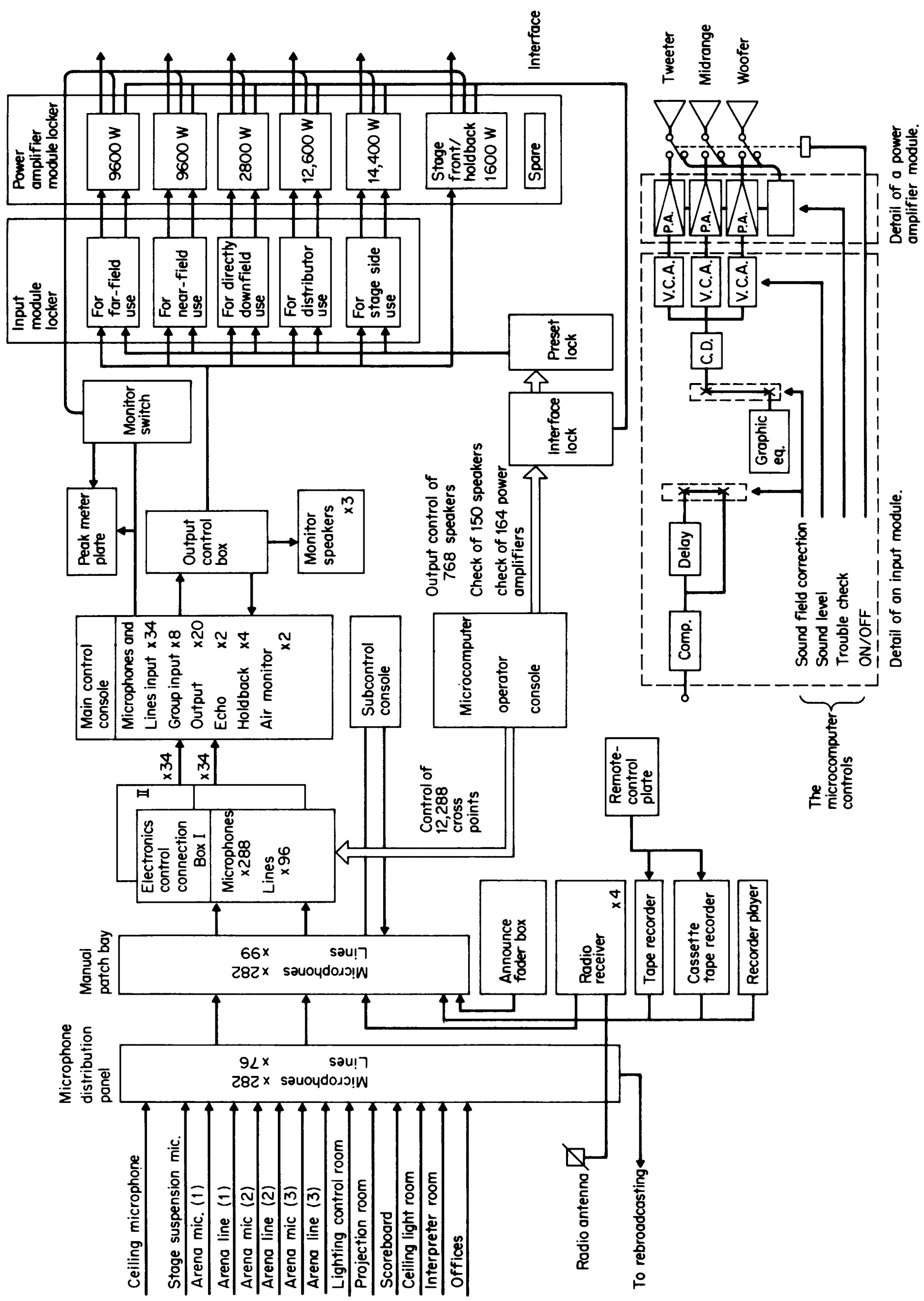


Figure 5.6.6 A computerized control system

Thus, it follows that to secure a 90-dB sound pressure at a position 300 m from the speaker, 12 speaker units with 104 dB (1 W × 1 m) and a power of 600 W × 2 are required. Actually, considering the program peak factor, a 200-W power amplifier per unit is used with a 108 dB/(1 W × 1 m) driver unit, a total of approximately 2400 W must be generated by the power amplifier units.

