PROFESSIONAL

Sound System Design Reference Manual

Table of Contents

Table of Contents (cont.)

Preface to the 1999 Edition:

This third edition of JBL Professional's Sound System Design Reference Manual is presented in a new graphic format that makes for easier reading and study. Like its predecessors, it presents in virtually their original 1977 form George Augspurger's intuitive and illuminating explanations of sound and sound system behavior in enclosed spaces. The section on systems and case studies has been expanded, and references to JBL components have been updated.

The fundamentals of acoustics and sound system design do not change, but system implementation improves in its effectiveness with ongoing developments in signal processing, transducer refinement, and front-end flexibility in signal routing and control.

As stated in the Preface to the 1986 edition: The technical competence of professional dealers and sound contractors is much higher today than it was when the Sound Workshop manual was originally introduced. It is JBL's feeling that the serious contractor or professional dealer of today is ready to move away from simply plugging numbers into equations. Instead, the designer is eager to learn what the equations really mean, and is intent on learning how loudspeakers and rooms interact, however complex that may be. It is for the student with such an outlook that this manual is intended.

John Eargle
January 1999

Chapter 1: Wave Propagation

Wavelength, Frequency, and Speed of Sound

Sound waves travel approximately 344 m/sec (1130 ft/sec) in air. There is a relatively small velocity dependence on temperature, and under normal indoor conditions we can ignore it. Audible sound covers the frequency range from about 20 Hz to 20 kHz. The wavelength of sound of a given frequency is the distance between successive repetitions of the waveform as the sound travels through air. It is given by the following equation:

wavelength = speed/frequency

or, using the common abbreviations of *c* for speed, *f* for frequency, and λ for wavelength:

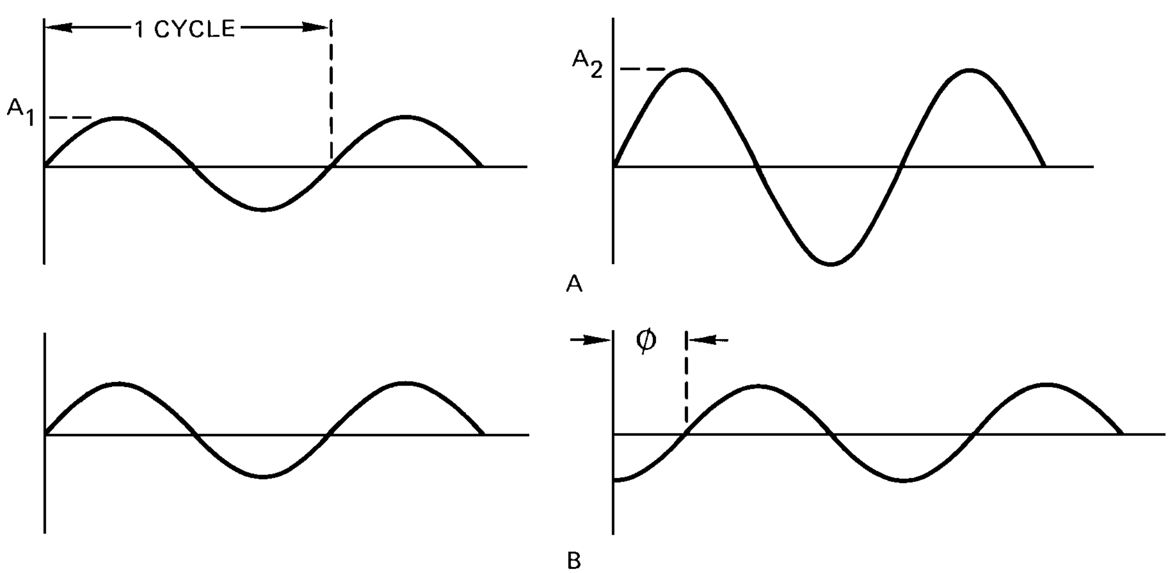
$$\lambda = c/f$$

Period (T) is defined as the time required for one cycle of the waveform. T = 1/f.

For f = 1 kHz, T = 1/1000, or 0.001 sec, and λ = 344/1000, or .344 m (1.13 ft.)

The lowest audible sounds have wavelengths on the order of 10 m (30 ft), and the highest sounds have wavelengths as short as 20 mm (0.8 in). The range is quite large, and, as we will see, it has great bearing on the behavior of sound.

The waves we have been discussing are of course *sine waves*, those basic building blocks of all speech and music signals. Figure 1-1 shows some of the basic aspects of sine waves. Note that waves of the same frequency can differ in both amplitude and in phase angle. The amplitude and phase angle relationships between sine waves determine how they combine, either acoustically or electrically.



A — TWO SINE WAVES DIFFERING IN AMPLITUDE
B — TWO SINE WAVES DIFFERING IN PHASE RELATIONSHIP

Figure 1-1. Properties of sine waves

Combining Sine Waves

Referring to Figure 1-2, if two or more sine wave signals having the same frequency and amplitude are added, we find that the resulting signal also has the same frequency and that its amplitude depends upon the phase relationship of the original signals. If there is a phase difference of 120°, the resultant has exactly the same amplitude as either of the original signals. If they are combined in phase, the resulting signal has twice the amplitude of either original. For phase differences between l20° and 240°, the resultant signal always has an amplitude less than that of either of the original signals. If the two signals are exactly 180° out of phase, there will be total cancellation.

In electrical circuits it is difficult to maintain identical phase relationships between all of the sine components of more complex signals, except for the special cases where the signals are combined with a 0° or 180° phase relationship. Circuits which maintain some specific phase relationship (45°, for example) over a wide range of frequencies are fairly complex. Such wide range, all-pass phase-shifting networks are used in acoustical signal processing.

When dealing with complex signals such as music or speech, one must understand the concept of *coherence*. Suppose we feed an electrical signal through a high quality amplifier. Apart from very small amounts of distortion, the output signal is an exact replica of the input signal, except for its amplitude. The two signals, although not identical, are said to be highly coherent. If the signal is passed through a poor amplifier, we can expect substantial differences between input and output, and coherence will not be as great. If we compare totally different signals, any similarities occur purely at random, and the two are said to be non-coherent.

When two non-coherent signals are added, the *rms* (root mean square) value of the resulting signal can be calculated by adding the relative powers of the two signals rather than their voltages. For example, if we combine the outputs of two separate noise generators, each producing an rms output of 1 volt, the resulting signal measures 1.414 volts rms, as shown in Figure 1-3.

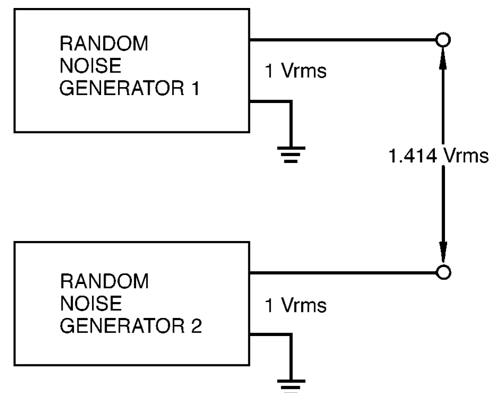


Figure 1-3. Combining two random noise generators

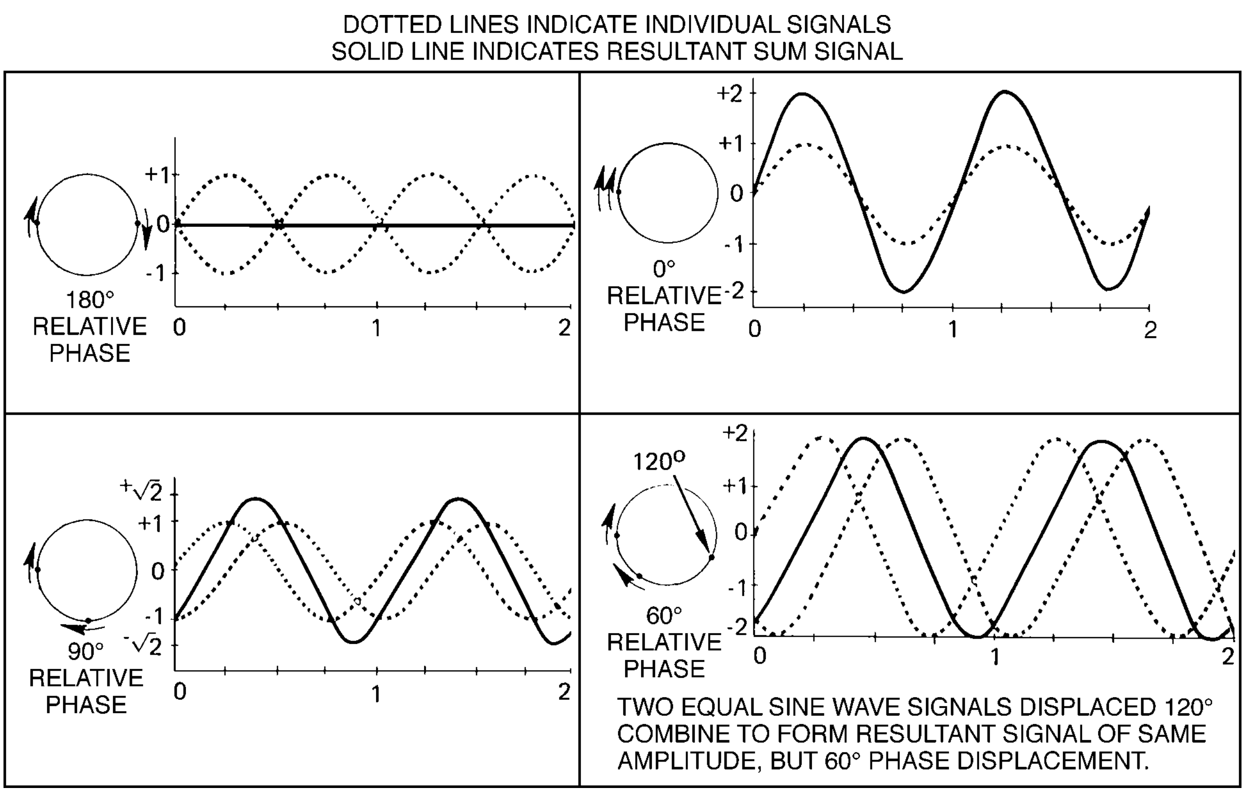


Figure 1-2. V ector addition of two sine waves

Combining Delayed Sine Waves

If two coherent wide-range signals are combined with a specified time difference between them rather than a fixed phase relationship, some frequencies will add and others will cancel. Once the delayed signal arrives and combines with the original signal, the result is a form of "comb filter," which alters the frequency response of the signal, as shown in Figure 1-4. Delay can be achieved electrically through the use of all-pass delay networks or digital processing. In dealing with acoustical signals in air, there is simply no way to avoid delay effects, since the speed of sound is relatively slow.

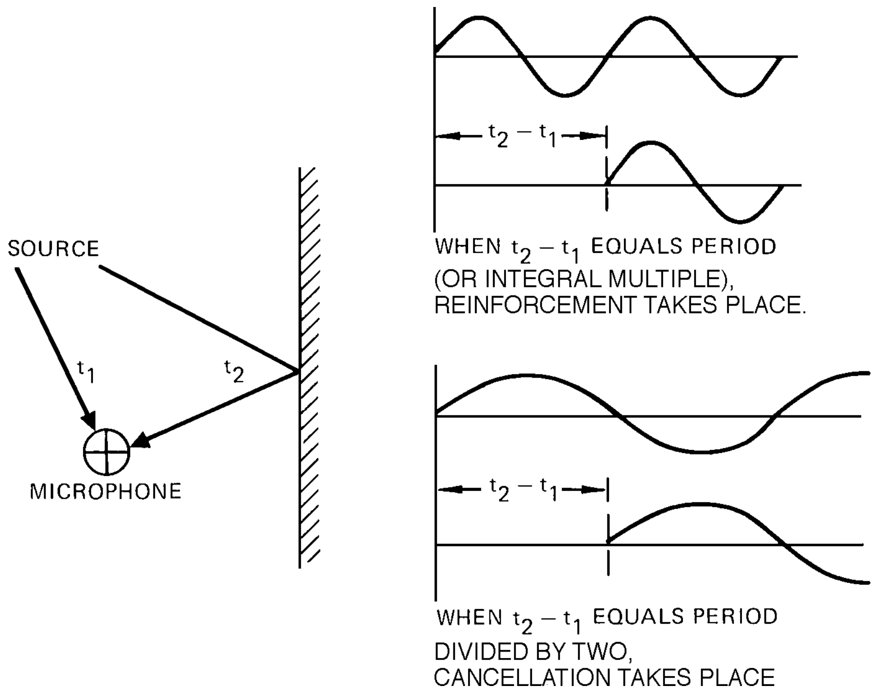
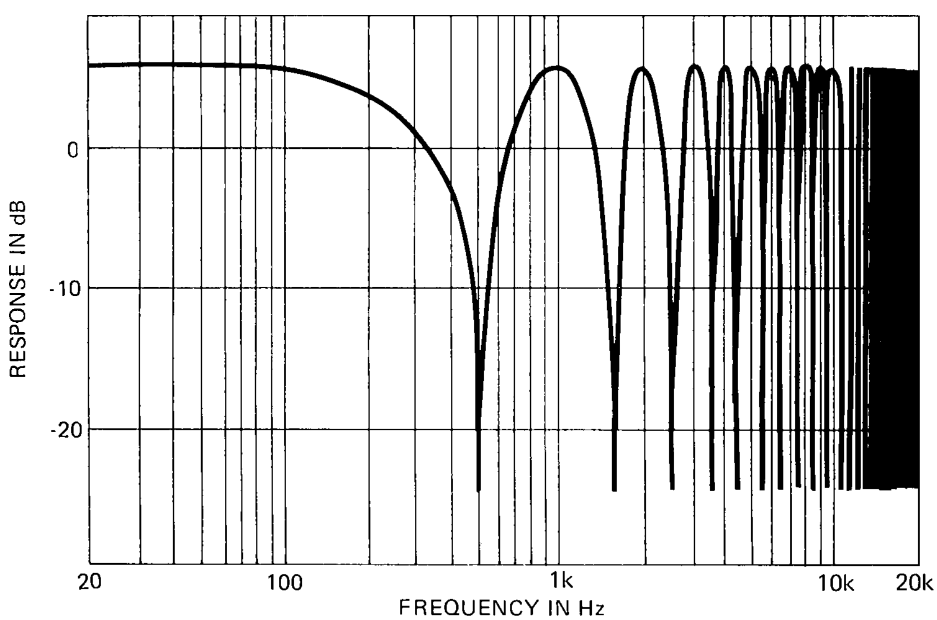


Figure 1-4A. Combining delayed signals



FREQUENCY RESPONSE OF TWO WIDE-RANGE PROGRAM CHANNELS COMBINED WITH ONE MILLISECOND DELAY BETWEEN CHANNELS. SAME PROGRAM SIGNAL FED TO BOTH, EXCEPT THAT AMPLITUDES DIFFER BY 0.5 dB.

Figure 1-4B. Combining of coherent signals with constant time delay

A typical example of combining delayed coherent signals is shown in Figure 1-5. Consider the familiar outdoor PA system in which a single microphone is amplified by a pair of identical separated loudspeakers. Suppose the loudspeakers in question are located at each front corner of the stage, separated by a distance of 6 m (20 ft). At any distance from the stage along the center line, signals from the two loudspeakers arrive simultaneously. But at any other location, the distances of the two loudspeakers are unequal, and sound from one must arrive slightly later than sound from the other. The illustration shows the dramatically different frequency response resulting from a change in listener position of only 2.4 m (8 ft). Using random noise as a test signal, if you walk from Point B to Point A and proceed across the center line, you will hear a pronounced swishing effect, almost like a siren. The change in sound quality is most pronounced near the center line, because in this area the response peaks and dips are spread farther apart in frequency.

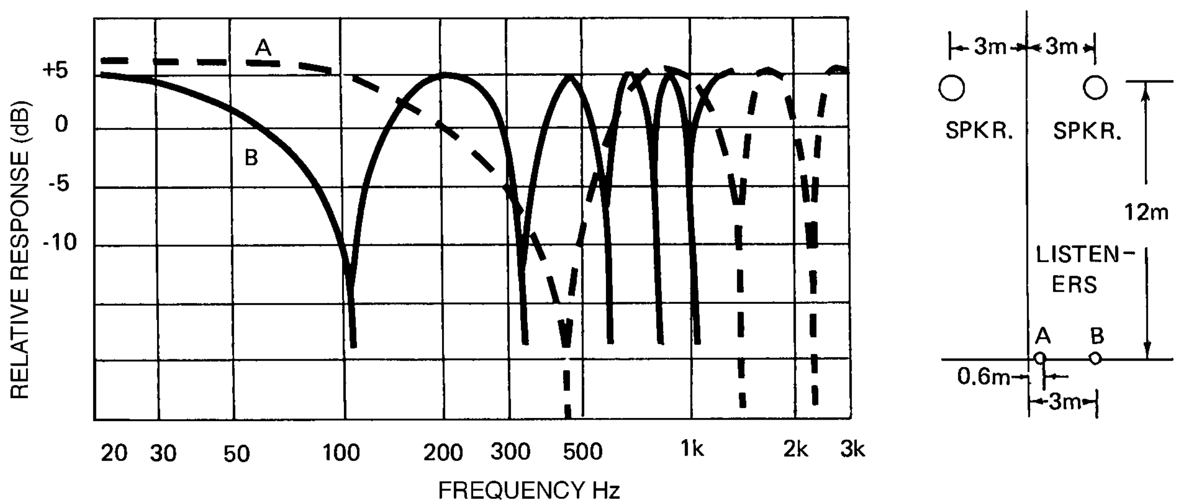


Figure 1-5. Generation of interference effects (comb filter response) by a split array

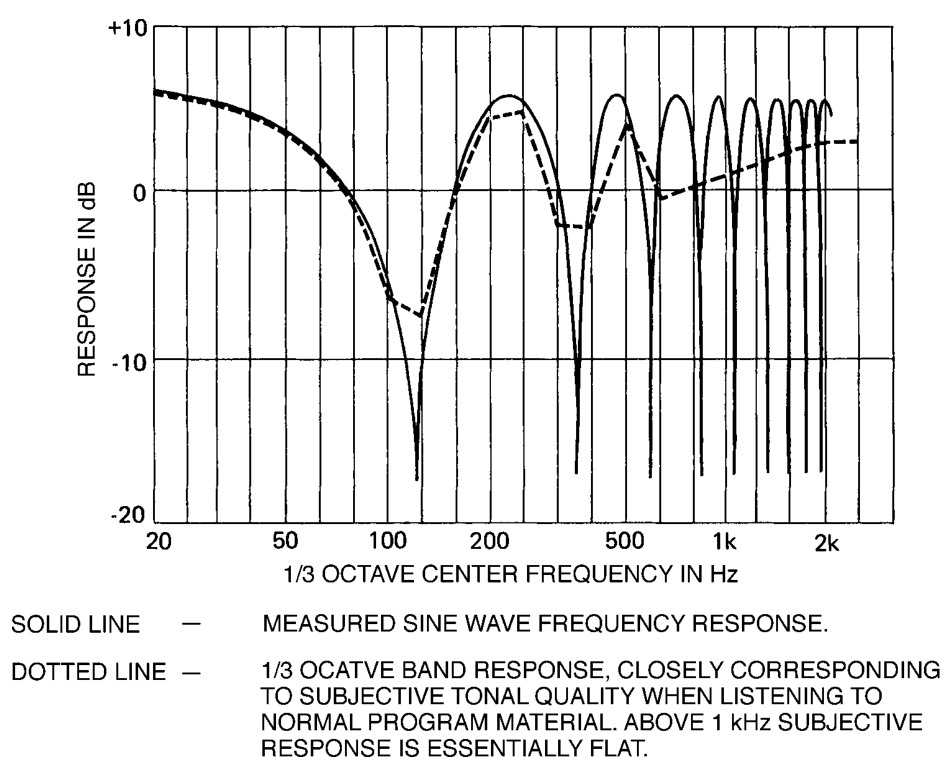


Figure 1-6. Audible effect of comb filters shown in Figure 1-5

Subjectively, the effect of such a comb filter is not particularly noticeable on normal program material as long as several peaks and dips occur within each one-third octave band. See Figure 1-6. Actually, the controlling factor is the “critical bandwidth.” In general, amplitude variations that occur within a critical band will not be noticed as such. Rather, the ear will respond to the signal power contained within that band. For practical work in sound system design and architectural acoustics, we can assume that the critical bandwidth of the human ear is very nearly one-third octave wide.

In houses of worship, the system should be suspended high overhead and centered. In spaces which do not have considerable height, there is a strong temptation to use two loudspeakers, one on either side of the platform, feeding both the same program. We do not recommend this.

Diffraction of Sound

Diffraction refers to the bending of sound waves as they move around obstacles. When sound strikes a hard, non-porous obstacle, it may be reflected or diffracted, depending on the size of the obstacle relative to the wavelength. If the obstacle is large compared to the wavelength, it acts as an effective barrier, reflecting most of the sound and casting a substantial “shadow” behind the object. On the other hand, if it is small compared with the wavelength, sound simply bends around it as if it were not there. This is shown in Figure 1-7.

An interesting example of sound diffraction occurs when hard, perforated material is placed in the path of sound waves. So far as sound is concerned, such material does not consist of a solid barrier interrupted by perforations, but rather as an open area obstructed by a number of small individual objects. At frequencies whose wavelengths are small compared with the spacing between perforations, most of the sound is reflected. At these frequencies, the percentage of sound traveling through the openings is essentially proportional to the ratio between open and closed areas.

At lower frequencies (those whose wavelengths are large compared with the spacing between perforations), most of the sound passes through the openings, even though they may account only for 20 or 30 percent of the total area.