Installation Location

The speaker system should be installed so as not to obstruct the audience's field of vision. It is desirable that the front side of the system be acoustically transparent so as to emit sound as efficiently as possible. Also, proper measures should be taken against rain and wind. Moreover, in ballparks and other stadiums, the scoreboard, electric signboard, and/or large-screen video display system are set up where they can be seen clearly by the audience, as they are important for the enjoyment of games. Accordingly, it is desirable that sound be transmitted from the same general location.

Weather Resistance of Speaker Unit

As the horn loudspeaker's diaphragm for midrange and high range is metallic and the horn also is made of metal or reinforced plastic, weather resistance does not present a problem. However, for the low-frequency range the cone at the driver portion usually is made of paper and the housing of wood; therefore, proper measures should be taken to protect them from rain. It is necessary to apply sufficient preservative to the housing and assure proper surface treatment. The cone paper should be given a moisture-proof treatment with silicone or have reinforced olefin (or a similar substance) applied to the surface.

Directivity at Midrange and High Range

To transmit PA sound uniformly over an entire stadium, the frequency spectrum to be covered by mid-range and high-range speakers should be determined after careful consideration. In many cases, sound reflected on buildings and grounds reaches seats, impairing clarity. Advances in speaker techniques have produced horn speakers with sharp horizontal and vertical directivity control, and computer-based simulation design programs make it possible to calculate sound pressure distribution, the effect of long-distance echoes, and related factors in advance of installation. In setting up a speaker system for a large outdoor facility, factors such as weather conditions, time, geography, and so on should be taken into consideration, as well as the noise level for residents near the facility.

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Chapter

5.7

Headphones and Headphone Systems

Katsuaki Satoh

5.7.1 Introduction

A headphone consists of an electrical acoustic transducer, the ear cups containing it, ear cushions for forming a small space with the external ear, a headband connecting the right and left ear cups, and various adjusting mechanisms for attaching the headphone correctly. Construction differs, depending on:

- The number of channels
- Ear-coupling conditions
- Sound insulation
- The electricity and sound conversion principle

Headphones are classified into monaural, stereo, and four-channel headphones according to the number of channels. The monaural headphone plays back the same sound at the right and left ears, while the four-channel headphone plays back the right and left front signal and rear signal from two transducers within the right and left ear cups. Most current headphones are stereo headphones.

Headphones are next classified into pressure and velocity types according to ear-coupling conditions. (See Figure 5.7.1.) A small space formed at the front side of the transducer that acts as the acoustic load is divided into a closed chamber or a *leak space*. For example, for the pressure-type headphone, the ear cushion is constructed so that a core material, such as foamed urethane having excellent elasticity, is wrapped with nonporous vinyl chloride sheet or synthetic leather. Sometimes, liquid is sealed inside, improving sealing performance in the front chamber.

In the velocity-type headphone the ear cushion may be formed of foamed urethane with acoustic permeability, or numerous through-holes may be provided on the surface of the pressure-type headphone ear cushion. With ear-coupling conditions, priority is given to the acoustic permeability of the ear cushion; while with sound insulation, priority is given to both the ear cushion and the acoustic permeability of the ear cup. This depends on whether external sound is heard or not. If either the ear cushion or the ear cup is acoustically permeable, the headphone is an open-air type (velocity type). If both are nonporous, the headphone is a closed type (pressure type). Because the closed-type headphone is interrupted acoustically from the outside, it suffers from a unique incompatibility, while the open-air type has no such drawback. Therefore, the

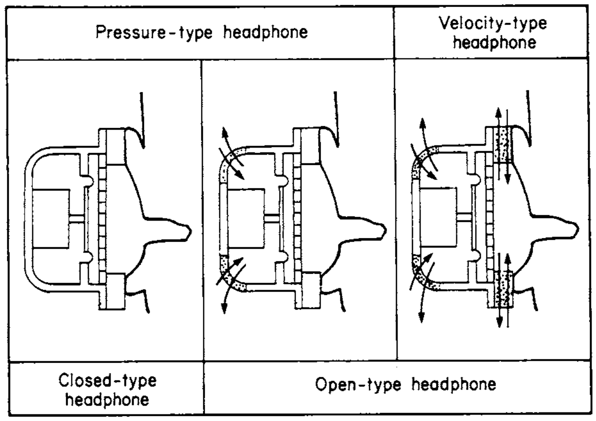


Figure 5.7.1 Classification of headphones by ear coupling and sound isolation.

open-air type has been commonly used. The closed-type headphone is typically used for live recording and monitoring in broadcasting studios.

5.7.2 Electrical Transducer

A number of electrical acoustic transducers can be used for headphone applications, although the dynamic headphone is the principal type. Assuming voltage and current applied to the electric terminal of the transducer to be $\dot{E}$ and $\dot{I}$, electric-system impedance to be Z_E, admittance to be Y_E, mechanical-system impedance to be Z_M, and load impedance converting the acoustic-system impedance to the mechanical-system impedance to be Z_{MA}, the vibrating speed V of the mechanical system is as shown in Table 5.7.1. However, A is the *force factor*, and for the dynamic type it corresponds to the product of magnetic-flux density of magnetic space B, in webers per square meter, and effective voice-coil wire length, in meters. For the static type, assuming the capacitance in stationary conditions to be C_0 in farads, and field strength to be G_0, in volts per meter,

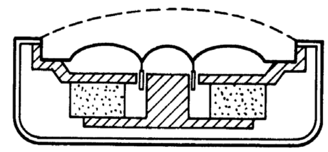
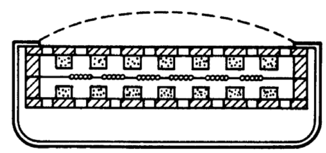
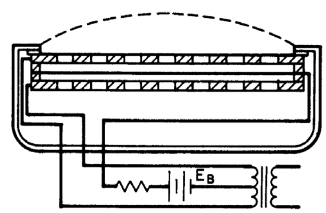
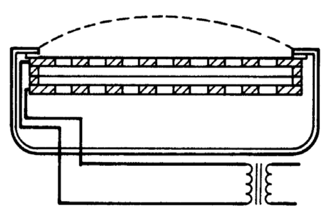
$$A = C_0 G_0 \tag{5.7.1}$$

For an electret static type, assuming the induced charge to be q, in coulombs, and gap size to be d_0, in meters,

$$A = \frac{q}{d_0} \tag{5.7.2}$$

Normally, for the sound-pressure-frequency characteristics of the headphone, the sound pressure within a standard vessel (coupler) corresponding to the volume of space consisting of the

Table 5.7.1 Structure and Velocity of Various Headphone Systems

Velocity of diaphragm	Structure
Dynamic transducer type Concentrate drive $V = \dfrac{A}{Z_E(Z_M + Z_{MA}) + A^2}\dot{E}$ $A = B\ell$	
Distribution drive $V = \dfrac{A}{Z_E(Z_M + Z_{MA}) + A^2}\dot{E}$ $A = B\ell$	
Electrostatic transducer type Bias type $V = \dfrac{A}{Y_B(Z_M + Z_{MA}) - A^2}\dot{I}$ $A = \dfrac{2\varepsilon_0 S E_B}{d^2}$	
Electret type $V = \dfrac{AI}{Y_E(Z_M + Z_{MA}) - A^2}$ $A = \dfrac{2q}{g_0}$	

ear and the headphone is measured, but since resonance occurs according to the shape, an objective comparative evaluation is difficult.