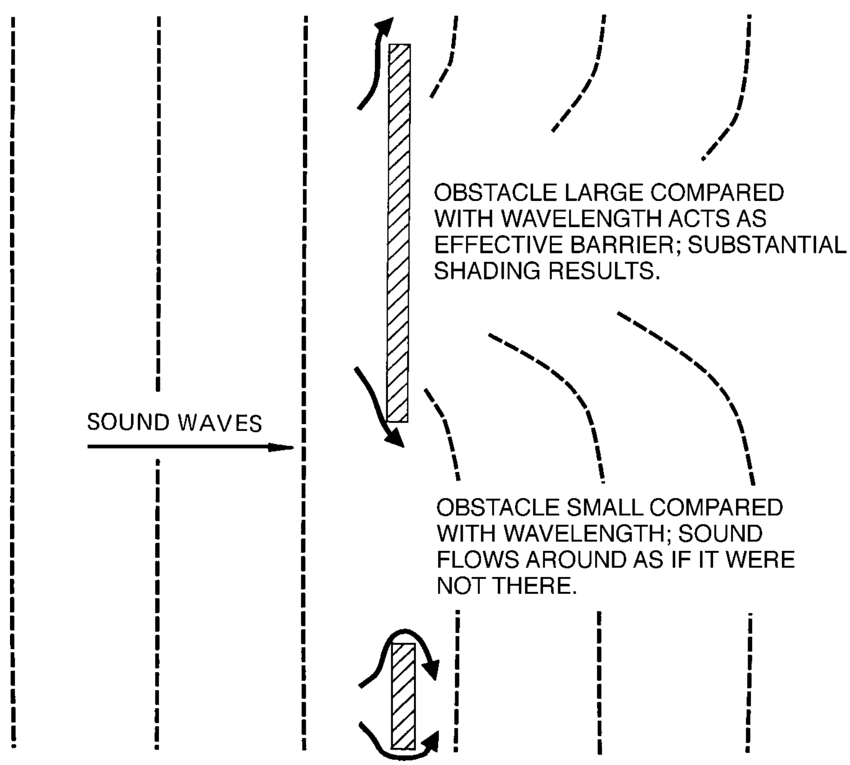


Figure 1-7. Diffraction of sound around obstacles

Effects of Temperature Gradients on Sound Propagation

If sound is propagated over large distances out of doors, its behavior may seem erratic. Differences (gradients) in temperature above ground level will affect propagation as shown in Figure 1-8. Refraction of sound refers to its changing direction as its velocity increases slightly with elevated temperatures. At Figure 1-8A, we observe a situation which often occurs at nightfall, when the ground is still warm. The case shown at *B* may occur in the morning, and its "skipping" characteristic may give rise to hot spots and dead spots in the listening area.

Effects of Wind Velocity and Gradients on Sound Propagation

Figure 1-9 shows the effect wind velocity gradients on sound propagation. The actual velocity of sound in this case is the velocity of sound in still air plus the velocity of the wind itself. Figure 1-10 shows the effect of a cross breeze on the apparent direction of a sound source.

The effects shown in these two figures may be evident at large rock concerts, where the distances covered may be in the 200 - 300 m (600 - 900 ft) range.

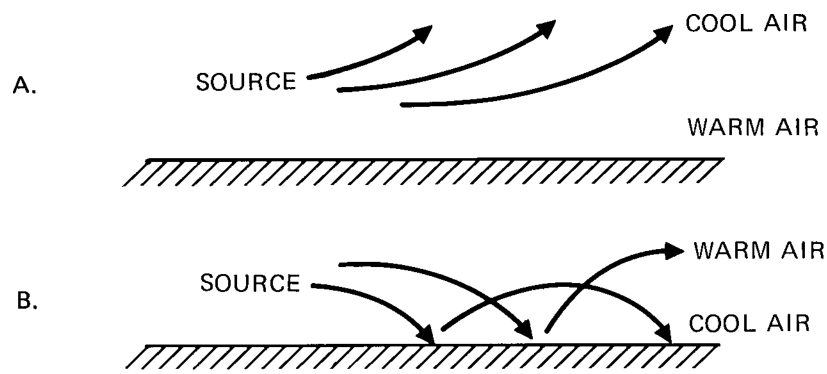


Figure 1-8. Effects of temperature gradients on sound propagation

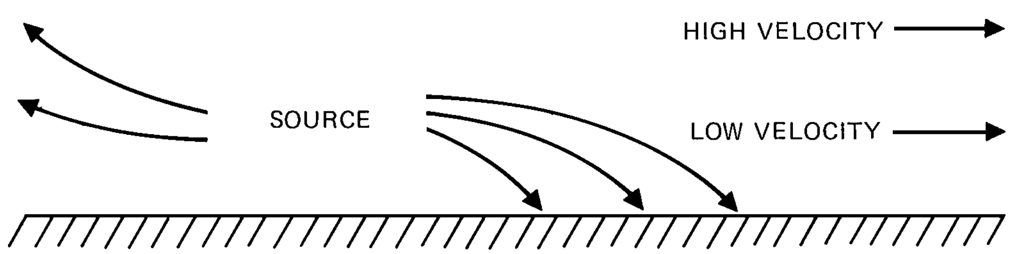


Figure 1-9. Effect of wind velocity gradients on sound propagation

Effects of Humidity on Sound Propagation

Contrary to what most people believe, there is more sound attenuation in dry air than in damp air. The effect is a complex one, and it is shown in Figure 1-11. Note that the effect is significant only at frequencies above 2 kHz. This means that high frequencies will be attenuated more with distance than low frequencies will be, and that the attenuation will be greatest when the relative humidity is 20 percent or less.

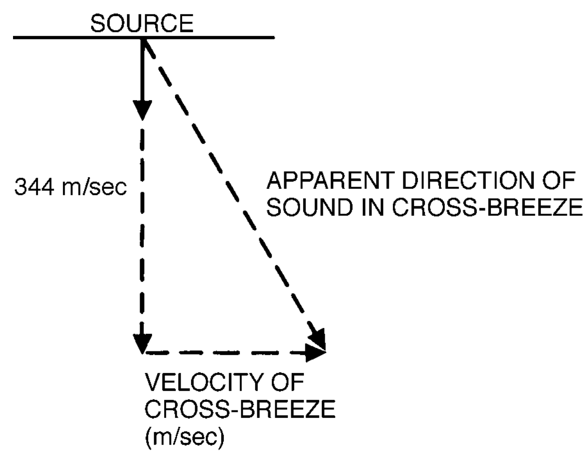


Figure 1-10. Effect of cross breeze on apparent direction of sound

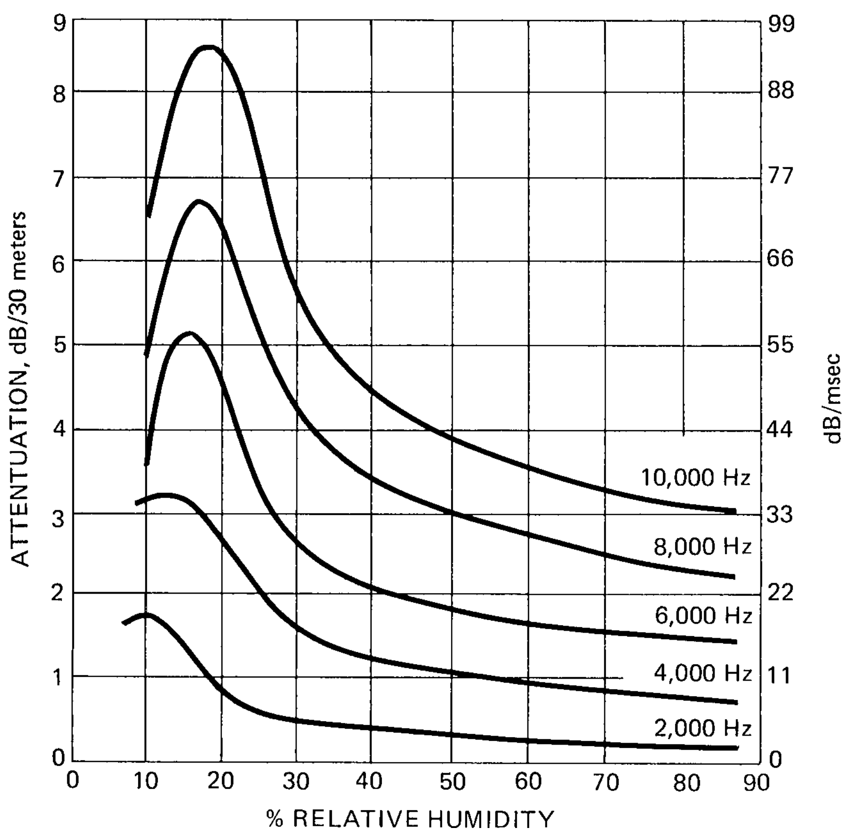


Figure 1-11. Absorption of sound in air vs. relative humidity

Chapter 2: The Decibel

Introduction

In all phases of audio technology the decibel is used to express signal levels and level differences in sound pressure, power, voltage, and current. The reason the decibel is such a useful measure is that it enables us to use a comparatively small range of numbers to express large and often unwieldy quantities. The decibel also makes sense from a psychoacoustical point of view in that it relates directly to the effect of most sensory stimuli.

Power Relationships

Fundamentally, the *bel* is defined as the common logarithm of a power ratio:

$$\text{bel} = \log (P_1/P_0)$$

For convenience, we use the *decibel*, which is simply one-tenth bel. Thus:

$$\text{Level in decibels (dB)} = 10 \log (P_1/P_0)$$

The following tabulation illustrates the usefulness of the concept. Letting P_0 = 1 watt:

P_1 (watts)	Level in dB
1	0
10	10
100	20
1000	30
10,000	40
20,000	43

Note that a 20,000-to-1 range in power can be expressed in a much more manageable way by referring to the powers as levels in dB above one watt. Psychoacoustically, a ten-times increase in power results in a level which most people judge to be Òtwice as loud.Ó Thus, a 100-watt acoustical signal would be twice as loud as a 10-watt signal, and a 10-watt signal would be twice as loud as a 1-watt signal. The convenience of using decibels is apparent; each of these power ratios can be expressed by the same level, 10 dB. Any 10 dB level difference, regardless of the actual powers involved, will represent a 2-to-1 difference in subjective loudness.

We will now expand our power decibel table:

P_1 (watts)	Level in dB
1	0
1.25	1
1.60	2
2	3
2.5	4
3.15	5
4	6
5	7
6.3	8
8	9
10	10

This table is worth memorizing. Knowing it, you can almost immediately do mental calculations, arriving at power levels in dB above, or below, one watt.

Here are some examples:

1. What power level is represented by 80 watts? First, locate 8 watts in the left column and note that the corresponding level is 9 dB. Then, note that 80 is *10 times* 8, giving another 10 dB. Thus:

$$9 + 10 = 19 \text{ dB}$$

2. What power level is represented by 1 milliwatt? 0.1 watt represents a level of minus 10 dB, and 0.01 represents a level 10 dB lower. Finally, 0.001 represents an additional level decrease of 10 dB. Thus:

$$-10 -10 -10 = -30 \text{ dB}$$

3. What power level is represented by 4 milliwatts? As we have seen, the power level of 1 milliwatt is –30 dB. Two milliwatts represents a level increase of 3 dB, and from 2 to 4 milliwatts there is an additional 3 dB level increase. Thus:

$$-30 + 3 + 3 = -24\ dB$$

4. What is the level difference between 40 and 100 watts? Note from the table that the level corresponding to 4 watts is 6 dB, and the level corresponding to 10 watts is 10 dB, a difference of 4 dB. Since the level of 40 watts is 10 dB greater than for 4 watts, and the level of 80 watts is 10 dB greater than for 8 watts, we have:

$$6 - 10 + 10 - 10 = -4\ dB$$

We have done this last example the long way, just to show the rigorous approach. However, we could simply have stopped with our first observation, noting that the dB level difference between 4 and 10 watts, .4 and 1 watt, or 400 and 1000 watts will always be the same, 4 dB, because they all represent the same power ratio.

The level difference in dB can be converted back to a power ratio by means of the following equation:

$$\text{Power ratio} = 10^{dB/10}$$

For example, find the power ratio of a level difference of 13 dB:

$$\text{Power ratio} = 10^{13/10} = 10^{1.3} = 20$$

The reader should acquire a reasonable skill in dealing with power ratios expressed as level differences in dB. A good "feel" for decibels is a qualification for any audio engineer or sound contractor. An extended nomograph for converting power ratios to level differences in dB is given in Figure 2-1.

Voltage, Current, and Pressure Relationships

The decibel fundamentally relates to power ratios, and we can use voltage, current, and pressure ratios as they relate to power. Electrical power can be represented as:

$$P = EI$$

$$P = I^2Z$$

$$P = E^2/Z$$

Because power is proportional to the square of the voltage, the effect of *doubling* the voltage is to *quadruple* the power:

$$(2E)^2/Z = 4(E)^2/Z$$

As an example, let E = 1 volt and Z = 1 ohm. Then, $P = E^2/Z = 1$ watt. Now, let E = 2 volts; then, $P = (2)^2/1 = 4$ watts.

The same holds true for current, and the following equations must be used to express power levels in dB using voltage and current ratios:

$$\text{dB level} = 10 \log\left(\frac{E_1}{E_0}\right)^2 = 20 \log\left(\frac{E_1}{E_0}\right), \text{ and}$$

$$\text{dB level} = 10 \log\left(\frac{I_1}{I_0}\right)^2 = 20 \log\left(\frac{I_1}{I_0}\right).$$

Sound pressure is analogous to voltage, and levels are given by the equation:

$$\text{dB level} = 20 \log\left(\frac{P_1}{P_0}\right).$$

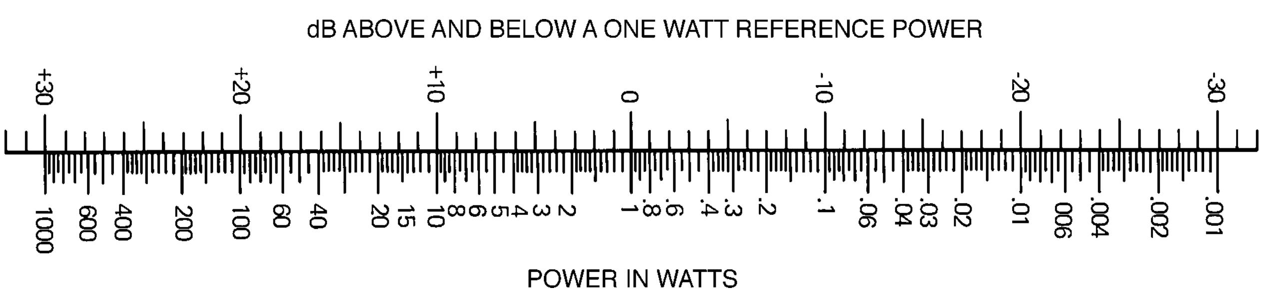


Figure 2-1. Nomograph for determining power ratios directly in dB

The normal reference level for voltage, E_0, is one volt. For sound pressure, the reference is the extremely low value of 20×10^{-6} newtons/m^2. This reference pressure corresponds roughly to the minimum audible sound pressure for persons with normal hearing. More commonly, we state pressure in *pascals* (Pa), where 1 Pa = 1 newton/m^2. As a convenient point of reference, note that an rms pressure of 1 pascal corresponds to a sound pressure level of 94 dB.

We now present a table useful for determining levels in dB for ratios given in voltage, current, or sound pressure:

Voltage, Current or Pressure Ratios	Level in dB
1	0
1.25	2
1.60	4
2	6
2.5	8
3.15	10
4	12
5	14
6.3	16
8	18
10	20

This table may be used exactly the same way as the previous one. Remember, however, that the reference impedance, whether electrical or acoustical, must remain fixed when using these ratios to determine level differences in dB. A few examples are given:

1. Find the level difference in dB between 2 volts and 10 volts. Directly from the table we observe

$$20 - 6 = 14\ dB.$$

2. Find the level difference between 1 volt and 100 volts. A 10-to-1 ratio corresponds to a level difference of 20 dB. Since 1-to-100 represents the product of *two* such ratios (1-to-10 and 10-to-100), the answer is

$$20 + 20 = 40 \text{ dB}.$$

3. The signal input to an amplifier is 1 volt, and the input impedance is 600 ohms. The output is also 1 volt, and the load impedance is 15 ohms. What is the gain of the amplifier in dB? Watch this one carefully!

If we simply compare input and output voltages, we still get 0 dB as our answer. The *voltage gain* is in fact unity, or one. Recalling that decibels refer primarily to power ratios, we must take the differing input and output impedances into account and actually compute the input and output powers.

$$\text{Input power } = \frac{E_2}{Z} = \frac{1}{600} \text{ watt}$$

$$\text{Output power } = \frac{E_2}{Z} = \frac{1}{15}$$

$$\text{Thus, } 10 \log\left(\frac{600}{15}\right) = 10 \log 40 = 16 \text{ dB}$$

Fortunately, such calculations as the above are not often made. In audio transmission, we keep track of operating levels primarily through voltage level calculations in which the voltage reference value of 0.775 volts has an assigned level of 0 dBu. The value of 0.775 volts is that which is applied to a 600-ohm load to produce a power of 1 milliwatt (mW). A power level of 0 dBm corresponds to 1 mW. Stated somewhat differently, level values in dBu and dBm will have the same numerical value only when the load impedance under consideration is 600 ohms.

The level difference in dB can be converted back to a voltage, current, or pressure ratio by means of the following equation:

$$\text{Ratio} = 10^{dB/20}$$

For example, find the voltage ratio corresponding to a level difference of 66 dB:

$$\text{voltage ratio} = 10^{66/20} = 10^{3.3} = 2000.$$

Sound Pressure and Loudness Contours

We will see the term dB-SPL time and again in professional sound work. It refers to sound pressure levels in dB above the reference of 20 x 10^{-6} N/m^2. We commonly use a *sound level meter* (SLM) to measure SPL. Loudness and sound pressure obviously bear a relation to each other, but they are not the same thing. Loudness is a subjective sensation which differs from the measured level in certain important aspects. To specify loudness in scientific terms, a different unit is used, the *phon*. Phons and decibels share the same numerical value only at 1000 Hz. At other frequencies, the phon scale deviates more or less from the sound level scale, depending on the particular frequency and the sound pressures; Figure 2-2 shows the relationship between phons and decibels, and illustrates the well-known Robinson-Dadson equal loudness contours. These show that, in general, the ear becomes less sensitive to sounds at low frequencies as the level is reduced.

When measuring sound pressure levels, weighted response may be employed to more closely approximate the response of the ear. Working with sound systems, the most useful scales on the sound level meter will be the A-weighting scale and the linear scale, shown in Figure 2-3. Inexpensive sound level meters, which cannot provide linear response over the full range of human hearing, often have no linear scale but offer a C-weighting scale instead. As can be seen from the illustration, the C-scale rolls off somewhat at the frequency extremes. Precision sound level meters normally offer *A, B,* and *C* scales in addition to linear response. Measurements made with a sound level meter are normally identified by noting the weighting factor, such as: dB(A) or dB(lin).

Typical levels of familiar sounds, as shown in Figure 2-4, help us to estimate dB(A) ratings when a sound level meter is not available. For example, normal conversational level in quiet surrounds is about 60 dB(A). Most people find levels higher than 100 dB(A) uncomfortable, depending on the length of exposure. Levels much above 120 dB(A) are definitely dangerous to hearing and are perceived as painful by all except dedicated rock music fans.

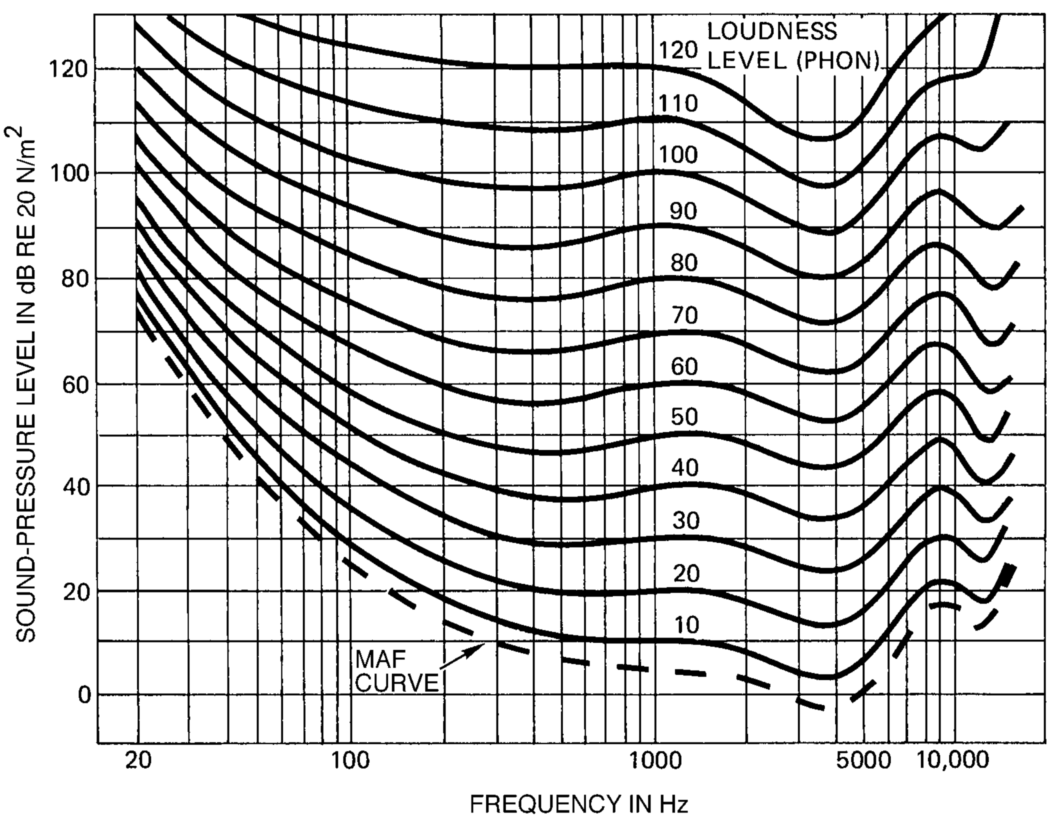


Figure 2-2. Free-field equal loudness contours

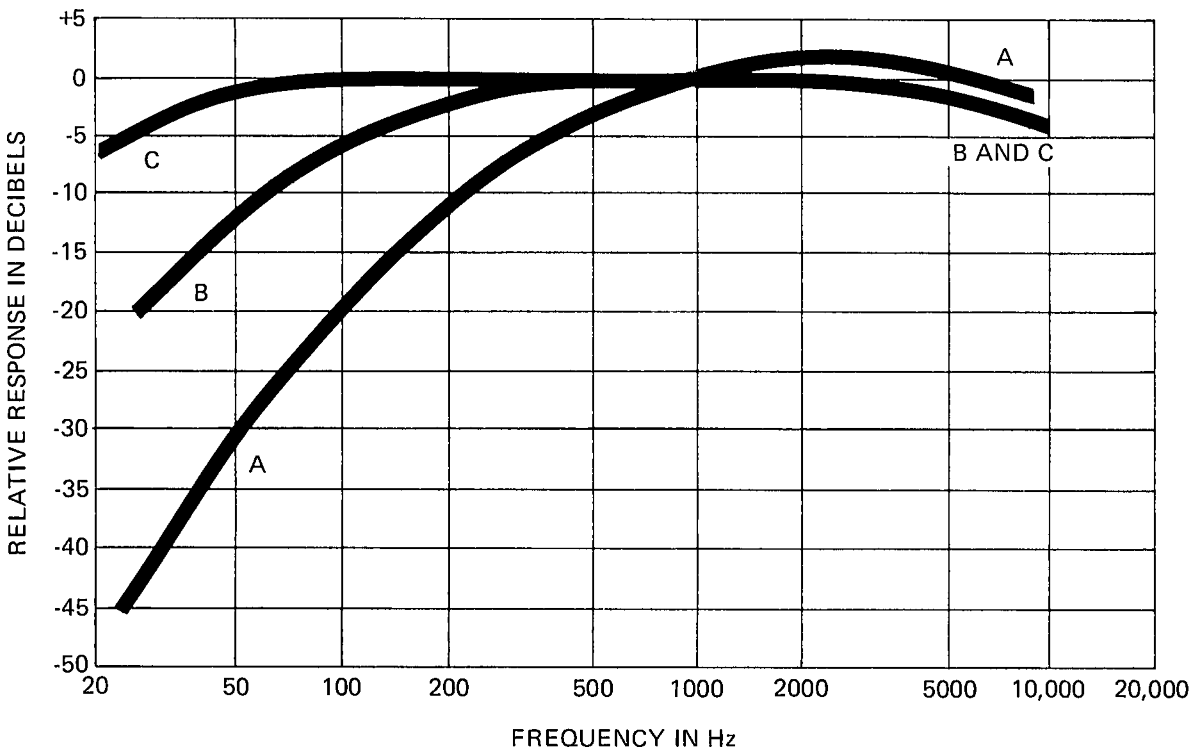


Figure 2-3. Frequency responses for SLM weighting characteristics

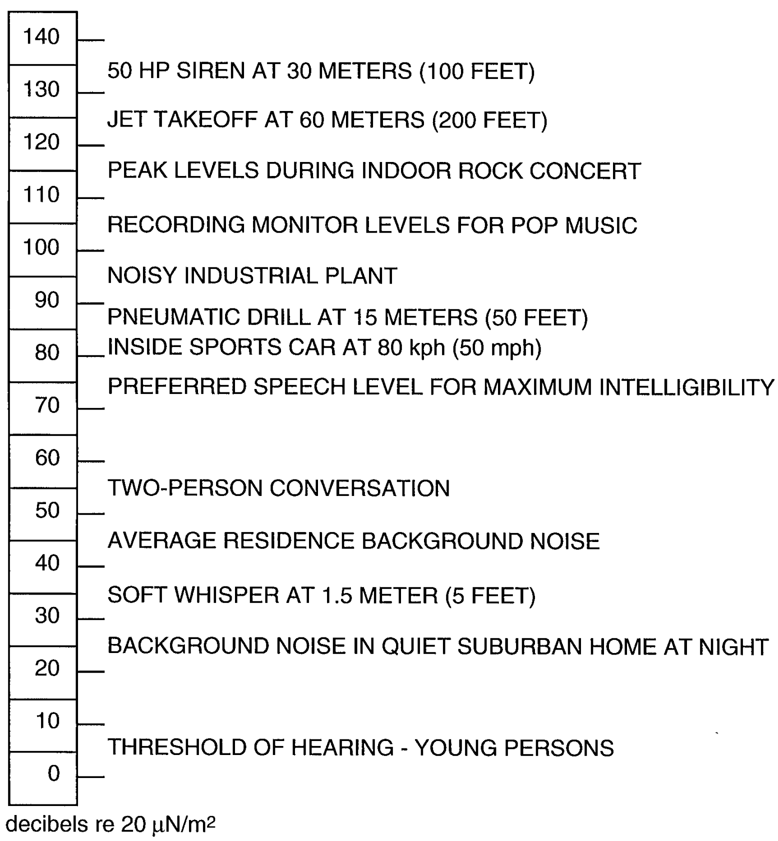


Figure 2-4. Typical A-weighted sound levels

Inverse Square Relationships

When we move away from a *point source* of sound out of doors, or in a *free field*, we observe that SPL falls off almost exactly 6 dB for each doubling of distance away from the source. The reason for this is shown in Figure 2-5. At A there is a sphere of radius one meter surrounding a point source of sound P_1 representing the SPL at the surface of the sphere. At B, we observe a sphere of twice the radius, 2 meters. The area of the larger sphere is *four times* that of the smaller one, and this means that the acoustical power passing through a small area on the larger sphere will be *one-fourth* that passing through the same small area on the smaller sphere. The 4-to-1 power ratio represents a level difference of 6 dB, and the corresponding sound pressure ratio will be 2-to-1.

A convenient nomograph for determining inverse square losses is given in Figure 2-6. Inverse square calculations depend on a theoretical point source in a free field. In the real world, we can closely approach an ideal free field, but we still must take into account the factors of finite source size and non-uniform radiation patterns.

Consider a horn-type loudspeaker having a rated sensitivity of 100 dB, 1 watt at 1 meter. One meter from where? Do we measure from the mouth of the horn, the throat of the horn, the driver diaphragm, or some indeterminate point in between? Even if the measurement position is specified, the information may be useless. Sound from a finite source does not behave according to inverse square law at distances close to that source. Measurements made in the "near field" cannot be used to estimate performance at greater distances. This being so, one may well wonder why loudspeakers are rated at a distance of only 1 meter.

The method of rating and the accepted methods of measuring the devices are two different things. The manufacturer is expected to make a number of measurements at various distances under free field conditions. From these he can establish

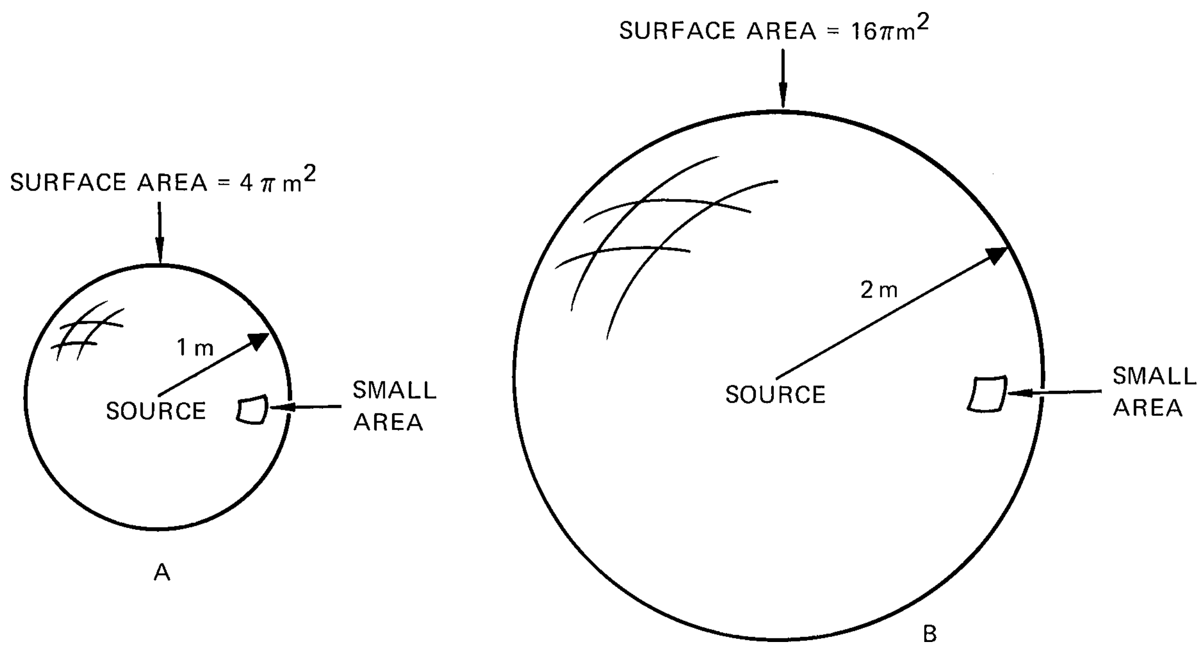


Figure 2-5. Inverse square relationships

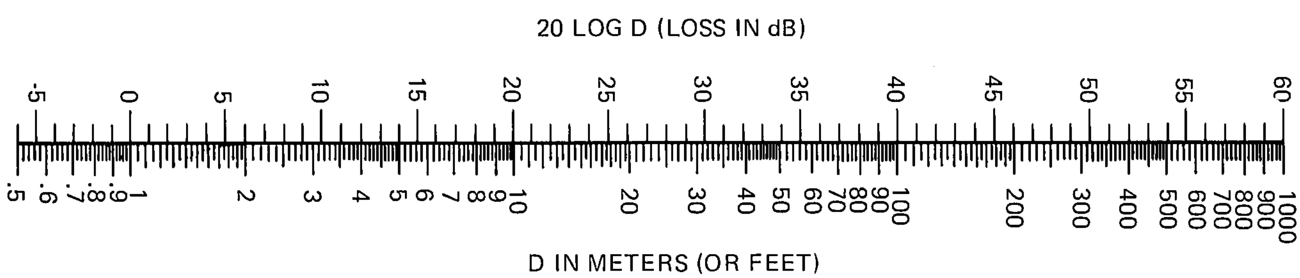


Figure 2-6. Nomograph for determining inverse square losses

that the measuring microphone is far enough away from the device to be in its *far field*, and he can also calculate the imaginary point from which sound waves diverge, according to inverse square law. This point is called the *acoustic center* of the device. After accurate field measurements have been made, the results are converted to an equivalent one meter rating. The rated sensitivity at one meter is that SPL which would be measured if the inverse square relationship were actually maintained that close to the device.

Let us work a few exercises using the nomograph of Figure 2-6:

1. A JBL model 2360 horn with a 2446 HF driver produces an output of 113 dB, 1 watt at 1 meter. What SPL will be produced by 1 watt at 30 meters? We can solve this by inspection of the nomograph. Simply read the difference in dB between 1 meter and 30 meters: 29.5 dB. Now, subtracting this from 113 dB:

$$113 - 29.5 = 83.5 \text{ dB}$$

2. The nominal power rating of the JBL model 2446 driver is 100 watts. What maximum SPL will be produced at a distance of 120 meters in a free field when this driver is mounted on a JBL model 2366 horn?

There are three simple steps in solving this problem. First, determine the inverse square loss from Figure 2-6; it is approximately 42 dB. Next, determine the level difference between one watt and 100 watts. From Figure 2-1 we observe this to be 20 dB. Finally, note that the horn-driver sensitivity is 118 dB, 1 watt at 1 meter. Adding these values:

$$118 - 42 + 20 = 96 \text{ dB-SPL}$$

Calculations such as these are very commonplace in sound reinforcement work, and qualified sound contractors should be able to make them easily.

Adding Power Levels in dB

Quite often, a sound contractor will have to add power levels expressed in dB. Let us assume that two sound fields, each 94 dB-SPL, are combined. What is the resulting level? If we simply add the levels numerically, we get 188 dB-SPL, clearly an absurd answer! What we must do in effect is convert the levels back to their actual powers, add them, and then recalculate the level in dB. Where two levels are involved, we can accomplish this easily with the data of Figure 2-7. Let *D* be the difference in dB between the two levels, and determine the value *N* corresponding to this difference. Now, add *N* to the *higher* of the two original values.

As an exercise, let us add two sound fields, 90 dB-SPL and 84 dB-SPL. Using Figure 2-7, a *D* of 6 dB corresponds to an *N* of about 1 dB. Therefore, the new level will be 91 dB-SPL.

Note that when two levels differ by more than about 10 dB, the resulting summation will be substantially the same as the higher of the two values. The effect of the lower level will be negligible.

Reference Levels

Although we have discussed some of the common reference levels already, we will list here all of those that a sound contractor is likely to encounter.

In acoustical measurements, *SPL* is always measured relative to 20 x 10^{-6} Pa. An equivalent expression of this is .0002 dynes/cm^2.

In broadcast transmission work, power is often expressed relative to 1 milliwatt (.001 watt), and such levels are expressed in *dBm*.

The designation *dBW* refers to levels relative to one watt. Thus, 0 dBW = 30 dBm.

In signal transmission diagrams, the designation *dBu* indicates voltage levels referred to .775 volts.

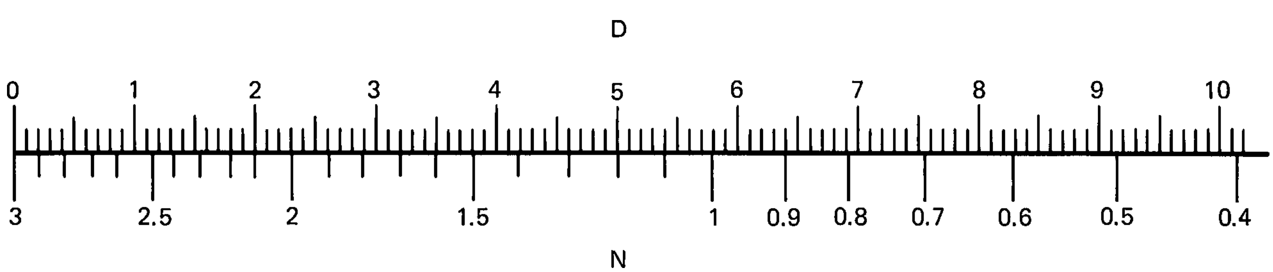


Figure 2-7. Nomograph for adding levels expressed in dB.
Summing sound level output of two sound sources where D is their output difference in dB.
N is added to the higher to derive the total level.

In other voltage measurements, *dBV* refers to levels relative to 1 volt.

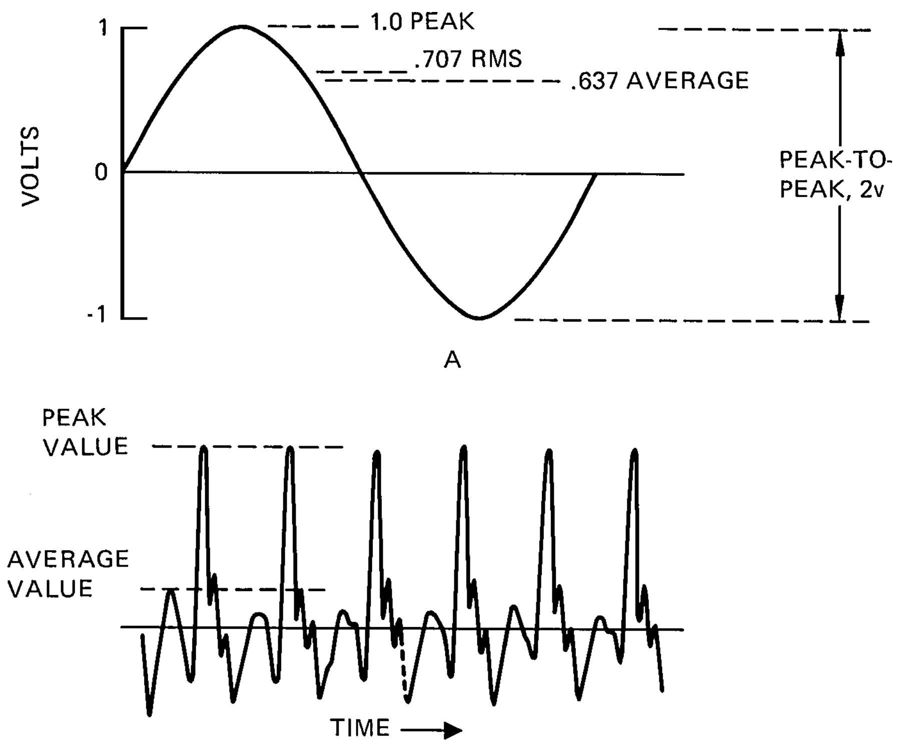
Rarely encountered by the sound contractor will be acoustical power levels. These are designated dB-PWL, and the reference power is 10^{-12} watts. This is a very small power indeed. It is used in acoustical measurements because such small amounts of power are normally encountered in acoustics.

Peak, Average, and *rms* Signal Values

Most measurements of voltage, current, or sound pressure in acoustical engineering work are given as *rms* (root mean square) values of the waveforms. The rms value of a repetitive waveform equals its equivalent DC value in power transmission. Referring to Figure 2-8A for a sine wave with a peak value of one volt, the rms value is .707 volt, a 3 dB difference. The average value of the waveform is .637 volt.

For more complex waveforms, such as are found in speech and music, the peak values will be considerably higher than the average or rms values. The waveform shown at Figure 2-8B is that of a trumpet at about 400 Hz, and the spread between peak and average values is 13 dB.

In this chapter, we have in effect been using rms values of voltage, current, and pressure for all calculations. However, in all audio engineering applications, the time-varying nature of music and speech demands that we consider as well the instantaneous values of waveforms likely to be encountered. The term *headroom* refers to the extra margin in dB designed into a signal transmission system over its normal operating level. The importance of headroom will become more evident as our course develops.



Waveform of a trumpet tone (about 400 Hz). The spread between average and peak values of the waveform is 13 dB.

B

Figure 2-8. Peak, average, and rms values. Sinewave (A); complex waveform (B).

Chapter 3: Directivity and Angular Coverage of Loudspeakers

Introduction

Proper coverage of the audience area is one of the prime requirements of a sound reinforcement system. What is required of the sound contractor is not only a knowledge of the directional characteristics of various components but also how those components may interact in a multi-component array. Such terms as directivity index (DI), directivity factor (Q), and beamwidth all variously describe the directional properties of transducers with their associated horns and enclosures. Detailed polar data, when available, gives the most information of all. In general, no one has ever complained of having too much directivity information. In the past, most manufacturers have supplied too little; however, things have changed for the better in recent years, largely through data standardization activities on the part of the Audio Engineering Society.

Some Fundamentals

Assume that we have an omnidirectional radiator located in free space and that there is a microphone at some fixed distance from it. This is shown in Figure 3-1A. Let the power radiated from the loudspeaker remain constant, and note the SPL at the mcrophone. Now, as shown at *B*, let us place a large reflecting boundary next to the source, cutting the radiation solid angle in half. We then note the SPL at the microphone has increased 6 dB, due to the “pressure doubling” effect of a reflected image of the source produced by the plane. We continue the process of successively halving the radiation solid angle at *C* and *D*, noting that, each time we do this, the level at the microphone increases 6 dB. Another way to view this phenomenon is to observe that, for each halving of the radiation solid angle, the source radiation impedance has been halved,

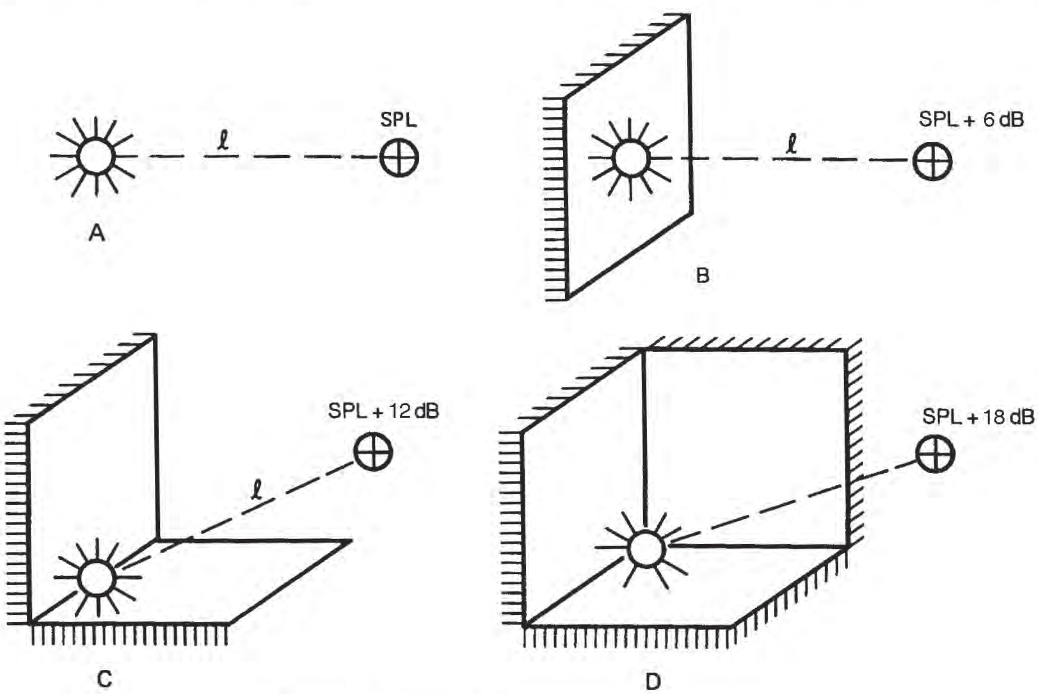


Figure 3-1. Directivity and angular coverage

causing a *doubling* of acoustical power. At the same time, the halving of the solid radiation angle causes a *doubling* of the directivity factor. Each of these effects results in a 3-dB increase in level, and thus a total 6 dB increase in SPL for each halving of the solid radiation angle will be observed. In going from A to D in successive steps we have increased the directivity index by 3 dB and doubled the directivity factor at each step.