Another tone evaluation method is a response-measuring technique that uses actual ears to measure the sound pressure at the outer-ear entrance or eardrum surface with a probe tube microphone. However, because the results of this method differed among individuals and there was a danger of ear injury, the artificial ear shown in Figure 5.7.2 was recommended by the International Electrotechnical Commission (IEC) in 1970. This is as near to the value of acoustic impedance of the human ear as practical; the space capacity is 2.5 cc at approximately 8 kHz or more, 11.8 cc at approximately 200 Hz, and 4.3 cc in between. This artificial ear generally is used as a measuring instrument for headphone sound-pressure-frequency characteristics.

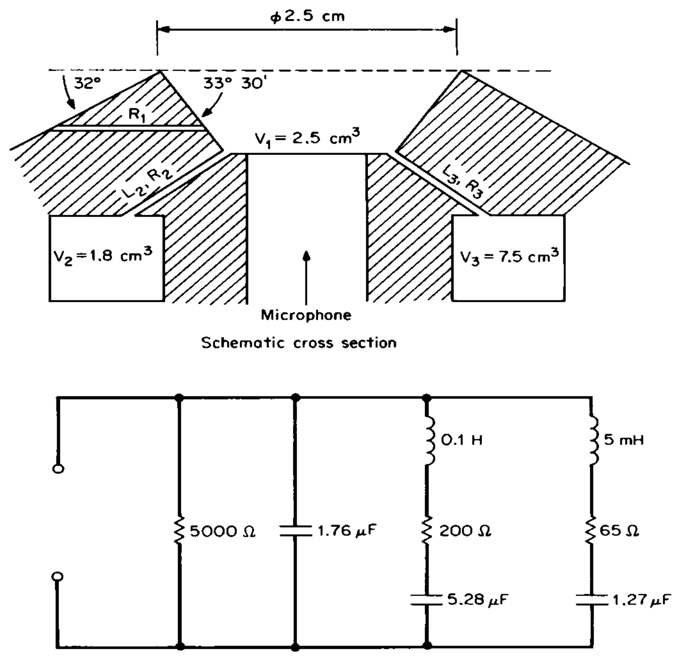


Figure 5.7.2 Standard ear specified in IEC Publication 318. In this analog 1 electrical ohm = 10^5 ns/m^{-5}. (*Courtesy of IEC.*)

5.7.2a Equivalent Circuit and Operational Analysis

In the preceding section, the relationship between the electrical and mechanical systems was discussed. In this section the relationship of the mechanical system to the acoustic system is described; also shown is the equation for headphone sound pressure.

Figure 5.7.3 shows (*a*) a sketch of a headphone held at the ears and (*b*) mechanical and acoustic circuits. Assume that a system consists of a headphone, eardrum, and ear cushion and that a small space of volume W_0, in cubic meters—including a volume equivalent to the ear acoustic impedance—is open to the outside by means of acoustic resistance r_A, in newton seconds per quintuple meter, caused by leaks and egress M_A, in kilograms per quadruple meter. The acoustic capacity C_A, in quintuple meters per newton, of this small space is determined by

$$C_A = \frac{W_0}{p_0 c^2} \tag{5.7.3}$$

When pressure $\dot{P}$, in newtons per square meter, is applied to this small space by displacing the diaphragm of effective S_d, in square meters, to the inside by X, in meters, the volume velocity $\dot{U}$, in cubic meters per second, is relieved not only by the small space but by r, M, and ear acoustic impedance Z. Therefore, the following two equations are obtained:

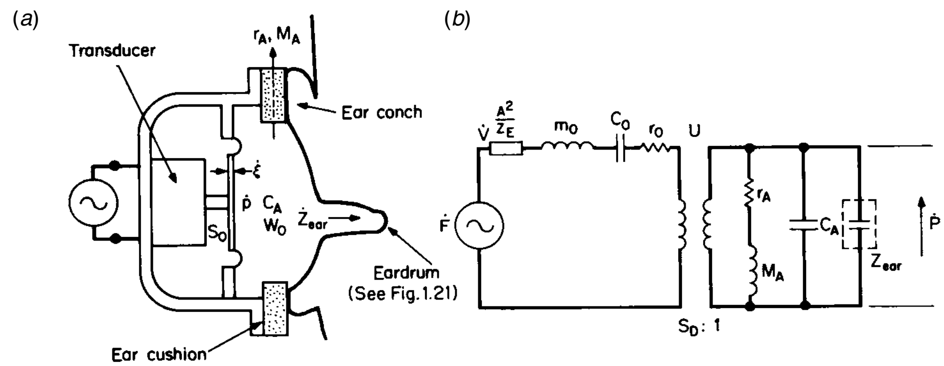


Figure 5.7.3 Headphone characteristics: (*a*) placement of hearing aid in ear, (*b*) equivalent circuit of acoustic and mechanical systems.

$$\dot{U} = \left(j\omega C_A + \frac{1}{r_A + j\omega M_A}\right)\dot{P} \tag{5.7.4}$$

$$\dot{U} = S_d \cdot \frac{d\dot{x}}{dt} = S_d \times \dot{V} \tag{5.7.5}$$

Next, the relationship between the sound pressure P generated inside the small space through the headphone and the diaphragm vibrating velocity is found:

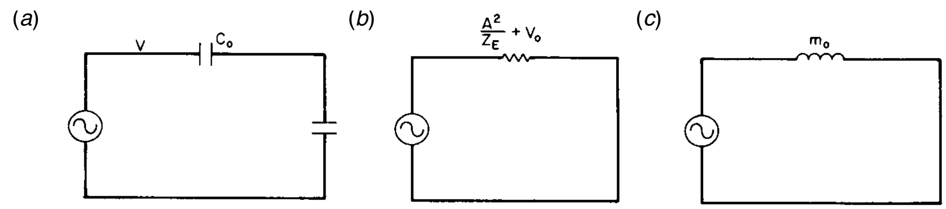
$$\dot{P} = \left(\frac{S_d}{j\omega C_A + \frac{1}{r_A + j\omega M_A}}\right)\dot{V} \tag{5.7.6}$$

Consequently, taking a dynamic headphone, for example, the relationship between sound pressure $\dot{P}$ and input voltage E is

$$\dot{P} = \frac{S_d}{j\omega C_A + \frac{1}{r_A + j\omega M_A}} \times \frac{A\dot{E}}{Z_E\left(Z_M + \frac{A^2}{Z_E}\right)} \tag{5.7.7}$$

Pressure-Type Headphone

For a pressure-type headphone (Figure 5.7.4), normally



(a) in the case of $f \ll f_0$, $\dot{V} = A\dot{E}/Z_E \times j\omega CC_0 + C_{MA}$

(b) in the case of $f = f_0$, $\dot{V} = A^2/\ Z_E + \gamma_0$

(c) in the case of $f \ll f$, $\dot{V} = A\dot{E}/\ Z_E \times\ 1/\ j\omega m$

Figure 5.7.4 Equivalent circuit of pressure-type headphone.

$$\frac{1}{WC_A} \ll r_A \ll \omega M_A$$

Consequently, Equation (5.7.7) is

$$\dot{P} = \left(\frac{S_d}{j\omega C_A}\right)\dot{V} = \frac{p_0 C^2 Sd}{j\omega W_0}\,\frac{A\dot{E}}{Z_E}\,\frac{1}{Z_M + A^2/Z_E} \tag{5.7.8}$$

If the value, when acoustic capacity C_A is converted to the mechanical system, is

$$C_{MA} = \frac{C_A}{S_d^2}$$

the following is obtained

$$Z_M + A^2/Z_E = A^2/Z_E + r_0 + j\left[\omega m_0 - \frac{(C_0 + C_{MA})}{\omega(C_0 \times C_{MA})}\right] \tag{5.7.9}$$

Consequently, the sound pressure characteristics of a pressure-type headphone are found by

$$|\dot{P}| = \frac{p_0 C^2 S_d}{W_0} \frac{A|\dot{E}|}{Z_E} \frac{1}{\sqrt{\left(\frac{C_0 + C_{MA}}{C_0 \times C_{MA}} - \omega m_0\right)^2 + (A^2/Z_E + r_0)^2}} \tag{5.7.10}$$

When $f \ll f_0$ and $f_0 \ll f$, check the operation in the equivalent circuit shown in Figure 5.7.4.