Directivity index (DI) can be defined as the level difference along a given axis, and at a given distance, from a sound radiator compared to the intensity that would be produced by an omnidirectional source radiating the same power. Directivity factor (Q) is the ratio of the two values of intensity. These are shown graphically in Figure 3-2. *DI* and *Q* are related as follows:

$$DI = 10 \log Q$$

$$Q = 10^{DI/10}$$

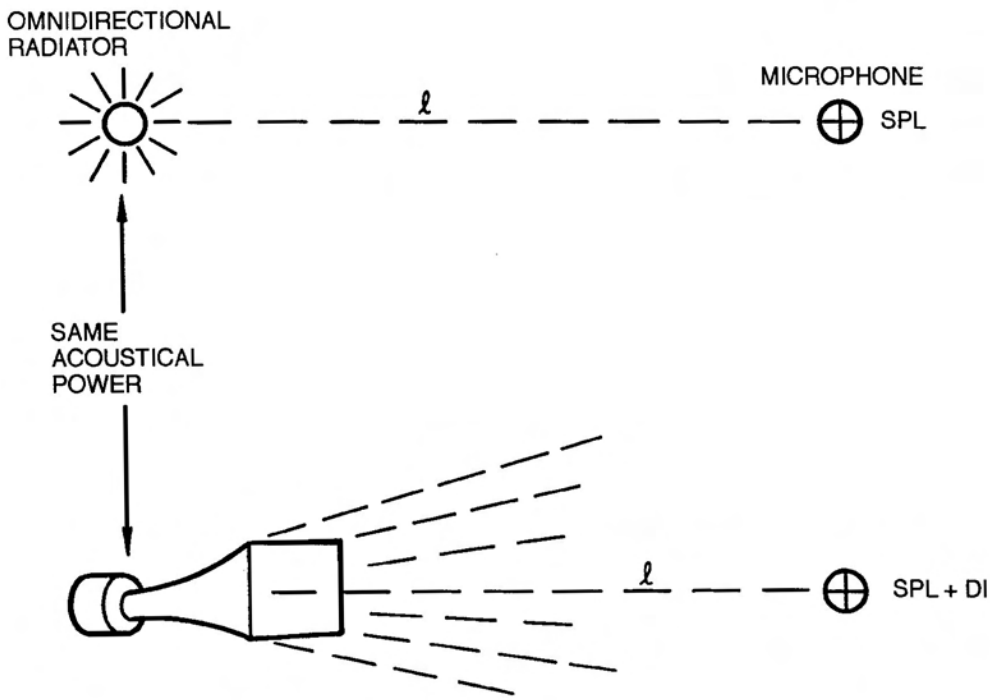


Figure 3-2. Directivity index and directivity factor

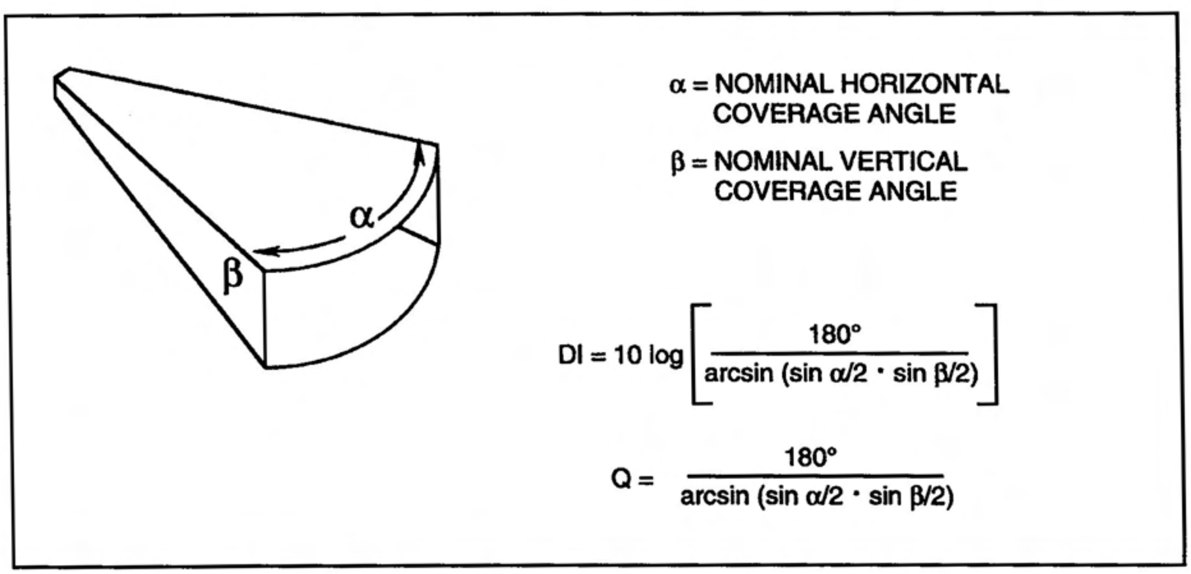


Figure 3-3. Illustration of Molloy's equation

The data of Figure 3-1 was generalized by Molloy (7) and is shown in Figure 3-3. Here, note that DI and Q are related to the solid angular coverage of a hypothetical sound radiator whose horizontal and vertical coverage angles are specified. Such ideal sound radiators do not exist, but it is surprising how closely these equations agree with measured DI and Q of HF horns that exhibit fairly steep cut-off outside their normal coverage angles.

As an example of this, a JBL model 2360 Bi-Radial horn has a nominal 90°-by-40° pattern measured between the 6 dB down points in each plane. If we insert the values of 90° and 40° into Molloy's equation, we get DI = 11 and Q = 12.8. The published values were calculated by integrating response over 360° in both horizontal and vertical planes, and they are DI = 10.8 and Q = 12.3. So the estimates are in excellent agreement with the measurements.

For the JBL model 2366 horn, with its nominal 6 dB down coverage angles of 40° and 20°, Molloy's equation gives DI = 17.2 and Q = 53. The published values are DI = 16.5 and Q = 46. Again, the agreement is excellent.

Is there always such good correlation between the 6 dB down horizontal and vertical beamwidth of a horn and its calculated directivity? The answer is *no*. Only when the response cut-off is sharp beyond the 6 dB beamwidth limits and when there is minimal radiation outside rated beamwidth will the correlation be good. For many types of radiators, especially those operating at wavelengths large compared with their physical dimensions, Molloy's equation will not hold.

A Comparison of Polar Plots, Beamwidth Plots, Directivity Plots, and Isobars

There is no one method of presenting directional data on radiators which is complete in all regards. Polar plots (Figure 3-4A) are normally presented in only the horizontal and vertical planes. A single polar plot covers only a single frequency, or frequency band, and a complete set of polar plots takes up considerable space. Polars are, however, the only method of presentation giving a clear picture of a radiator's response outside its normal operating beamwidth. Beamwidth plots of the 6 dB down coverage angles (Figure 3-4B) are very common because considerable information is contained in a single plot. By itself, a plot of DI or Q conveys information only about the on-axis performance of a radiator (Figure 3-4C). Taken together, horizontal and vertical beamwidth plots and DI or Q plots convey sufficient information for most sound reinforcement design requirements.

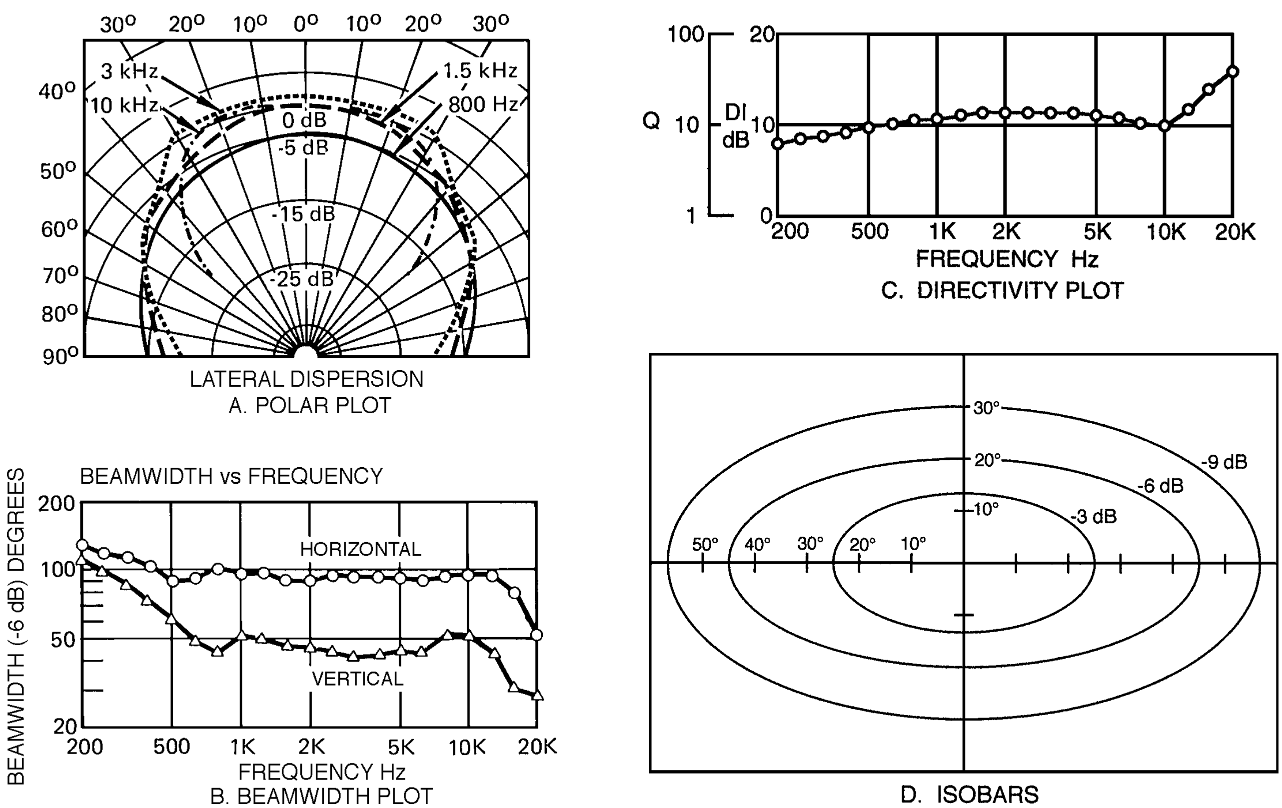


Figure 3-4. Methods of presenting directional information

Isobars have become popular in recent years. They give the angular contours in spherical coordinates about the principal axis along which the response is -3, -6, and -9 dB, relative to the on-axis maximum. It is relatively easy to interpolate visually between adjacent isobars to arrive at a reasonable estimate of relative response over the useful frontal solid radiation angle of the horn. Isobars are useful in advanced computer layout techniques for determining sound coverage over entire seating areas. The normal method of isobar presentation is shown in Figure 3-4D.

Still another way to show the directional characteristics of radiators is by means of a family of off-axis frequency response curves, as shown in Figure 3-5. At *A*, note that the off-axis response curves of the JBL model 2360 Bi-Radial horn run almost parallel to the on-axis response curve. What this means is that a listener seated off the main axis will perceive smooth response when a Bi-Radial constant coverage horn is used. Contrast this with the off-axis response curves of the older (and obsolete) JBL model 2350 radial horn shown at *B*. If this device is equalized for flat on-axis response, then listeners off-axis will perceive rolled-off HF response.

Directivity of Circular Radiators

Any radiator has little directional control for frequencies whose wavelengths are large compared with the radiating area. Even when the radiating area is large compared to the wavelength, constant pattern control will not result unless the device has been specifically designed to maintain a constant pattern. Nothing demonstrates this better than a simple radiating piston. Figure 3-6 shows the sharpening of on-axis response of a piston mounted in a flat baffle. The wavelength varies over a 24-to-1 range. If the piston were, say a 300 mm (12”) loudspeaker, then the wavelength illustrated in the figure would correspond to frequencies spanning the range from about 350 Hz to 8 kHz.

Among other things, this illustration points out why “full range,” single-cone loudspeakers are of little use in sound reinforcement engineering. While the on-axis response can be maintained through equalization, off-axis response falls off drastically above the frequency whose wavelength is about equal to the diameter of the piston. Note that when the diameter equals the wavelength, the radiation pattern is approximately a 90° cone with - 6 dB response at ±45°.

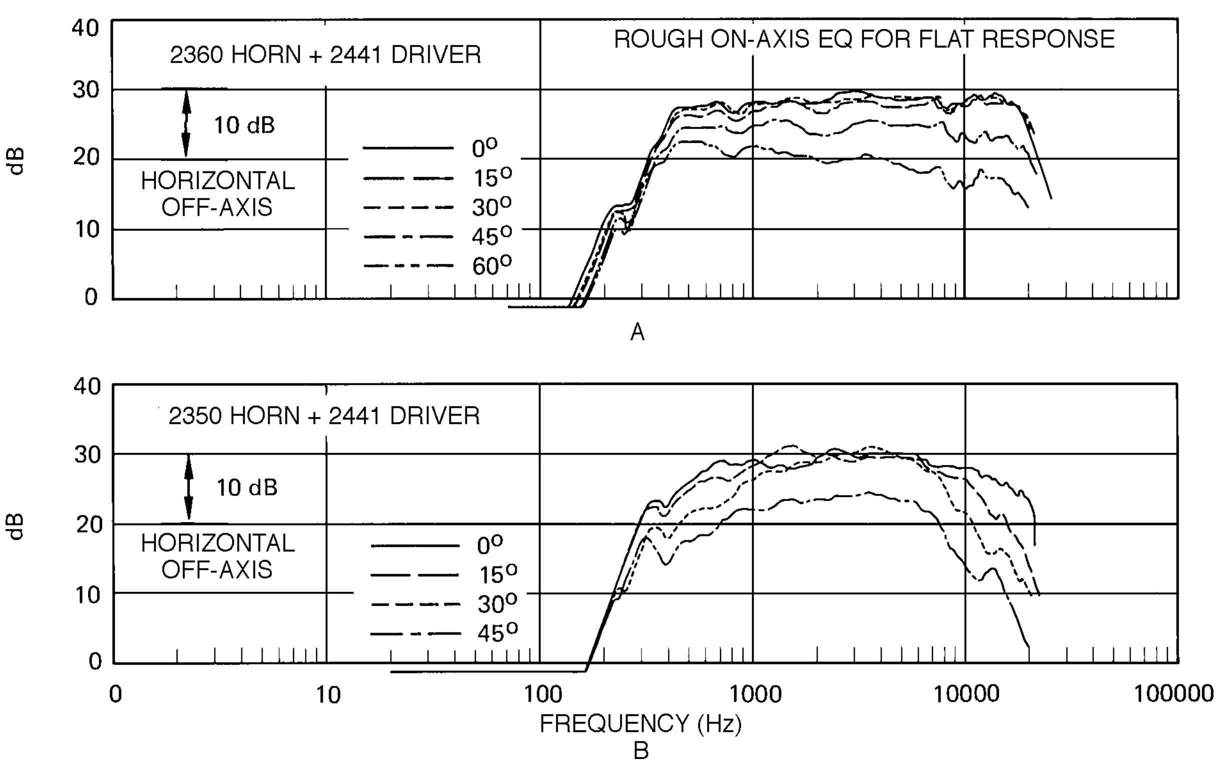


Figure 3-5. Families of off-axis frequency response curves

The values of DI and Q given in Figure 3-6 are the on-axis values, that is, along the axis of maximum loudspeaker sensitivity. This is almost always the case for published values of DI and Q. However, values of DI and Q exist along *any* axis of the radiator, and they can be determined by inspection of the polar plot. For example, in Figure 3-6, examine the polar plot corresponding to Diameter = l. Here, the on-axis DI is 10 dB. If we simply move off-axis to a point where the response has dropped 10 dB, then the DI along that direction will be 10 - 10, or 0 dB, and the Q will be unity. The off-axis angle where the response is 10 dB down is marked on the plot and is at about 55°. Normally, we will not be concerned with values of DI and Q along axes other than the principal one; however, there are certain calculations involving interaction of microphones and loudspeakers where a knowledge of off-axis directivity is essential.

Omnidirectional microphones with circular diaphragms respond to on- and off-axis signals in a manner similar to the data shown in Figure 3-6. Let us assume that a given microphone has a diaphragm about 25 mm (1") in diameter. The frequency corresponding to l/4 is about 3500 Hz, and the response will be quite smooth both on and off axis. However, by the time we reach 13 or 14 kHz, the diameter of the diaphragm is about equal to l, and the DI of the microphone is about 10 dB. That is, it will be 10 dB more sensitive to sounds arriving on axis than to sounds which are randomly incident to the microphone.

Of course, a piston is a very simple radiator — or receiver. Horns such as JBL's Bi-Radial series are complex by comparison, and they have been designed to maintain constant HF coverage through attention to wave-guide principles in their design. One thing is certain: no radiator can exhibit much pattern control at frequencies whose wavelengths are much larger than the circumference of the radiating surface.

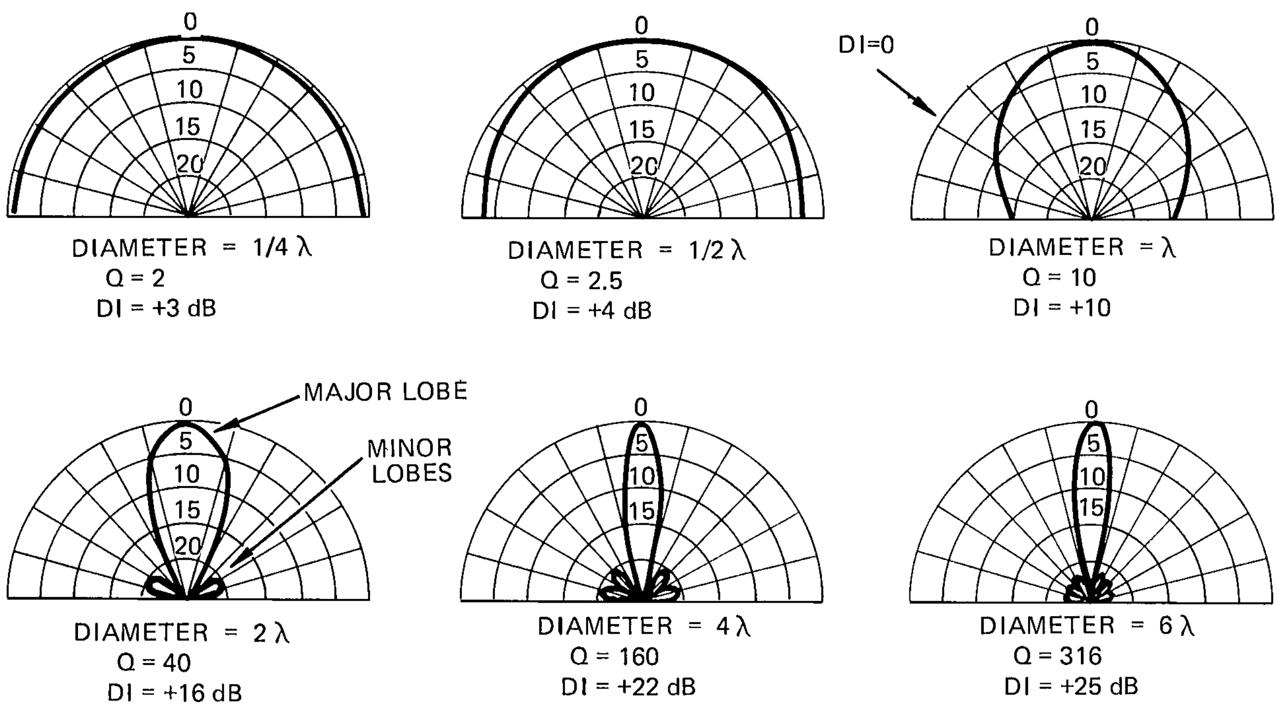


Figure 3-6. Directional characteristics of a circular-piston source mounted in an infinite baffle as a function of diameter and l.

The Importance of Flat Power Response

If a radiator exhibits flat power response, then the power it radiates, integrated over all directions, will be constant with frequency. Typical compression drivers inherently have a rolled-off response when measured on a *plane wave tube* (PWT), as shown in Figure 3-7A. When such a driver is mounted on a typical radial horn such as the JBL model 2350, the on-axis response of the combination will be the sum of the PWT response and the DI of the horn. Observe at *B* that the combination is fairly flat on axis and does not need additional equalization. Off-axis response falls off, both vertically and horizontally, and the total power response of the combination will be the same as observed on the PWT; that is, it rolls off above about 3 kHz.

Now, let us mount the same driver on a Bi-Radial uniform coverage horn, as shown at *C*. Note that both on-and off-axis response curves are rolled off but run parallel with each other. Since the DI of the horn is essentially flat, the on-axis response will be virtually the same as the PWT response.

At *D*, we have inserted a HF boost to compensate for the driver's rolled off power response, and the result is now flat response both on and off axis. Listeners anywhere in the area covered by the horn will appreciate the smooth and extended response of the system.

Flat power response makes sense only with components exhibiting constant angular coverage. If we had equalized the 2350 horn for flat power response, then the on-axis response would have been too bright and edgy sounding.

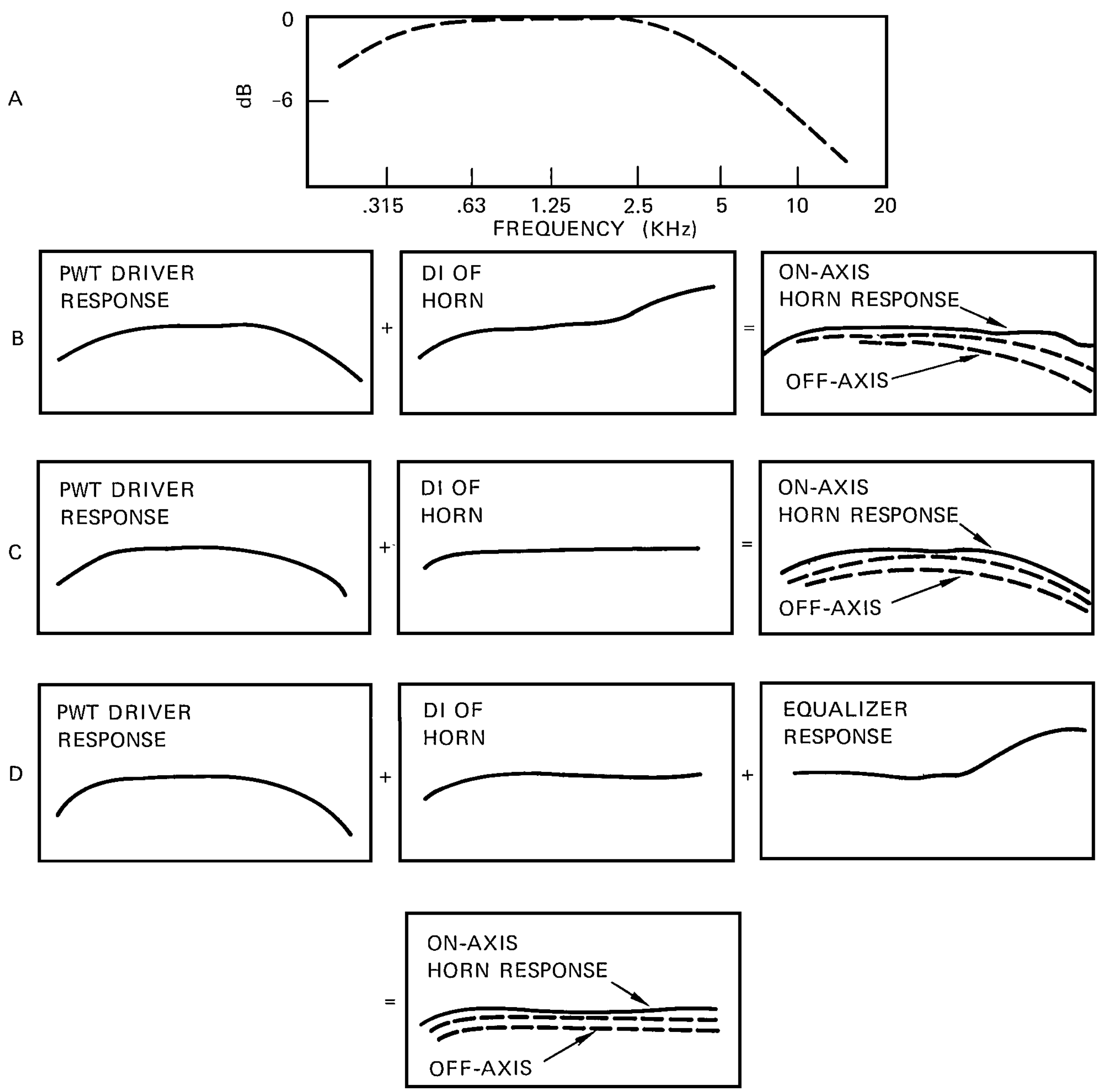


Figure 3-7. Power response of HF systems

The rising DI of most typical radial horns is accomplished through a narrowing of the vertical pattern with rising frequency, while the horizontal pattern remains fairly constant, as shown in Figure 3-8A. Such a horn can give excellent horizontal coverage, and since it is "self equalizing" through its rising DI, there may be no need at all for external equalization. The smooth-running horizontal and vertical coverage angles of a Bi-Radial, as shown at Figure 3-8B, will always require power response HF boosting.

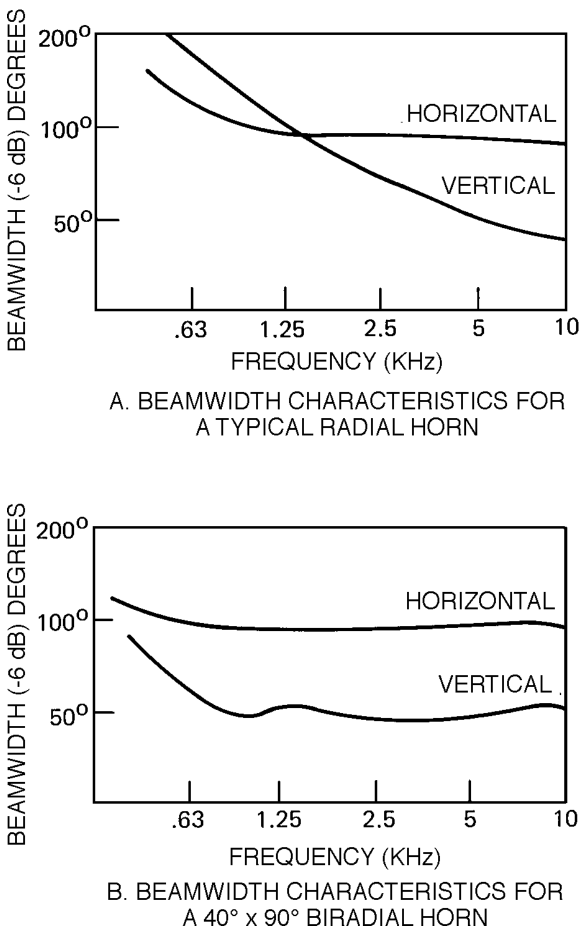


Figure 3-8. Increasing DI through narrowing vertical beamwidth

Measurement of Directional Characteristics

Polar plots and isobar plots require that the radiator under test be rotated about several of its axes and the response recorded. Beamwidth plots may be taken directly from this data.

DI and Q can be calculated from polar data by integration using the following equation:

$$DI = 10 \log \left[\frac{2}{\int_o^{\pi} (P_\theta)^2 \sin\theta d\theta} \right]$$

P_Q is taken as unity, and *q* is taken in 10° increments. The integral is solved for a value of DI in the horizontal plane and a value in the vertical plane. The resulting DI and Q for the radiator are given as:

$$DI = \frac{DI_h}{2} + \frac{DI_v}{2}$$

and

$$Q = \sqrt{Q_n \cdot Q_v}$$

(Note: There are slight variations of this method, and of course all commonly use methods are only approximations in that they make use of limited polar data.)

Using Directivity Information

A knowledge of the coverage angles of an HF horn is essential if the device is to be oriented properly with respect to an audience area. If polar plots or isobars are available, then the sound contractor can make calculations such as those indicated in Figure 3-9. The horn used in this example is the JBL 2360 Bi-Radial. We note from the isobars for this horn that the -3 dB angle off the vertical is 14°. The -6 dB and -9 dB angles are 23° and 30° respectively. This data is for the octave band centered at 2 kHz. The horn is aimed so that its major axis is pointed at the farthest seats. This will ensure maximum reach, or “throw,” to those seats. We now look at the -3 dB angle of the horn and compare the reduction in the horn’s output along that angle with the inverse square advantage at the closer-in seats covered along that axis. Ideally, we would like for the inverse square advantage to exactly match the horn’s off-axis fall-off, but this is not always possible. We similarly look at the response along the -6 and -9 dB axes of the horn,

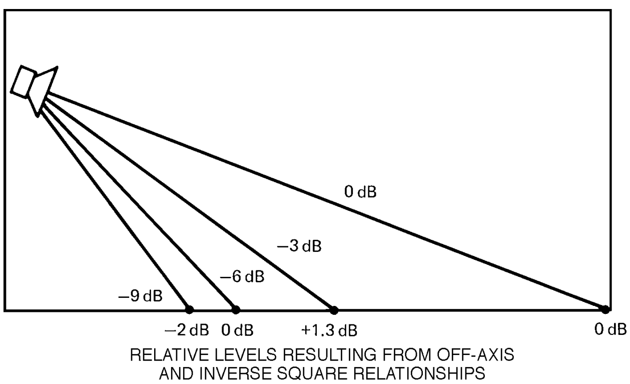


Figure 3-9. Off-axis and inverse square calculations

comparing them with the inverse square advantages afforded by the closer-in seats. When the designer has flexibility in choosing the horn’s location, a good compromise, such as that shown in this figure, will be possible. Beyond the -9 dB angle, the horn’s output falls off so rapidly that additional devices, driven at much lower levels, would be needed to cover the front seats (often called “front fill” loudspeakers).

Aiming a horn as shown here may result in a good bit of power being radiated toward the back wall. Ideally, that surface should be fairly absorptive so that reflections from it do not become a problem.

Directional Characteristics of Combined Radiators

While manufacturers routinely provide data on their individual items of hardware, most provide little, if any, data on how they interact with each other. The data presented here for combinations of HF horns is of course highly wavelength, and thus size, dependent. Appropriate scaling must be done if this data is to be applied to larger or smaller horns.

In general, at high frequencies, horns will act independently of each other. If a pair of horns are properly splayed so that their -6 dB angles just overlap, then the response along that common axis should be smooth, and the effect will be nearly that of a single horn with increased coverage in the plane of overlap. Thus, two horns with 60° coverage in the horizontal plane can be splayed to give 120° horizontal coverage. Likewise, dissimilar horns can be splayed, with a resulting angle being the sum of the two coverage angles in the plane of the splay. Splaying may be done in the vertical plane with similar results. Figure 3-10 presents an example of horn splaying in the horizontal plane.

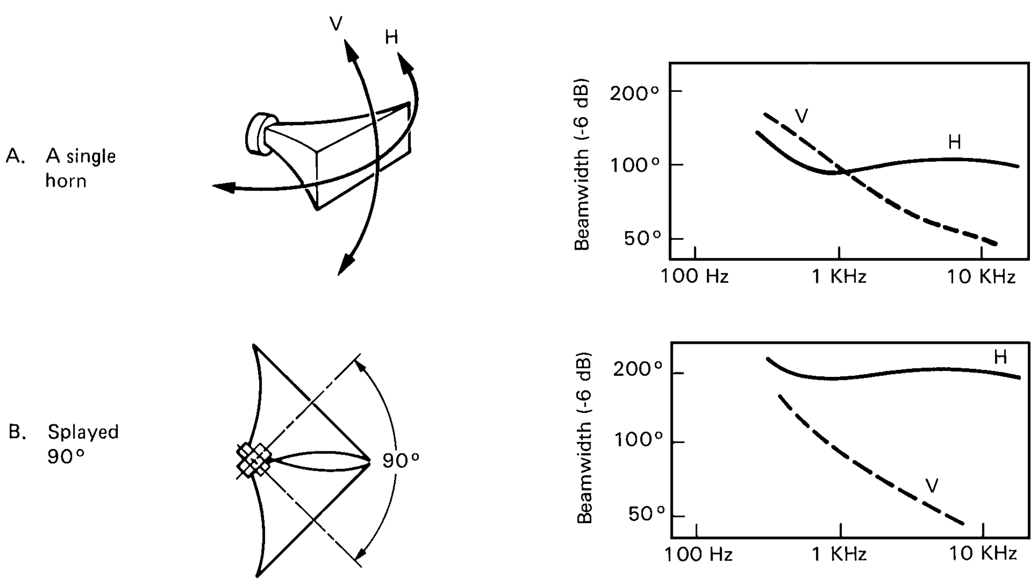


Figure 3-10. Horn splaying for wider coverage

Horns may be stacked in a vertical array to improve pattern control at low frequencies. The JBL Flat-Front Bi-Radials, because of their relatively small vertical mouth dimension, exhibit a broadening in their vertical pattern control below about 2 kHz. When used in vertical stacks of three or four units, the effective vertical mouth dimension is much larger than that of a single horn. The result, as shown in Figure 3-11, is tighter pattern control down to about 500 Hz. In such vertical in-line arrays, the resulting horizontal pattern is the same as for a single horn. Additional details on horn stacking are given in Technical Note Volume 1, Number 7.

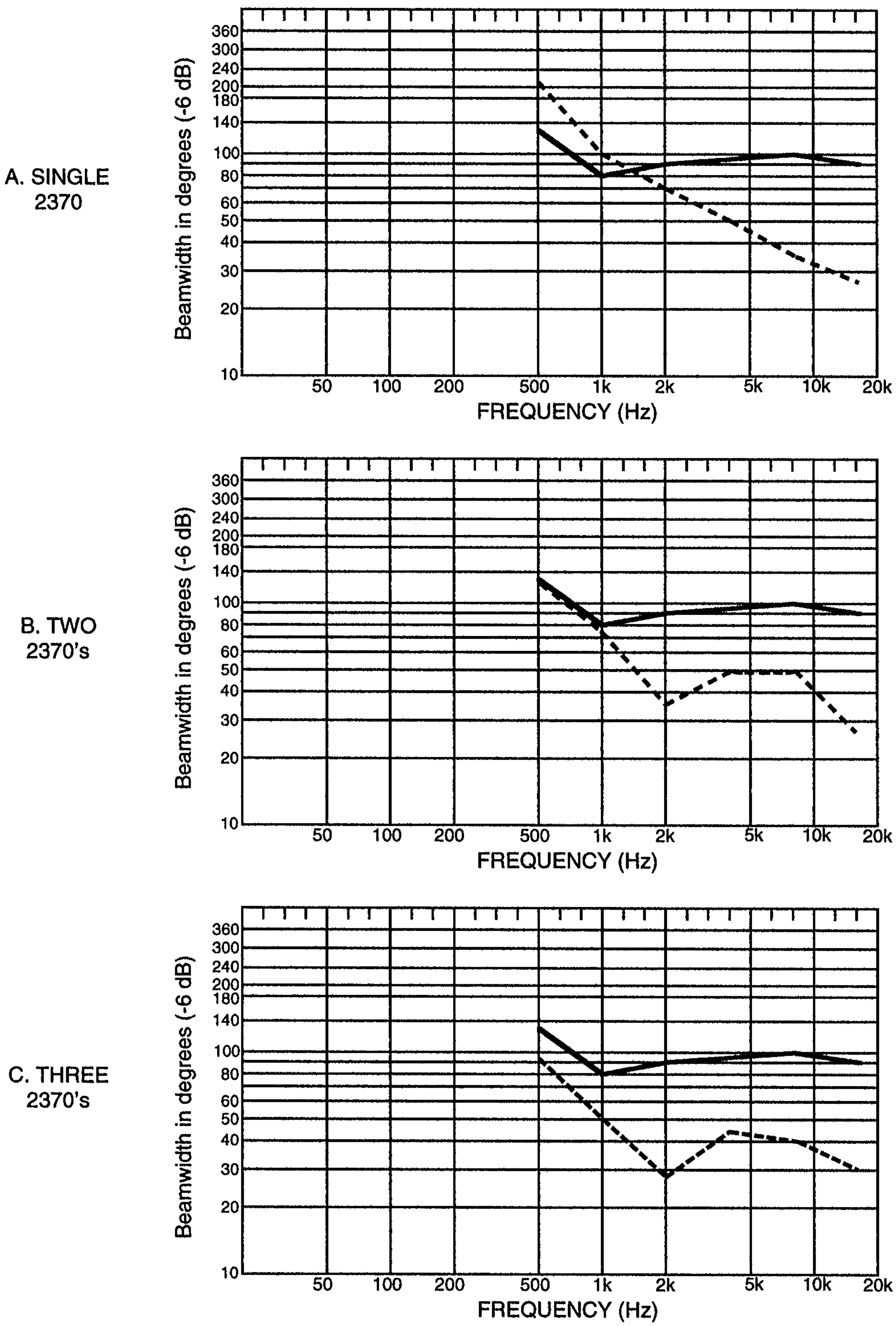


Figure 3-1 1. Stacking horns for higher directivity at low frequencies (solid line, horizontal -6 dB deamwidth, dashed line, vertical -6 dB beamwidth)

Chapter 4: An Outdoor Sound Reinforcement System

Introduction

Our study of sound reinforcement systems begins with an analysis of a simple outdoor system. The outdoor environment is relatively free of reflecting surfaces, and we will make the simplifying assumption that free field conditions exist. A basic reinforcement system is shown in Figure 4-1A. The essential acoustical elements are the talker, microphone, loudspeaker, and listener. The electrical diagram of the system is shown at *B*. The dotted line indicates the acoustical feedback path which can exist around the entire system.

When the system is turned on, the gain of the amplifier can be advanced up to some point at which the system will “ring,” or go into feedback. At the onset of feedback, the gain around the electro-acoustical path is unity and at a zero phase angle. This condition is shown at *C*, where the input at the microphone of a single pulse will give rise to a repetitive signal at the microphone, fed back from the loudspeaker and which will quickly give rise to sustained oscillation at a single frequency with a period related to Dt.

Even at levels somewhat below feedback, the response of the system will be irregular, due to the fact that the system is “trying” to go into feedback, but does not have enough loop gain to sustain it. This is shown in Figure 4-2. As a rule, a workable reinforcement system should have a gain margin of 6 to 10 dB before feedback if it is to sound natural on all types of program input.

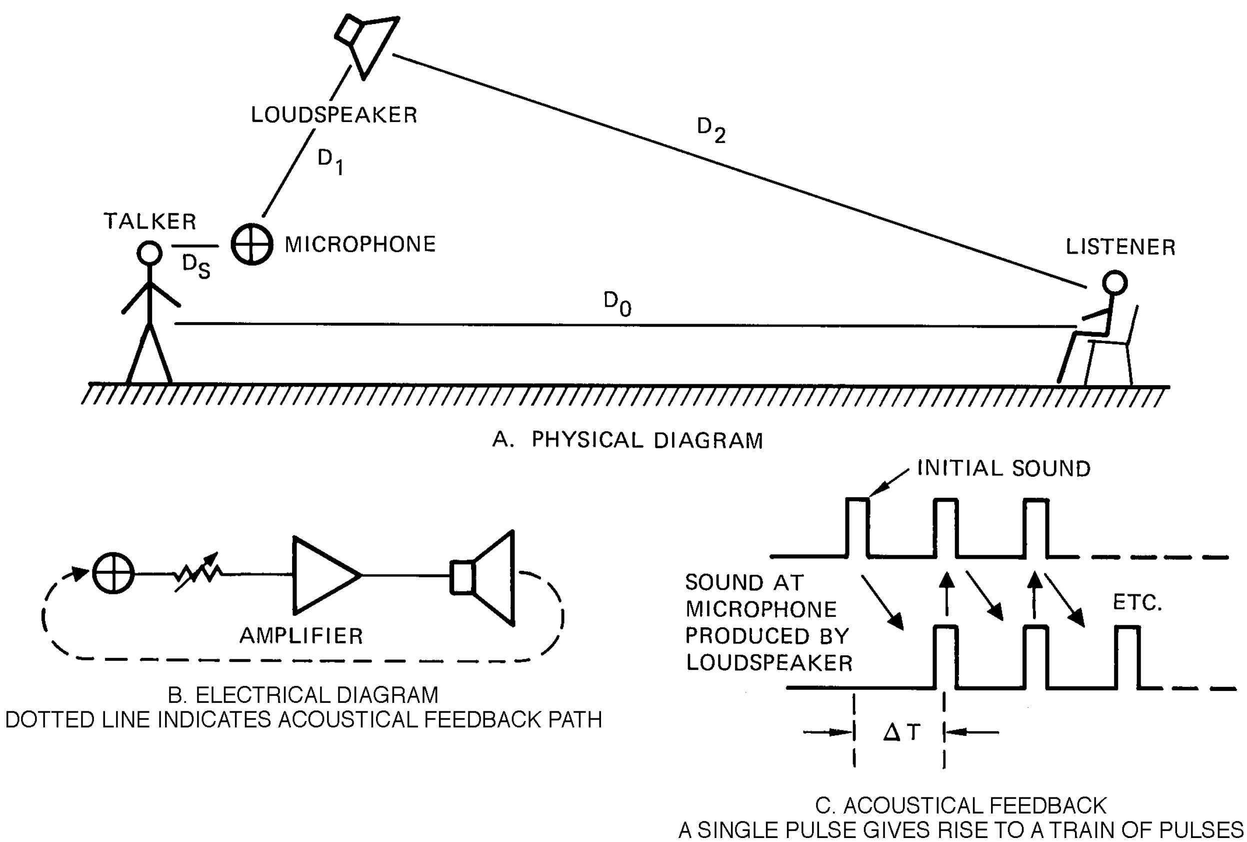


Figure 4-1. A simple outdoor reinforcement system

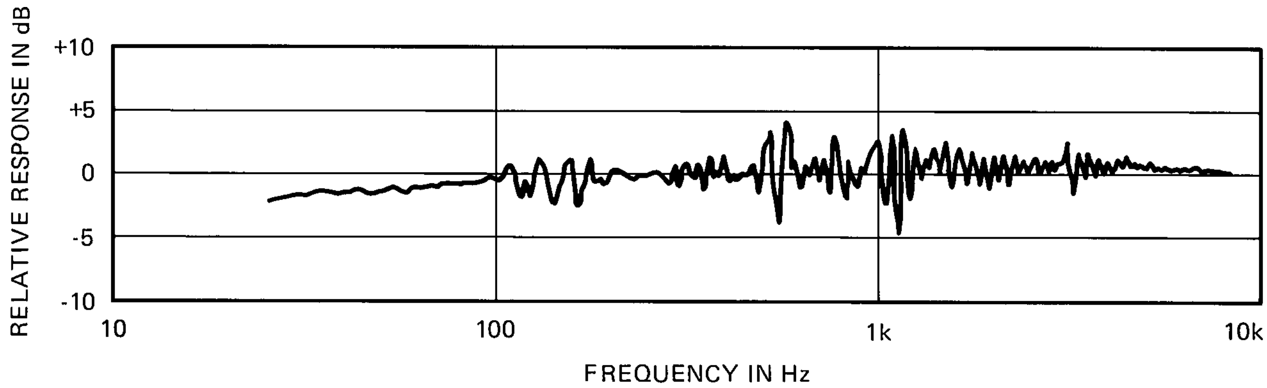


Figure 4-2. Electrical response of a sound system 3 dB below sustained acoustical feedback

The Concept of Acoustical Gain

Boner (4) quantified the concept of acoustical gain, and we will now present its simple but elegant derivation. Acoustical gain is defined as the increase in level that a given listener in the audience perceives with the system turned on, as compared to the level the listener hears directly from the talker when the system is off.

Referring to Figure 4-3, let us assume that both the loudspeaker and microphone are omnidirectional; that is, DI = 0 dB and Q = 1. Then by inverse square loss, the level at the listener will be:

$$70 \text{ dB} - 20 \log (7/1) = 70 - 17 = 53 \text{ dB}$$

Now, we turn the system on and advance the gain until we are just at the onset of feedback. This will occur when the loudspeaker, along the D_1 path, produces a level at the microphone equal to that of the talker, 70 dB.

If the loudspeaker produces a level of 70 dB at the microphone, it will produce a level at the listener of:

$$70 - 20 \log (6/4) = 70 - 3.5 = 66.5 \text{ dB}$$

With no safety margin, the maximum gain this system can produce is:

$$66.5 - 53 = 13.5 \text{ dB}$$

Rewriting our equations:

$$\text{Maximum gain} = 70 - 20 \log (D_2/D_1) - 70 - 20 \log (D_0/D_s)$$

This simplifies to:

$$\text{Maximum gain} = 20 \log D_0 - 20 \log D_s + 20 \log D_1 - 20 \log D_2$$

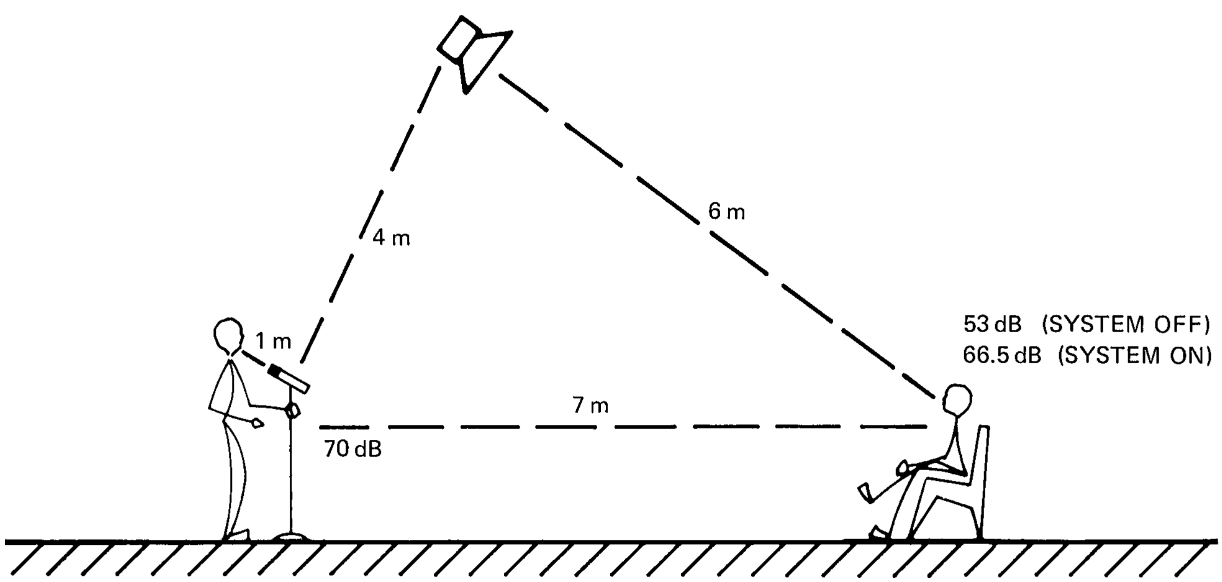


Figure 4-3. System gain calculations, loudspeaker and microphone both omnidirectional

Adding a 6 dB safety factor gives us the usual form of the equation:

Maximum gain =

$$20 \log D_0 - 20 \log D_s + 20 \log D_1 - 20 \log D_2 - 6$$

In this form, the gain equation tells us several things, some of them intuitively obvious:

1. That gain is independent of the level of the talker
2. That decreasing D_s will increase gain
3. That increasing D_1 will increase gain.

The Influence of Directional Microphones and Loudspeakers on System Maximum Gain

Let us rework the example of Figure 4-3, this time making use of a directional loudspeaker whose midband polar characteristics are as shown in Figure 4-4A. It is obvious from looking at Figure 4-4A that sound arriving at the microphone along the D_1 direction will be reduced 6 dB relative to the omnidirectional loudspeaker. This 6 dB results directly in added gain potential for the system.