From Figure 5.7.3 sound pressure is determined by

$$|\dot{P}|_{f<f_0} = \frac{p_0 C^2 S_d}{W_0} \times \frac{A|\dot{E}|(C_0 \times C_{MA})}{Z_E(C_0 + C_{MA})} \tag{5.7.11}$$

$$|\dot{P}|_{f<f_0} = \frac{p_0 C^2 S_d}{W_0} \frac{A|\dot{E}|}{Z_E} \frac{1}{m_0 \omega_0^2} \tag{5.7.12}$$

$$|\dot{P}|_{f \gg f_0} = \frac{p_0 C^2 S_d}{W_0} \frac{A|\dot{E}|}{Z_E} \frac{1}{m_0 \omega^2} \tag{5.7.13}$$

$$|\dot{P}|_{f = f_0} = \frac{p_0 C^2 S_d}{W_0} \times \frac{A|\dot{E}|}{Z_E} \times \frac{Q_0}{m_0 \omega_0^2} \left(Q_0 = \frac{\omega_0 m_0}{r_0 + A^2/Z_E}\right) \tag{5.7.14}$$

To decrease sound pressure sensitivity from Equation 5.7.14, increase the effective area of the diaphragm and the power coefficient, and reduce W_0 and m_0. To make the sound-pressure-frequency characteristics flat, set the resonance frequency of the vibrating system to the high-frequency limit of the playback band for stiffness control. The solid line of Figure 5.7.5 shows the sound-pressure-frequency characteristics of a general-pressure-type headphone. When Q_0 is high in the resonance zone, a peak is generated; therefore, satisfactory damping is required. Furthermore, in the low range, if leak resistance r_A is present, sensitivity will be lowered at a low frequency: $1/WC_A > r_A$.

Velocity- Type Headphone

The velocity-type headphone should be set to the relation

$WM_A \ll r_A \ll 1/WC_A$

in the acoustic circuit. Sound pressure $\dot{P}$ is

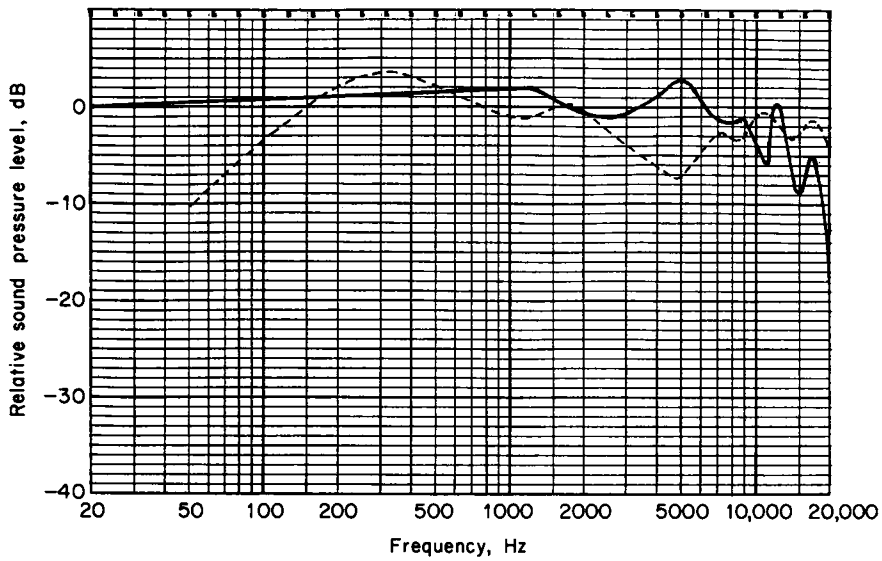


Figure 5.7.5 Frequency characteristics of different types of headphones (solid line = pressure type, dashed line = velocity type).