The same holds for directional microphones, as shown in Figure 4-5A. In Figure 4-5B, we show a system using an omnidirectional loudspeaker and a cardioid microphone with its -6 dB axis facing toward the loudspeaker. This system is equivalent to the one shown in Figure 4-4B; both exhibit a 6 dB increase in maximum gain over the earlier case where both microphone and loudspeaker were omnidirectional.

Finally, we can use both directional loudspeakers and microphones to pick up additional gain. We simply calculate the maximum gain using omnidirectional elements, and then add to that value the off-axis pattern advantage in dB for both loudspeaker and microphone. As a practical matter, however, it is not wise to rely too heavily on directional microphones and loudspeakers to make a significant increase in system gain. Most designers are content to realize no more than 4-to-6 dB overall added gain from the use of directional elements. The reason for this is that microphone and loudspeaker directional patterns are not constant with frequency. Most directional loudspeakers will, at low frequencies, appear to be nearly omnidirectional. If more gain is called for, the most straightforward way to get it is to reduce D_s or increase D_1.

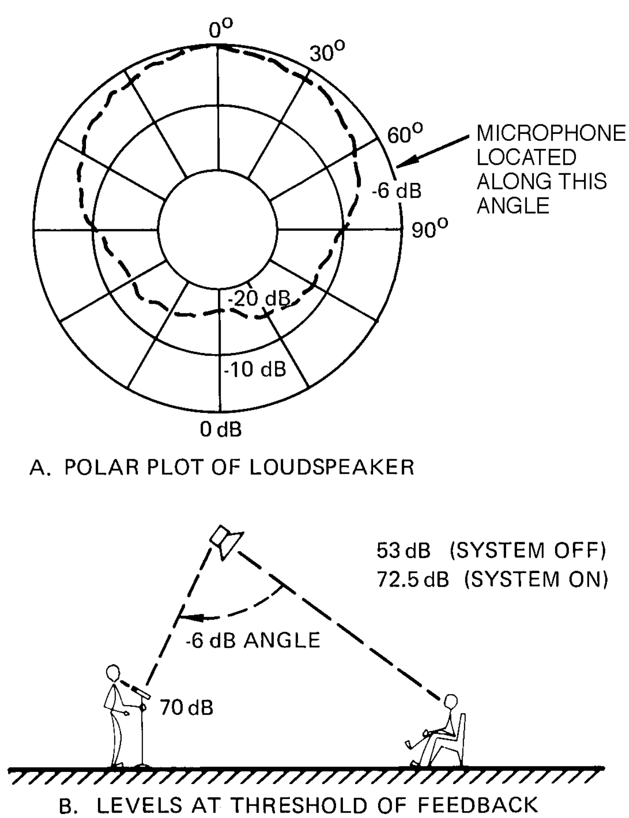


Figure 4-4. System gain calculations, directional loudspeaker

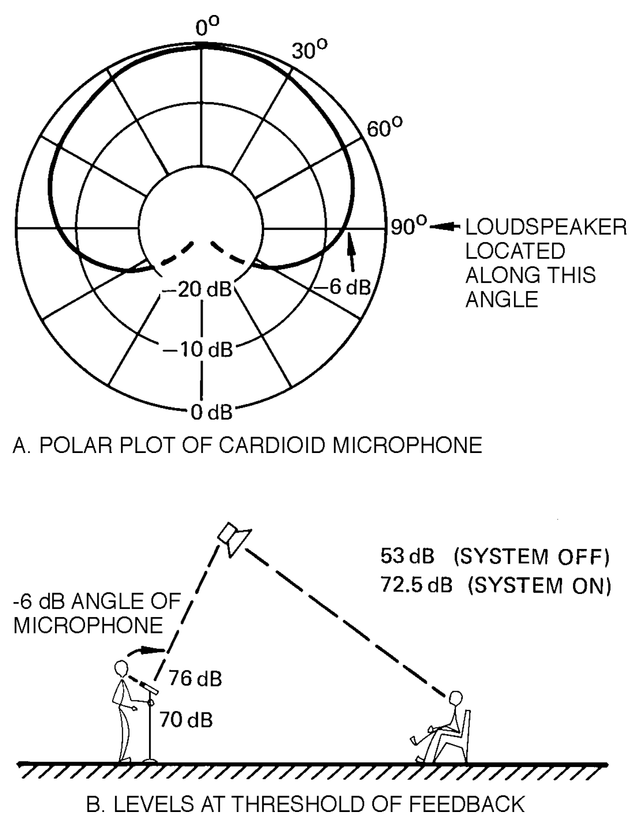


Figure 4-5. System gain calculations, directional microphone

How Much Gain is Needed?

The parameters of a given sound reinforcement system may be such that we have more gain than we need. When this is the case, we simply turn things down to a comfortable point, and everyone is happy. But things often do not work out so well. What is needed is some way of determining beforehand how much gain we will need so that we can avoid specifying a system which will not work. One way of doing this is by specifying the *equivalent*, or *effective, acoustical distance (EAD),* as shown in Figure 4-6. Sound reinforcement systems may be thought of as effectively moving the talker closer to the listener. In a quiet environment, we may not want to bring the talker any closer than, say, 3 meters from the listener. What this means, roughly, is that the loudness produced by the reinforcement system should approximate, for a listener at D_0, the loudness level of an actual talker at a distance of 3 meters. The gain necessary to do this is calculated from the inverse square relation between D_0 and *EAD*:

$$\text{Necessary gain} = 20 \log D_0 - 20 \log \text{EAD}$$

In our earlier example, D_0 = 7 meters. Setting EAD = 3 meters, then:

$$\text{Necessary gain} = 20 \log (7) - 20 \log (3) = 17 - 9.5 = 7.5 \text{ dB}$$

Assuming that both loudspeaker and microphone are omnidirectional, the maximum gain we can expect is:

$$\text{Maximum gain} = 20 \log (7) - 20 \log (1) + 20 \log (4) - 20 \log (6) - 6$$

$$\text{Maximum gain} = 17 - 0 + 12 - 15.5 - 6$$

$$\text{Maximum gain} = 7.5 \text{ dB}$$

As we can see, the necessary gain and the maximum gain are both 7.5 dB, so the system will be workable. If, for example, we were specifying a system for a noisier environment requiring a shorter *EAD*, then the system would not have sufficient gain. For example, a new *EAD* of 1.5 meters would require 6 dB more acoustical gain. As we have discussed, using a directional microphone and a directional loudspeaker would just about give us the needed 6 dB. A simpler, and better, solution would be to reduce D_s to 0.5 meter in order to get the added 6 dB of gain.

In general, in an outdoor system, satisfactory articulation will result when speech peaks are about 25 dB higher than the A-weighted ambient noise level. Typical conversation takes place at levels of 60 to 65 dB at a distance of one meter. Thus, in an ambient noise field of 50 dB, we would require speech peaks of 75 to 80 dB for comfortable listening, and this would require an EAD as close as 0.25 meter, calculated as follows:

Speech level at 1 meter = 65 dB

Speech level at 0.5 meter = 71 dB

Speech level at 0.25 meter = 77 dB

Let us see what we must do to our outdoor system to make it work under these demanding conditions. First, we calculate the necessary acoustical gain:

$$\text{Necessary gain} = 20 \log D_0 - 20 \log \text{EAD}$$

$$\text{Necessary gain} = 20 \log (7) - 20 \log (.25)$$

$$\text{Necessary gain} = 17 + 12 = 29 \text{ dB}$$

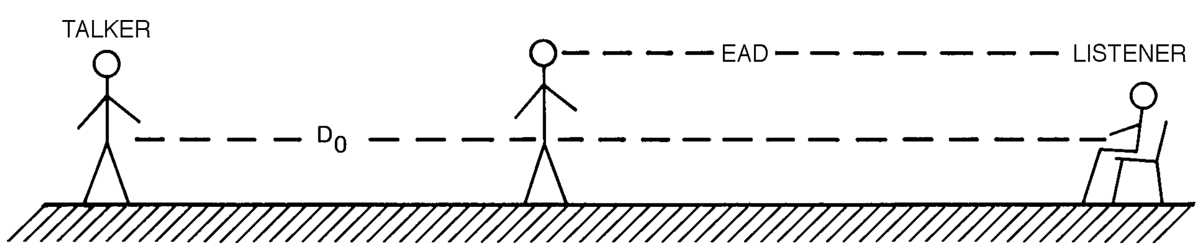


Figure 4-6. Concept of Effective Acoustical Dustance (EAD)

As we saw in an earlier example, our system only has 7.5 dB of maximum gain available with a 6 dB safety factor. By going to both a directional microphone and a directional loudspeaker, we can increase this by about 6 dB, yielding a maximum gain of 13.5 dB — still some 16 dB short of what we actually need.

The solution is obvious; a hand-held microphone will be necessary in order to achieve the required gain. For 16 dB of added gain, D_s will have to be reduced to the value calculated below:

$$16 = 20 \log (1/x)$$

$$16/20 = \log (1/x)$$

$$10^{.8} = 1/x$$

Therefore: $x = 1/10^{.8} = 0.16$ meter (6”)

Of course, the problem with a hand-held microphone is that it is difficult for the user to maintain a fixed distance between the microphone and his mouth. As a result, the gain of the system will vary considerably with only small changes in the performer-microphone operating distance. It is always better to use some kind of personal microphone, one worn by the user. In this case, a swivel type microphone attached to a headpiece would be best, since it provides the minimum value of D_s. This type of microphone is now becoming very popular on-stage, largely because a number of major pop and country artists have adopted it. In other cases a simple tietack microphone may be sufficient.

Conclusion

In this chapter, we have presented the rudiments of gain calculation for sound systems, and the methods of analysis form the basis for the study of indoor systems, which we will cover in a later chapter.

Chapter 5: Fundamentals of Room Acoustics

Introduction

Most sound reinforcement systems are located indoors, and the acoustical properties of the enclosed space have a profound effect on the system's requirements and its performance. Our study begins with a discussion of sound absorption and reflection, the growth and decay of sound fields in a room, reverberation, direct and reverberant sound fields, critical distance, and room constant.

If analyzed in detail, any enclosed space is quite complex acoustically. We will make many simplifications as we construct "statistical" models of rooms, our aim being to keep our calculations to a minimum, while maintaining accuracy on the order of 10%, or ±1 dB.

Absorption and Reflection of Sound

Sound tends to "bend around" non-porous, small obstacles. However, large surfaces such as the boundaries of rooms are typically partially flexible and partially porous. As a result, when sound strikes such a surface, some of its energy is reflected, some is absorbed, and some is transmitted through the boundary and again propagated as sound waves on the other side. See Figure 5-1.

All three effects may vary with frequency and with the angle of incidence. In typical situations, they *do not* vary with sound intensity. Over the range of sound pressures commonly encountered in audio work, most construction materials have the same characteristics of reflection, absorption and transmission whether struck by very weak or very strong sound waves.

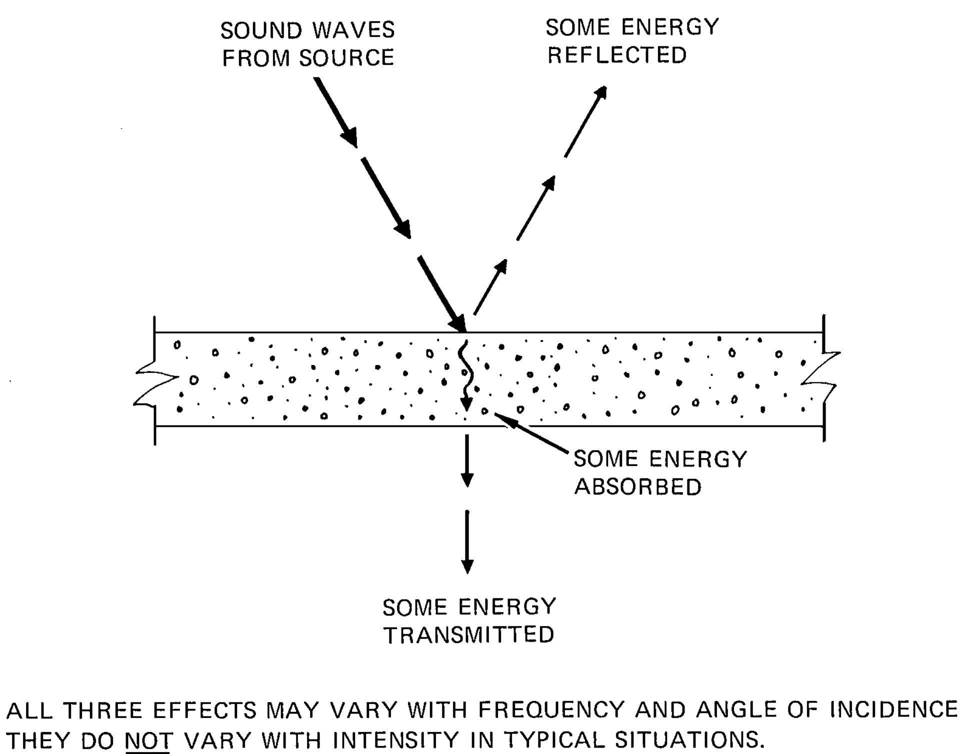


Figure 5-1. Sound impinging on a large boundary surface

When dealing with the behavior of sound in an enclosed space, we must be able to estimate how much sound energy will be lost each time a sound wave strikes one of the boundary surfaces or one of the objects inside the room. Tables of absorption coefficients for common building materials as well as special "acoustical" materials can be found in any architectural acoustics textbook or in data sheets supplied by manufacturers of construction materials.

Unless otherwise specified, published sound absorption coefficients represent average absorption over all possible angles of incidence. This is desirable from a practical standpoint since the random incidence coefficient fits the situation that exists in a typical enclosed space where sound waves rebound many times from each boundary surface in virtually all possible directions.

Absorption ratings normally are given for a number of different frequency bands. Typically, each band of frequencies is one octave wide, and standard center frequencies of 125 Hz, 250 Hz, 500 Hz, 1 kHz, etc., are used. In sound system design, it usually is sufficient to know absorption characteristics of materials in three or four frequency ranges. In this handbook, we make use of absorption ratings in the bands centered at 125 Hz, 1 kHz and 4 kHz.

The effects of mounting geometry are included in standardized absorption ratings by specifying the types of mounting according to an accepted numbering system. In our work, familiarity with at least three of these standard mountings is important.

Acoustical tile or other interior material cemented directly to a solid, non-absorptive surface is called "No. 1" mounting (see Figure 5-2). To obtain greater absorption, especially at lower frequencies, the material may be spaced out on nominal two-inch thick furring strips and the cavity behind loosely filled with fiberglass blanket. This type of mounting is called out as "No. 2". "No. 7" mounting is the familiary suspended "T"-bar ceiling system. Here the material is spaced at least 0.6 meter (2') away from a solid structural boundary.

Absorption coefficients fall within a scale from zero to one following the concept established by Sabine, the pioneer of modern architectural acoustics. Sabine suggested that an open window be considered a perfect absorber (since no sound is reflected) and that its sound absorption coefficient must therefore be 100 percent, or unity. At the other end of the scale, a material which reflects all sound and absorbs none has an absorption coefficient of zero.

In older charts and textbooks, the total absorption in a room may be given in sabins. The *sabin* is a unit of absorption named after Sabine and is the equivalent of one square foot of open window. For example, suppose a given material has an absorption coefficient of 0.1 at 1 kHz. One hundred square feet of this material in a room has a total absorption of 10 sabins. (Note: When using SI units, the *metric sabin* is equal to one square meter of totally absorptive surface.)

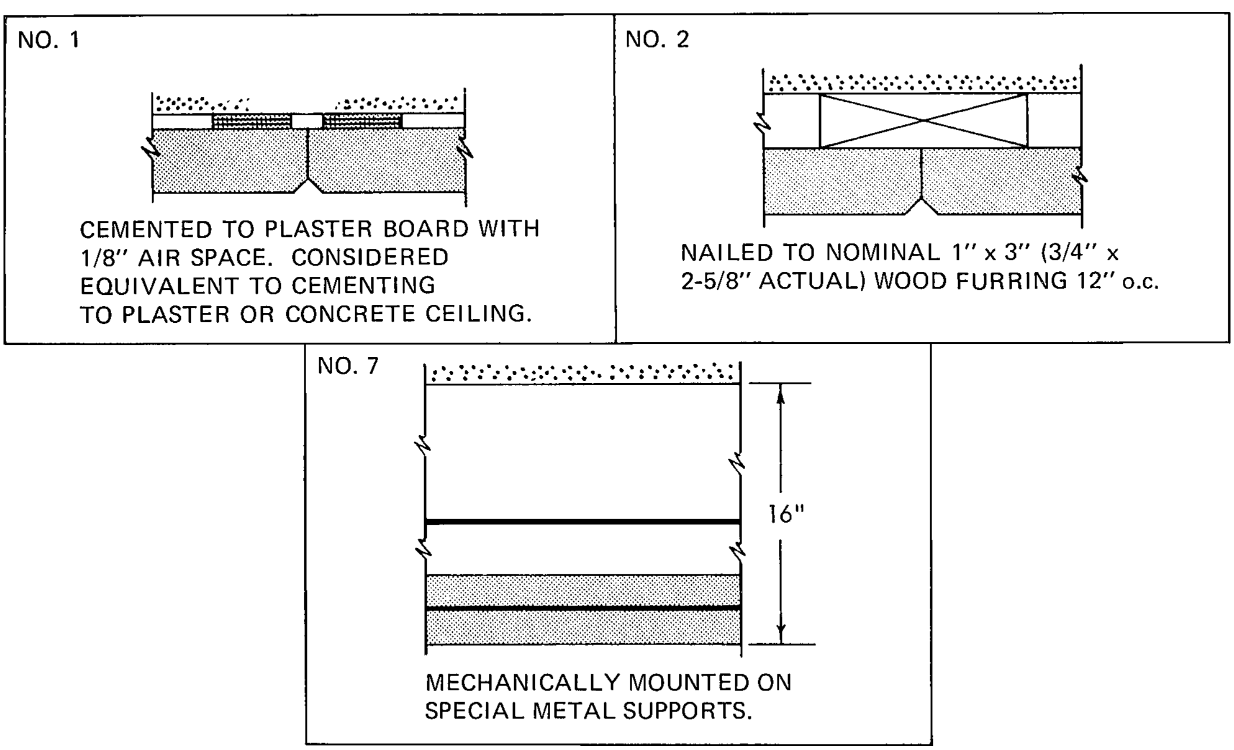


Figure 5-2. ASTM types of mounting (used in conducting sound absorption tests)

More recent publications usually express the absorption in an enclosed space in terms of the *average absorption coefficient*. For example, if a room has a total surface area of 1000 square meters consisting of 200 square meters of material with an absorption coefficient of .8 and 800 square meters of material with an absorption coefficient of .1, the average absorption coefficient for the entire internal surface area of the room is said to be .24:

Area:		Coefficient:	Sabins:
200	x	0.8	= 160
800	x	0.1	= 80
1000			240

$$\bar{a} = \frac{240}{1000} = 0.24$$

The use of the average absorption coefficient $\bar{a}$ has the advantage that it is not tied to any particular system of measurement. An average absorption coefficient of 0.15 is exactly the same whether the surfaces of the room are measured in square feet, square yards, or square meters. It also turns out that the use of an average absorption coefficient facilitates solving reverberation time, direct-to-reverberant sound ratio, and steady-state sound pressure.

ABSORP. COEFF. a	REFL. COEFF. $1-a$ (γ)	REFL. COEFF. dB
.01	.99	–.044
.02	.98	–.088
.03	.97	–.13
.04	.96	–.18
.05	.95	–.22
.06	.94	–.27
.07	.93	–.32
.08	.92	–.36
.09	.91	–.41
.10	.90	–.46
.20	.80	–.97
.30	.70	–1.5
.40	.60	–2.2
.50	.50	–3.0
.60	.40	–4.0
.70	.30	–5.2
.80	.20	–7.0
.90	.10	–10.0
.95	.05	–13.0

Figure 5-3. Reflection coefficient in decibels as a function of absorption coefficient

Although we commonly use published absorption coefficients without questioning their accuracy and perform simple arithmetic averaging to compute the average absorption coefficient of a room, the numbers themselves and the procedures we use are only approximations. While this does not upset the reliability of our calculations to a large degree, it is important to realize that the limit of confidence when working with published absorption coefficients is probably somewhere in the neighborhood of ±10%.

How does the absorption coefficient of the material relate to the intensity of the reflected sound wave? An absorption coefficient of 0.2 at some specified frequency and angle of incidence means that 20% of the sound energy will be absorbed and the remaining 80% reflected. The conversion to decibels is a simple 10 log function:

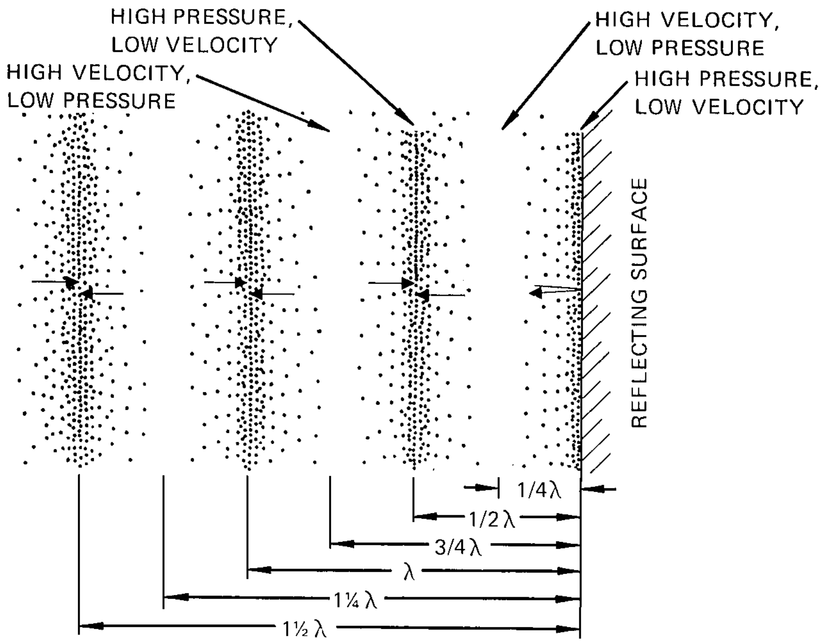
$$10 \log_{10} 0.8 = -0.97 \text{ dB}$$

In the example given, the ratio of reflected to direct sound energy is about -1 dB. In other words, the reflected wave is 1 dB weaker than it would have been if the surface were 100% reflective. See the table in Figure 5-3.

Thinking in terms of decibels can be of real help in a practical situation. Suppose we want to improve the acoustics of a small auditorium which has a pronounced “slap” off the rear wall. To reduce the intensity of the slap by only 3 dB, the wall must be surfaced with some material having an absorption coefficient of 0.5! To make the slap half as loud (a reduction of 10 dB) requires acoustical treatment of the rear wall to increase its absorption coefficient to 0.9. The difficulty is heightened by the fact that most materials absorb substantially less sound energy from a wave striking head-on than their random incidence coefficients would indicate.

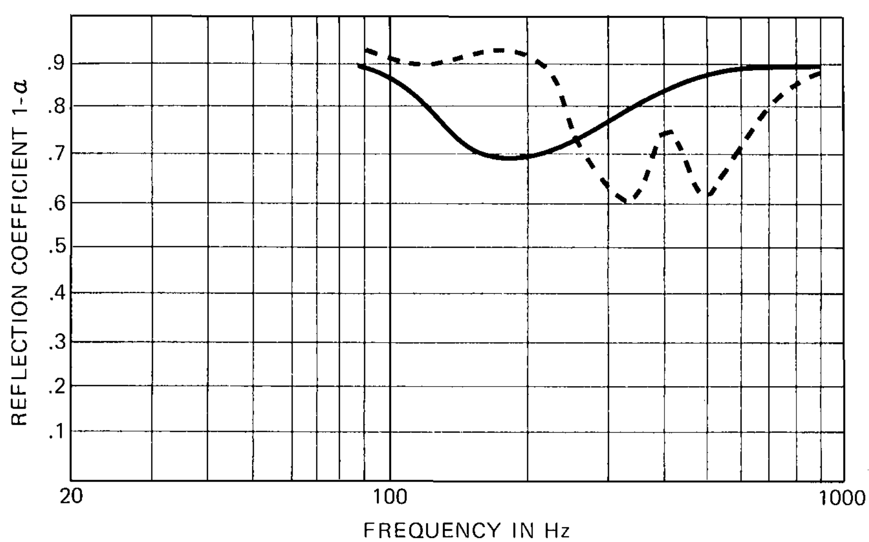
Most “acoustic” materials are porous. They belong to the class which acousticians elegantly label “fuzz”. Sound is absorbed by offering resistance to the flow of air through the material and thereby changing some of the energy to heat.

But when porous material is affixed directly to solid concrete or some other rigid non-absorptive surface, it is obvious that there can be no air motion and therefore no absorption at the boundary of the two materials.



A PLANE WAVE REFLECTED FROM A PLANE SURFACE AT NORMAL INCIDENCE PRODUCES WELL DEFINED ZONES OF HIGH PRESSURE ALTERNATING WITH ZONES OF HIGH PARTICLE VELOCITY AT DISTANCES OF ONE-QUARTER WAVELENGTH

Figure 5-4. Interference pattern of sound reflected from a solid boundary



SOLID LINE – 1/8" PLYWOOD
DOTTED LINE – 1/16" PLYWOOD

PANELS UNBACKED (NO ABSORPTIVE BLANKET) WITH 1¼" AIR SPACE.

(CHART SHOWS REFLECTION COEFFICIENT RATHER THAN ABSORPTION COEFFICIENT TO CONFORM WITH NORMAL FREQUENCY RESPONSE CURVES IN WHICH "UP" MEANS MORE LEVEL RATHER THAN MORE ATTENUATION.)

Figure 5-5. Reflectivity of thin plywood panels

Consider a sound wave striking such a boundary at normal incidence, shown in Figure 5-4. The reflected energy leaves the boundary in the opposite direction from which it entered and combines with subsequent sound waves to form a classic standing wave pattern. Particle velocity is very small (theoretically zero) at the boundary of the two materials and also at a distance 1/2 wavelength away from the boundary. Air particle velocity is at a maximum at 1/4 wavelength from the boundary. From this simple physical relationship it seems obvious that unless the thickness of the absorptive material is appreciable in comparison with a quarter wavelength, its effect will be minimal.

This physical model also explains the dramatic increase in absorption obtained when a porous material is spaced away from a boundary surface. By spacing the layer of absorptive material exactly one-quarter wavelength away from the wall, where particle velocity is greatest, its effective absorption is multiplied many times. The situation is complicated by the necessity of considering sound waves arriving from all possible directions. However, the basic effect remains the same: porous materials can be made more effective by making them thicker or by spacing them away from non-absorptive boundary surfaces.

A thin panel of wood or other material also absorbs sound, but it must be free to vibrate. As it vibrates in response to sound pressure, frictional losses change some of the energy into heat and sound is thus absorbed. Diaphragm absorbers tend to resonate at a particular band of frequencies, as any other tuned circuit, and they must be used with care. Their great advantage is the fact that low frequency absorption can be obtained in less depth than would be required for porous materials. See Figure 5-5.

A second type of tuned absorber occasionally used in acoustical work is the Helmholtz resonator: a reflex enclosure without a loudspeaker. (A patented construction material making use of this type of absorption is called “Soundblox”. These masonry blocks containing sound absorptive cavities can be used in gymnasiums, swimming pools, and other locations in which porous materials cannot be employed.)

The Growth and Decay of a Sound Field in a Room

At this point we should have sufficient understanding of the behavior of sound in free space and the effects of large boundary surfaces to understand what happens when sound is confined in an enclosure. The equations used to describe the behavior of sound systems in rooms all involve considerable “averaging out” of complicated phenomena. Our calculations, therefore, are made on the basis of what is typical or normal; they do not give precise answers for particular cases. In most situations, we can estimate with a considerable degree of confidence, but if we merely plug numbers into equations without understanding the underlying physical processes, we may find ourselves making laborious calculations on the basis of pure guesswork without realizing it.

Suppose we have an omnidirectional sound source located somewhere near the center of a room. The source is turned on and from that instant sound radiates outward in all directions at 344 meters per second (1130 feet per second) until it strikes the boundaries of the room. When sound strikes a boundary surface, some of the energy is absorbed, some is transmitted through the boundary and the remainder is reflected back into the room where it travels on a different course until another reflection occurs. After a certain length of time, so many reflections have taken place that the sound field is now a random jumble of waves traveling in all directions throughout the enclosed space.

If the source remains on and continues to emit sound at a steady rate, the energy inside the room builds up until a state of equilibrium is reached in which the sound energy being pumped into the room from the source exactly balances the sound energy dissipated through absorption and transmission through the boundaries. Statistically, all of the individual sound packets of varying intensities and varying directions can be averaged out, and at all points in the room not too close to the source or any of the boundary surfaces, we can say that a uniform diffuse sound field exists.

The geometrical approach to architectural acoustics thus makes use of a sort of “soup” analogy. As long as a sufficient number of reflections have taken place, and as long as we can disregard such anomalies as strong focused reflections, prominent resonant frequencies, the direct field near the source, and the strong possibility that all room surfaces do not have the same absorption characteristics, this statistical model may be used to describe the sound field in an actual room. In practice, the approach works remarkably well. If one is careful to allow for some of the factors mentioned, theory allows us to make simple calculations regarding the behavior of sound in rooms and arrive at results sufficiently accurate for most noise control and sound system calculations.

Going back to our model, consider what happens when the sound source is turned off. Energy is no longer pumped into the room. Therefore, as a certain amount of energy is lost with each reflection, the energy density of the sound field gradually decreases until all of the sound has been absorbed at the boundary surfaces.

Figure 5-6 gives a simple picture of this in idealized form. In the left graph, the vertical axis represents total sound energy in the room and the horizontal axis represents some convenient time scale. From the instant the sound source is turned on, the total energy in the room increases until it gradually levels off at a steady state value. Once this balance has been achieved, the sound source is turned off and the total energy in the room decreases until all of it has been absorbed. Note that in this idealized picture, growth and decay are exponential functions. The curve is exactly the same as the familiar graph of the charging and discharging of the capacitor.

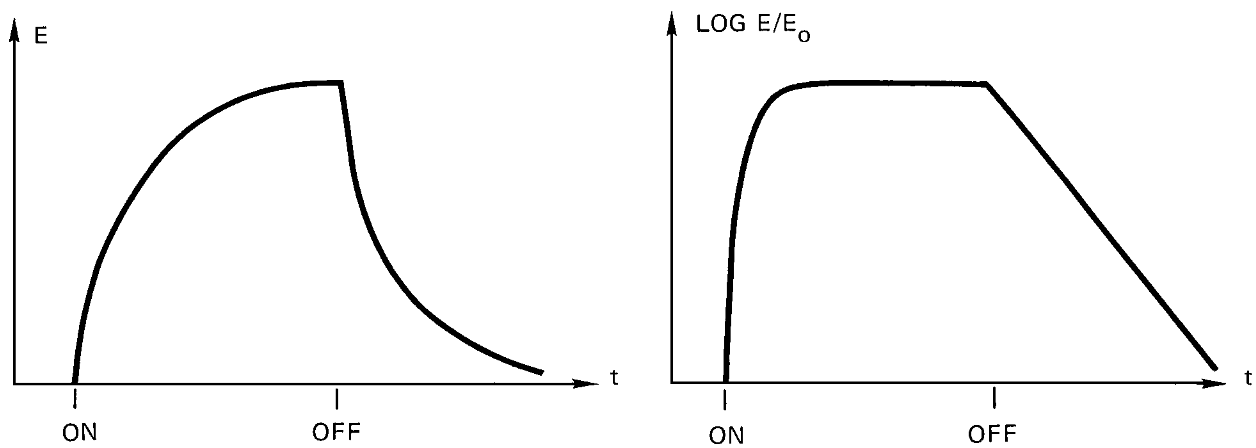


Figure 5-6. Idealized growth and decay of sound energy in an enclosure

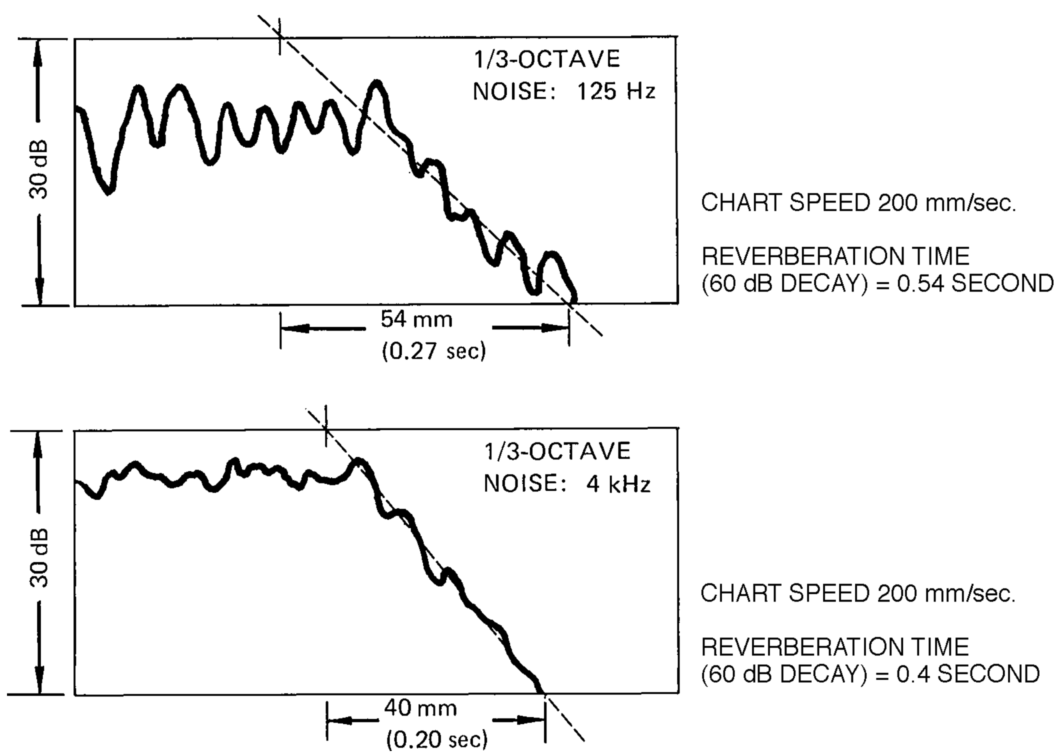


Figure 5-7. Actual chart recordings of decay of sound in a room

It is easier for us to comprehend this theoretical state of affairs if energy growth and decay are plotted on a decibel scale. This is what has been done in the graph. In decibel relationships, the growth of sound is very rapid and decay becomes a straight line. The slope of the line represents the rate of decay in decibels per second.

How closely does the behavior of sound in a real room approach this statistical picture? Figure 5-7 shows actual chart recordings of the decay of sound in a fairly absorptive room. Each chart was made by using a one-third octave band of random noise as the test signal. A sound level meter was located in the reverberant sound field. (In practice several readings would be taken at a number of different locations in the room).

The upper graph illustrates a measurement made in the band centered at 125 Hz. Note the great fluctuations in the steady state level and similar fluctuations as the sound intensity decreases. The fluctuations are sufficiently great to make any “exact” determination of the decay rate impossible. Instead, a straight line which seems to represent the “best fit” is drawn and its slope measured. In this case, the slope of the line is such that sound pressure seems to be decaying at a rate of 30 dB per 0.27 seconds. This works out to a decay rate of 111 dB per second.

The lower chart shows a similar measurement taken with the one-third octave band centered at 4 kHz. The fluctuations in level are not as pronounced, and it is much easier to arrive at what seems to be the correct slope of the sound decay. In this instance sound pressure appears to be decreasing at a rate of 30 dB in 0.2 seconds, or a decay rate of 150 dB per second.

Reverberation and Reverberation Time

The term *decay rate* is relatively unfamiliar; usually we talk about *reverberation time*. Originally, reverberation time was described simply as the length of time required for a very loud sound to die away to inaudibility. It was later defined in more specific terms as the actual time required for sound to decay 60 decibels. In both definitions it is assumed that decay rate is uniform and that the ambient noise level is low enough to be ignored.

In the real world, the decay rate in a particular band of frequencies may not be uniform and it may be very difficult to measure accurately over a total 60 dB range. Most acousticians are satisfied to measure the first 30 dB decay after a test signal is turned off and to use the slope of this portion of the curve to define the average decay rate and thus the reverberation time. In the example just given, estimates must be made over a useful range of only 20 dB or so. However, the height of the chart paper corresponds to a total range of 30 dB and this makes calculation of reverberation time quite simple. At 125 Hz a sloping line drawn across the full width of the chart paper is equivalent to a 30 dB decay in 0.27 seconds. Reverberation time (60 dB decay) must therefore be twice this value, or 0.54 seconds. Similarly, the same room has a reverberation time of only 0.4 seconds in the 4 kHz band.

In his original work in architectural acoustics, Sabine assumed the idealized exponential growth and decay of sound we showed in Figure 5-6. However, his equation based on this model was found to be inaccurate in rooms having substantial absorption. In other words, the Sabine equation works well in live rooms, but not in moderately dead ones. In the 1920’s and 1930’s, a great deal of work was done in an effort to arrive at a model that would more accurately describe the growth and decay of sound in all types of rooms. On the basis of the material presented thus far, let us see if we can construct such a model.

We start by accepting the notion of a uniform diffuse steady state sound field. Even though the sound field in a real room may fluctuate, and although it may not be exactly the same at every point in the room, some sort of overall intensity average seems to be a reasonable simplifying assumption.

If we can average out variations in the sound field throughout the room, perhaps we can also find an average distance that sound can travel before striking one of the boundary surfaces. This notion of an average distance between bounces is more accurately known as the *mean free path* (MFP) and is a common statistical notion in other branches of physics. For typical rooms, the MFP turns out to be equal to 4V/S, where *V* is the enclosed volume and *S* is the area of all the boundary surfaces.

Since sound waves will have bounced around all parts of the room striking all of the boundary surfaces in almost all possible angles before being completely absorbed, it seems reasonable that there should be some sort of average absorption coefficient $\bar{a}$ which would describe the total boundary surface area. We will use the simple arithmetic averaging technique to calculate this coefficient.

At this point we have postulated a highly simplified acoustical model which assumes that, on the average, the steady state sound intensity in an actual room can be represented by a single number. We also have assumed that, on the average, sound waves in this room travel a distance equivalent to MFP between bounces. Finally, we have assumed that, on the average, each time sound encounters a boundary surface it impinges upon a material having a random incidence absorption coefficient denoted

$S = 126m^2$
$V = 90m^3$
$4V/S = 3m$
MEAN FREE TIME = .008 sec

FLOOR = $30m^2 \times .30 = 9m^2$
CEILING = $30m^2 \times .33 = 10m^2$
WALLS = $66m^2 \times .09 = 5.9m^2$
$126m^2$ $24.9m^2$

$$\bar{\alpha} = \frac{24.9}{126} = 0.2$$

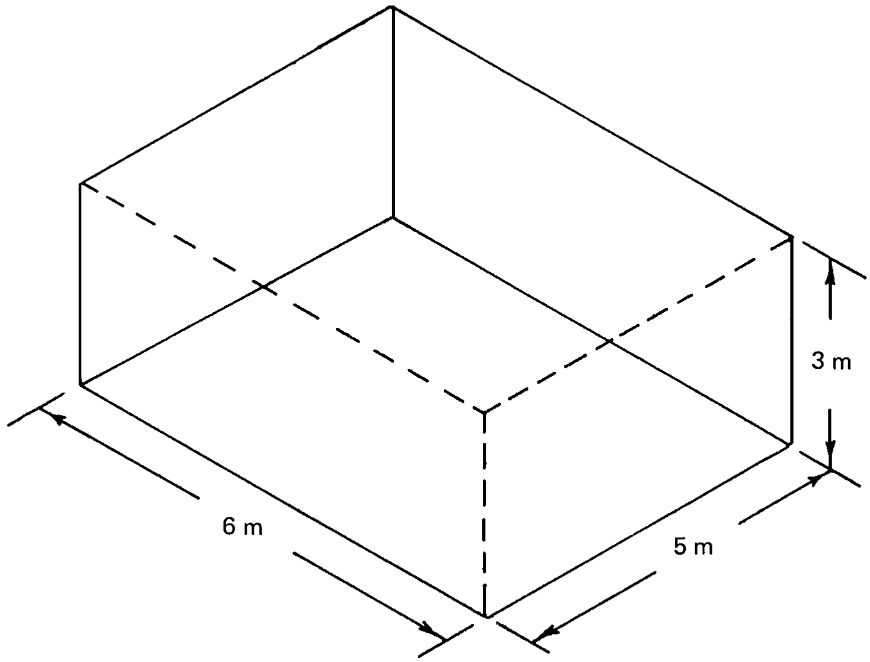


Figure 5-8. Calculating reverberation time

REVERBERATION TIME EQUATIONS: T = 60 dB DECAY TIME IN SECONDS		
EQUATION:	ENGLISH UNITS: S = SURFACE AREA IN FT^2 V = VOLUME IN FT^3	SI UNITS: S = SURFACE AREA IN m^2 V = VOLUME IN m^3
SABINE – GIVES BEST CORRESPONDENCE WITH PUBLISHED ABSORPTION COEFFICIENTS WHERE $\bar{\alpha}$ IS LESS THAN 0.2	$T = \frac{.049V}{S\bar{\alpha}}$	$T = \frac{.16V}{S\bar{\alpha}}$
EYRING – PREFERRED FORMULA FOR WELL-BEHAVED ROOMS HAVING $\bar{\alpha}$ GREATER THAN 0.2 OR SO	$T = \frac{.049V}{-S \ln(1-\bar{\alpha})}$	$T = \frac{.16V}{-S \ln(1-\bar{\alpha})}$
FITZROY-(SABIN) – FOR RECTANGULAR ROOMS IN WHICH ABSORPTION IS NOT WELL DISTRIBUTED.	$T = \frac{.049V}{S^2}\left(\frac{X^2}{X\alpha_x} + \frac{Y^2}{Y\alpha_y} + \frac{Z^2}{Z\alpha_z}\right)$	$T = \frac{.16V}{S^2}\left(\frac{X^2}{X\alpha_x} + \frac{Y^2}{Y\alpha_y} + \frac{Z^2}{Z\alpha_z}\right)$
αx, αy, AND αz ARE AVERAGE ABSORPTION COEFFICIENTS OF OPPOSING PAIRS OF SURFACES WITH TOTAL AREAS x, y, AND z.		

Figure 5-9. Reverberation time equations

by a single number, ā. Only one step remains to complete our model. Since sound travels at a known rate of speed, the mean free path is equivalent to a certain *mean free time* between bounces.

Now imagine what must happen if we apply our model to the situation that exists in a room immediately after a uniformly emitting sound source has been turned off. The sound waves continue to travel for a distance equal to the mean free path. At this point they encounter a boundary surface having an absorption coefficient of ā and a certain percentage of the energy is lost. The remaining energy is reflected back into the room and again travels a distance equal to the mean free path before encountering another boundary with absorption coefficient ā. Each time sound is bounced off a new surface, its energy is decreased by a proportion determined by the average absorption coefficient ā.

If we know the proportion of energy lost with each bounce and the length of time between bounces, we can calculate the average rate of decay and the reverberation time for a particular room.

Example: Consider a room 5m x 6m x 3m, as diagrammed in Figure 5-8. Let us calculate the decay rate and reverberation time for the octave band centered at 1 kHz.

The volume of the room is 90 cubic meters, and its total surface area is 126 square meters; therefore, the MFP works out to be about 3 meters.

The next step is to list individually the areas and absorption coefficient of the various materials used on room surfaces.

The total surface area is 126 square meters; the total absorption (Sā) adds up to 24.9 absorption units. Therefore, the average absorption coefficient (ā) is 24.9 divided by 126, or .2.

If each reflection results in a decrease in energy of 0.2, the reflected wave must have an equivalent energy of 0.8. A ratio of 0.8 to 1 is equivalent to a loss of 0.97 decibel per reflection. For simplicity, let us call it 1 dB per reflection.

Since the MFP is 2.9 meters, the mean free time must be about 0.008 seconds (2.9/334 = 0.008).

We now know that the rate of decay is equivalent to 1 dB per 0.008 seconds. The time for sound to decay 60 dB must, therefore, be:

$$60 \times 0.008 = 0.48 \text{ seconds.}$$

The Eyring equation in its standard form is shown in Figure 5-9. If this equation is used to calculate the reverberation of our hypothetical room, the answer comes out 0.482 seconds. If the Sabine formula is used to calculate the reverberation time of this room, it provides an answer of 0.535 seconds or a discrepancy of a little more than 10%.