$$\dot{P} = r_A S_d \dot{V} = r_A S_d \frac{A\dot{E}}{Z_E} \times \frac{1}{Z_M + \frac{A^2}{Z_E}} \qquad (5.7.15)$$

As with the pressure type, assume the value when C_A is converted to the mechanical system of acoustic resistance r_A to be $C_{MA} = S_d^2 r_A$). The following is obtained

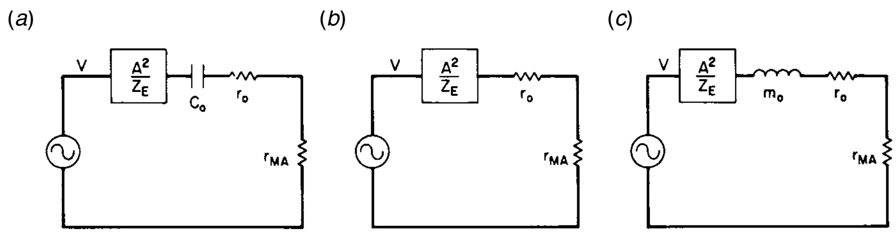
$$Z_M + \frac{A^2}{Z_E} = \frac{A^2}{Z_E} + r_0 + r_{MA} + j\left(\omega m_0 - \frac{1}{\omega C_0}\right) \qquad (5.7.16)$$

The sound-pressure-frequency characteristics are determined by

$$|\dot{P}| = r_A S_d \times \frac{A|\dot{E}|}{Z_E} \sqrt{\left(\frac{A^2}{Z_E} + r_0 + r_{MA}\right)^2 + \left(\omega m_0 - \frac{1}{\omega C_0}\right)^2} \qquad (5.7.16)$$

The equivalent circuit when $f \ll f_0$ and $f \gg f_0$ is shown in Figure 5.7.6.

To increase sound pressure sensitivity from these equations, increase the effective area of the diaphragm and the power coefficient, and reduce the mechanical impedance. To make the sound-pressure-frequency-characteristics flat, reduce the vibrating-system mass for high compliance to



(a) in the case of $f \ll f_0$, $\dot{V} = \dfrac{A \times \dot{E}/\ Z_E}{A^2/\ Z_E + r_0 + r_{MA} + 1/\ j\omega C_0}$

(b) in the case of $f = f_0$, $\dot{V} = \dfrac{A \times \dot{E}/\ Z_E}{A^2/\ Z_E + r_0 + r_{MA}}$

(c) in the case of $f_0 \ll f$, $\dot{V} = \dfrac{A \times \dot{E}/\ Z_E}{A^2/\ (Z_E + r_0 + r_{MA} + j\omega M_0)}$

Where:
$\dot{E}$ = signal voltage, V
V = velocity of diaphragm, m/s
A = force factor, N/A
Z_E = electrical impedance, W
r_0 = mechanical resistance, mechanical ohms
r_{MA} = resistance of coupling space, mechanical ohms
C_0 = compliance of diaphragm, m/N
M_0 = mass of diaphragm, kg

Figure 5.7.6 Equivalent circuit of a velocity-type headphone:

obtain the vibrating system of resistance control. Consequently, the vibrating velocity of the diaphragm inevitably becomes constant (velocity type). The dashed line of Figure 5.7.5 shows the sound-pressure-frequency characteristics of a velocity-type headphone.

The preceding paragraphs presented the equations by which the sound-pressure-frequency characteristics of pressure- and velocity-type headphones are found for the simplest structures in which only the acoustic load at the front of the headphone is considered. In practice, optimum sound-pressure-frequency characteristics, including the acoustic load, are realized at the rear of the electrical acoustic transducer.

5.7.3 Bibliography

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