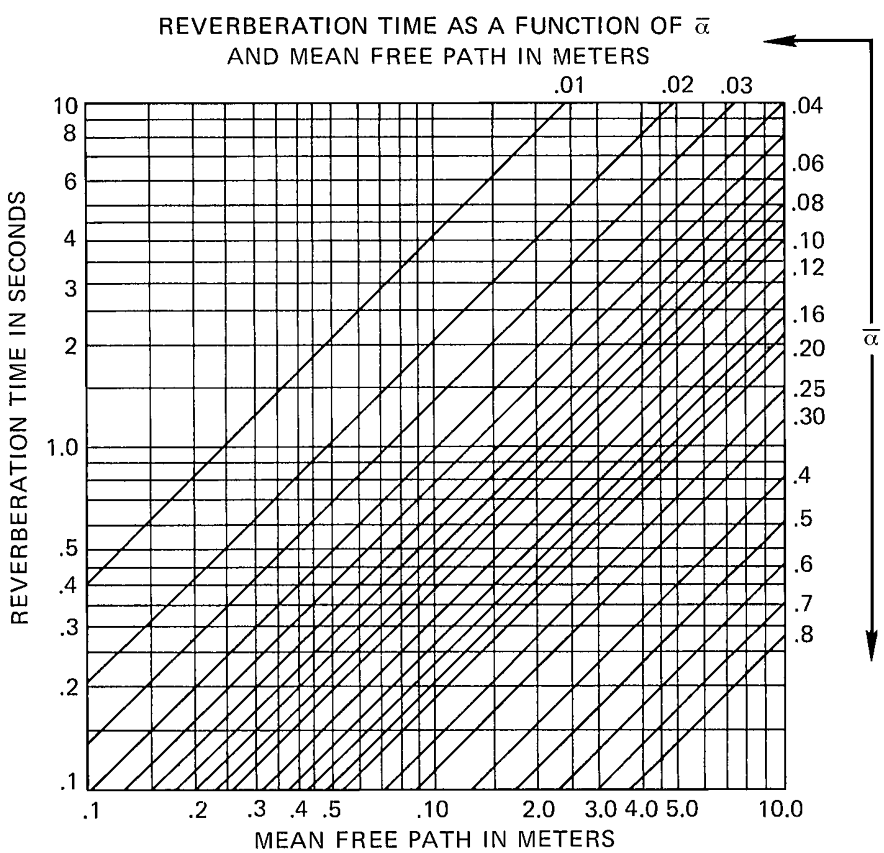


Figure 5-10. Reverberation time chart, SI units

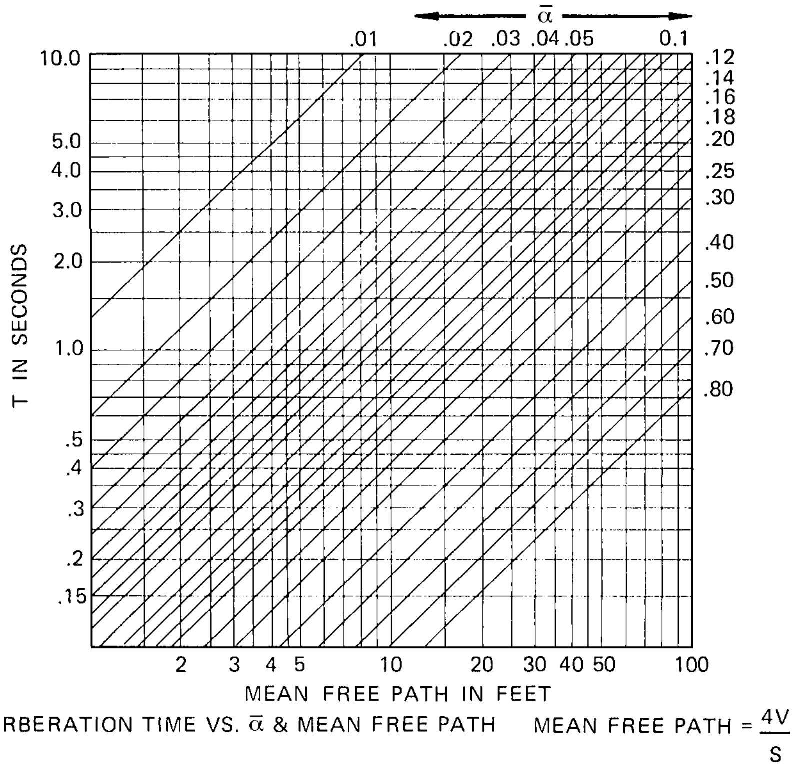


Figure 5-1 1. Reverberation time chart, English units

DESCRIPTION	125	1 kHz	4 kHz
BRICK WALL (18" THICK, UNPAINTED)	.02	.04	.07
BRICK WALL (18" THICK, PAINTED)	.01	.02	.02
INTERIOR PLASTER ON METAL LATH	.02	.06	.03
POURED CONCRETE	.01	.02	.03
PINE FLOORING	.09	.08	.10
CARPETING WITH PAD	.10	.30	.70
DRAPES (COTTON, 2X FULLNESS)	.07	.80	.50
DRAPES (VELOUR, 2X FULLNESS)	.15	.75	.65
ACOUSTIC TILE (5/8", #1 MOUNT)	.15	.70	.65
ACOUSTIC TILE (5/8", #2 MOUNT)	.25	.70	.65
ACOUSTIC TILE (5/8", #7 MOUNT)	.50	.75	.65
TECTUM PANELS (1", #2 MOUNT)	.08	.55	.65
TECTUM PANELS (1", #7 MOUNT)	.35	.35	.65
PLYWOOD PANELING (1/8", 2" AIR SPACE)	.30	.10	.07
PLYWOOD CYLINDERS (2 LAYERS 1/8")	.35	.20	.18
PERFORATED TRANSITE (W/PAD, #7 MOUNT)	.90	.95	.45
OCCUPIED AUDIENCE AREA	.50	.95	.85
UPHOLSTERED THEATRE SEATS ON HARD FLOOR	.45	.90	.70

#1 MOUNT: CEMENTED DIRECTLY TO PLASTER OR CONCRETE.
#2 MOUNT: FASTENED TO NOM. 1" THICK FURRING STRIPS.
#7 MOUNT: SUSPENDED CEILING WITH 16" AIR SPACE ABOVE.

Figure 5-12. Approximate absorption coefficients of common material (averaged and rounded-off from published data)

Rather than go through the calculations, it is much faster to use a simple chart. Charts calculated from the Eyring formula are given in Figures 5-10 and 5-11. Using the chart as a reference and again checking our hypothetical example, we find that a room having a mean free path just a little less than 3 meters and an average absorption coefficient of .2 must have a reverberation time of just a little less than .5 seconds.

Since reverberation time is directly proportional to the mean free path, it is desirable to calculate the latter as accurately as possible. However, this is not the only area of uncertainty in these equations. There is argument among acousticians as to whether published absorption coefficients, such as those of Figure 5-12, really correspond to the random incidence absorption implicit in the Eyring equation. There also is argument over the method used to find the "average" absorption coefficient for a room. In our example, we performed a simple arithmetic calculation to find the average absorption coefficient. It has been pointed out that this is an unwarranted simplification — that the actual state of affairs requires neither an arithmetic average nor a geometric mean, but some relation considerably more complicated than either.

Another source of uncertainty lies in determining the absorption coefficients of materials in situations other than those used to establish the rating. We know, for example, that the total absorption of a single large patch of material is less than if the same amount of material is spread over a number of separated, smaller patches. At higher frequencies, air absorption reduces reverberation time. Figure 5-13 can be used to estimate such deviations above 2 kHz.

A final source of uncertainty is inherent in the statistical nature of the model itself. We know from experience that reverberation time in a large concert hall may be different in the seating area than if measured out near the center of the enclosed space.

With all of these uncertainties, it is a wonder that the standard equations work as well as they do. The confidence limit of the statistical model is probably of the order of 10% in terms of time or decay rate, or ±1 dB in terms of sound pressure level. Therefore, carrying out calculations to 3 or 4 decimal places, or to fractions of decibels, is not only unnecessary but mathematically irrelevant.

Reverberation is only one of the characteristics that help our ears identify the "acoustical signature" of an enclosed space. Some acousticians separate acoustical qualities into three categories: the direct sound, early reflections, and the late-arriving reverberant field.

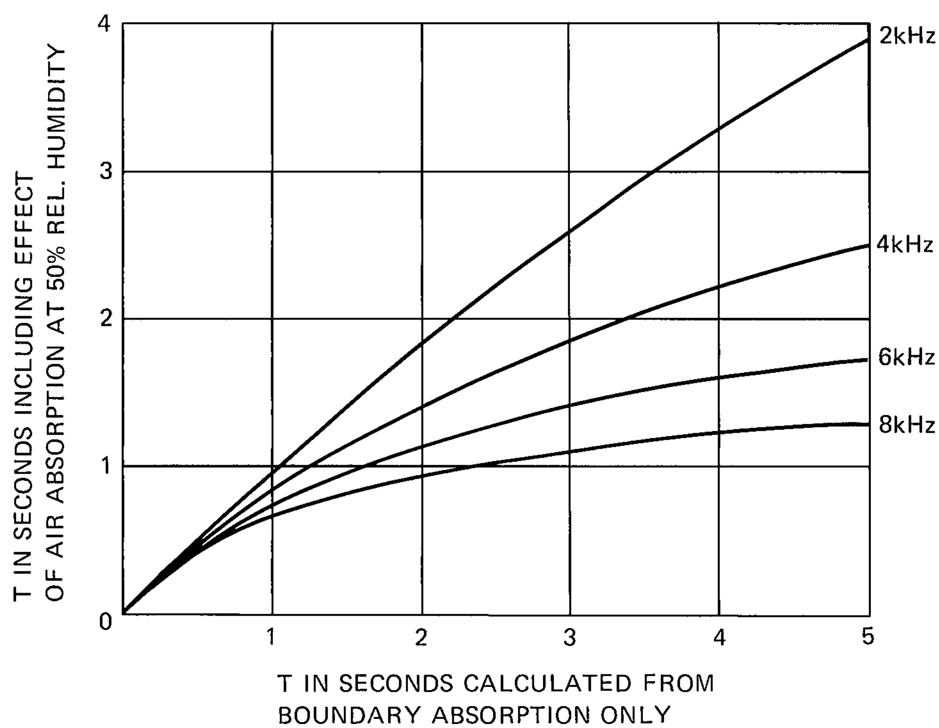


Figure 5-13. Effect of air absorption on calculated reverberation time

Another identifiable characteristic, particularly of small rooms, is the presence of identifiable resonance frequencies. Although this factor is ignored in our statistical model, a room is actually a complicated resonant system very much like a musical instrument. As mentioned previously, if individual resonances are clustered close together in frequency the ear tends to average out peaks and dips, and the statistical model seems valid. At lower frequencies, where resonances may be separated by more than a critical bandwidth, the ear identifies a particular timbral characteristic of that room at a specific listening location.

Since the direct sound field is independent of the room, we might say that the “three R’s” of room acoustics are *reverberation*, *room resonances* and early *reflections*.

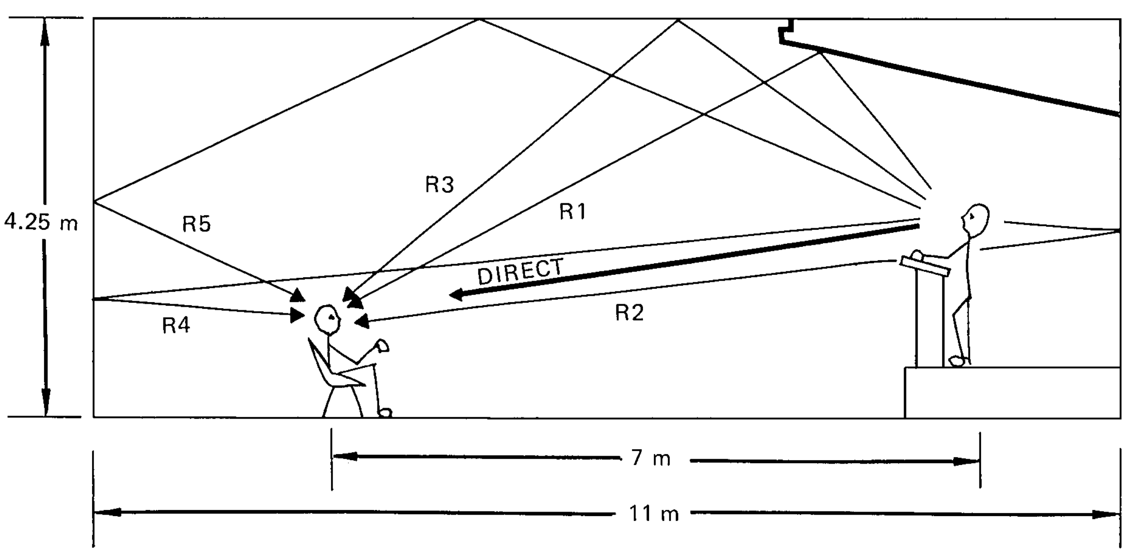
The distinction between early reflections and the later reverberation is usually made at some point between 20 and 30 milliseconds after the arrival of the direct sound. Most people with normal hearing find that early reflections are combined with the direct sound by the hearing mechanism, whereas later reflections become identified as a property of the enclosed space. See Figure 5-14. The early reflections, therefore, can be used by the brain as part of the decoding process. Late reverberation, while providing an agreeable aesthetic component for many kinds of music, tends to mask the early sound and interferes with speech intelligibility.

One final characteristic of sound is ignored in all standard equations. Localization of a sound source affects our subjective assessment of the sound field. In the design of sound reinforcement systems, localization is largely disregarded except for a few general rules. It achieves critical importance, however, in the design of multi-channel monitoring and mixdown rooms for recording studios.

Direct and Reverberant Sound Fields

What happens to the inverse square law in a room? As far as the direct sound is concerned (that which reaches a listener directly from the source without any reflections) the inverse square relationship remains unchanged. But in an enclosed space we now have a second component of the total sound field. In our statistical model we assumed that at some distance sufficiently far from the source, the direct sound would be buried in a “soup” of random reflections from all directions. This reverberant sound field was assumed to be uniform throughout the enclosed space.

Figure 5-15 illustrates how these two components of the total sound field are related in a typical situation. We have a sound source radiating uniformly through a hemispherical solid angle. The direct energy radiated by the source is represented by the black dots. Relative energy density is



SOUND RAY	DELAY IN MSEC.
DIRECT	0
R1	4.5
R2	9.0
R3	13.0
R4	14.0
R5	18.0

ALL REFLECTIONS SHOWN ARRIVE AT LISTENER’S EAR WITHIN 30 MSEC. AFTER DIRECT SOUND AND, THEREFORE, ARE “EARLY” REFLECTIONS.

Figure 5-14. Early reflections in relation to direct sound

indicated by the density of the dots on the page; near the source they are very close together and become more and more spread out at greater distances from the source.

The reverberant field is indicated by the circle dots. Their spacing is uniform throughout the enclosed space to represent the uniform energy density of the reverberant field.

Near the source the direct field predominates. As one moves farther away, however, the ratio of black dots to circle dots changes until the black dots are so few and far between that their presence can be ignored. In this area one is well into the reverberant field of the room. At some particular distance from the source a zone exists where the densities of the circle and black dots are equal. In the illustration, this zone takes the form of a semicircle; in three-dimensional space, it would take the form of a hemisphere.

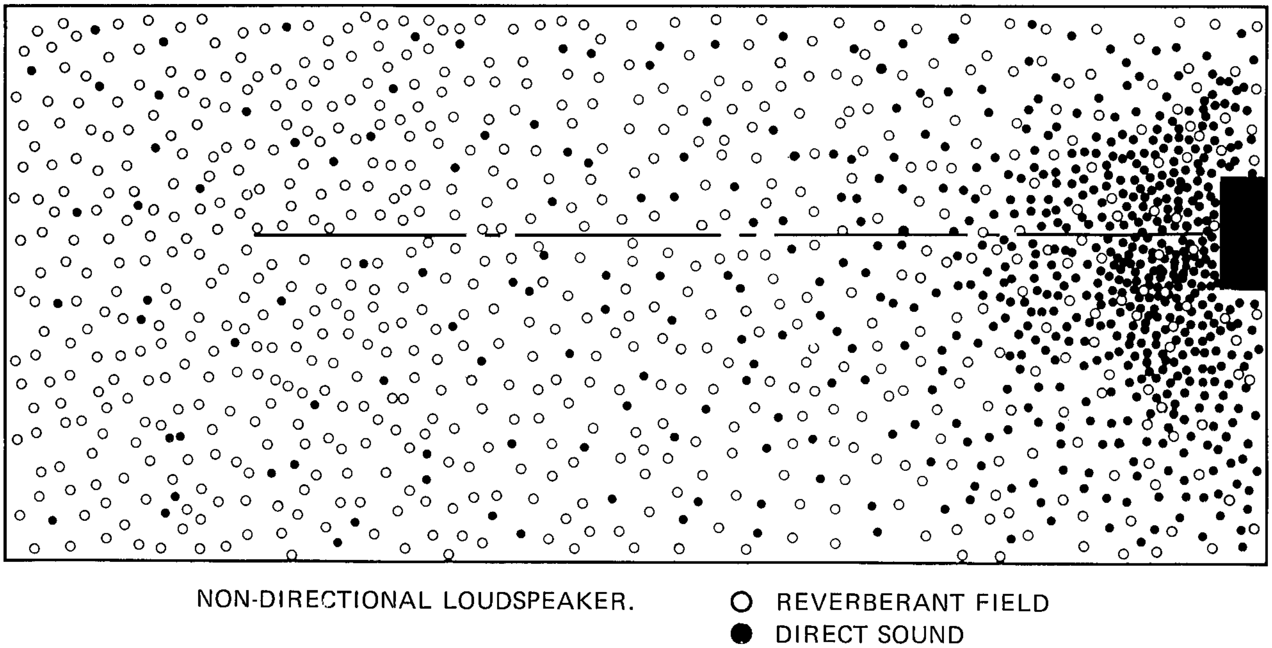


Figure 5-15. Direct and reverberant fields, non-directional loudspeaker

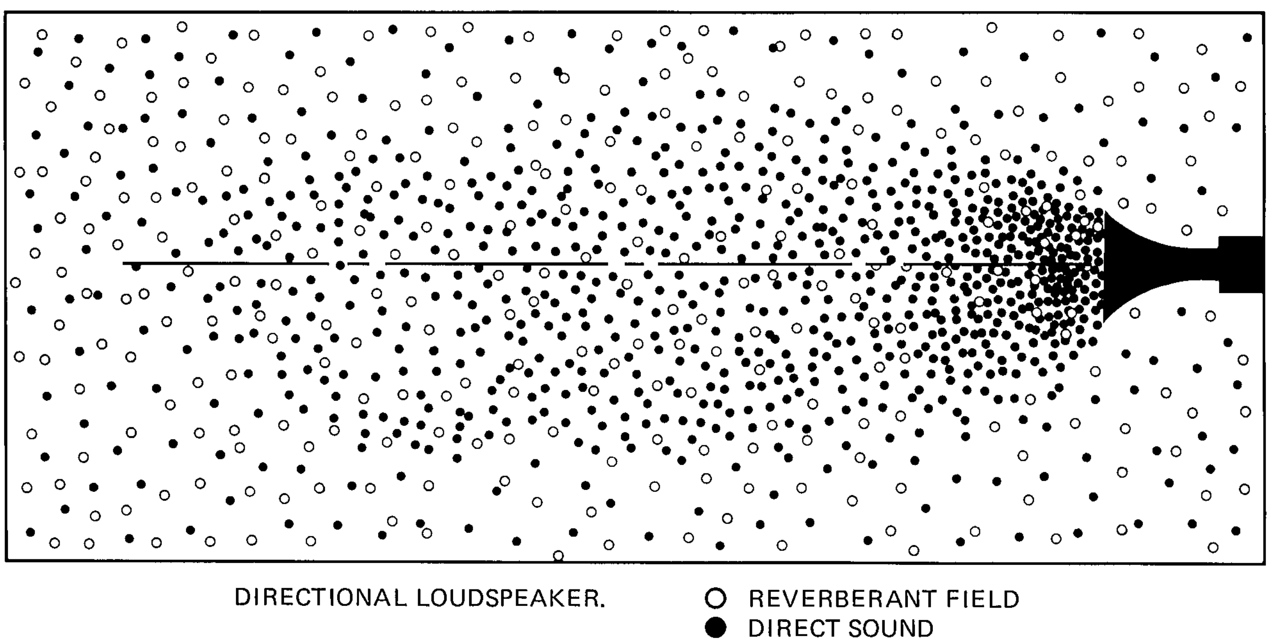


Figure 5-16. Direct and reverberant fields, directional loudspeaker

Critical Distance (D_c)

The distance from the acoustic center to the circle-black boundary is called the *critical distance*. Critical distance is the distance from the acoustic center of a sound source, along a specified axis, to a point at which the densities of direct and reverberant sound fields are equal.

Critical distance is affected by the directional characteristics of the sound source. Figure 5-16 illustrates the same room as in Figure 5-15, but with a more directional loudspeaker. In the instance the circle-black boundary no longer describes a semicircle. The black dots are concentrated along the major axis of the loudspeaker and maintain their dominance over the circle dots for a substantially greater distance than in the preceding example. However, at 45° or greater off the major axis, the black dots die out more rapidly and the circle-black boundary is much closer to the source.

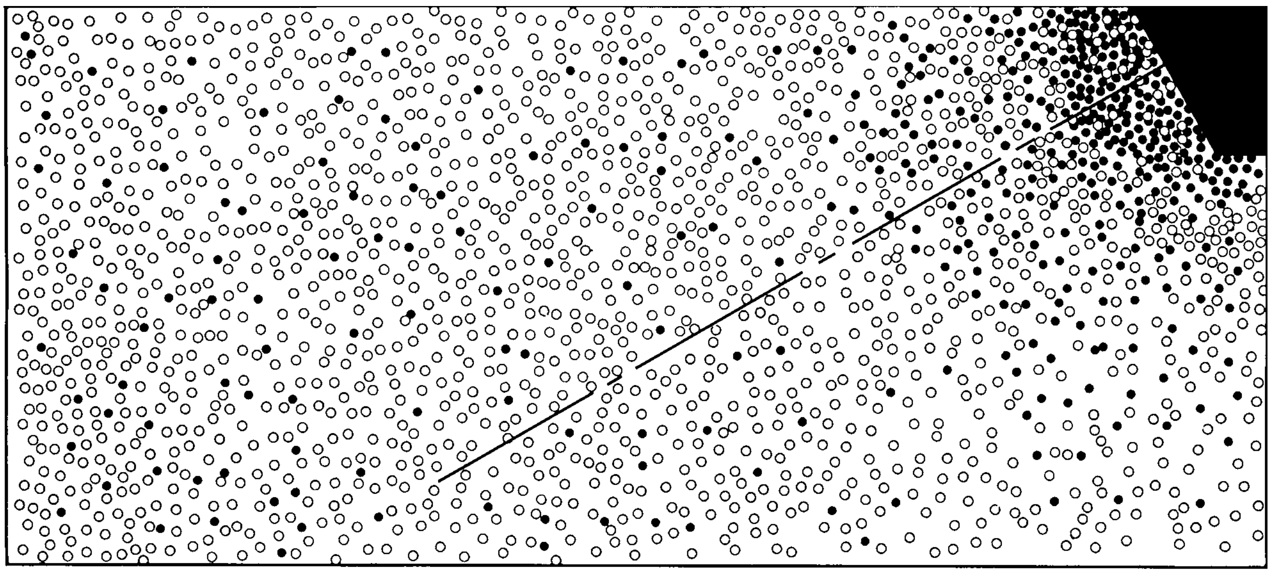


Figure 5-17. Direct and reverberant fields, live room

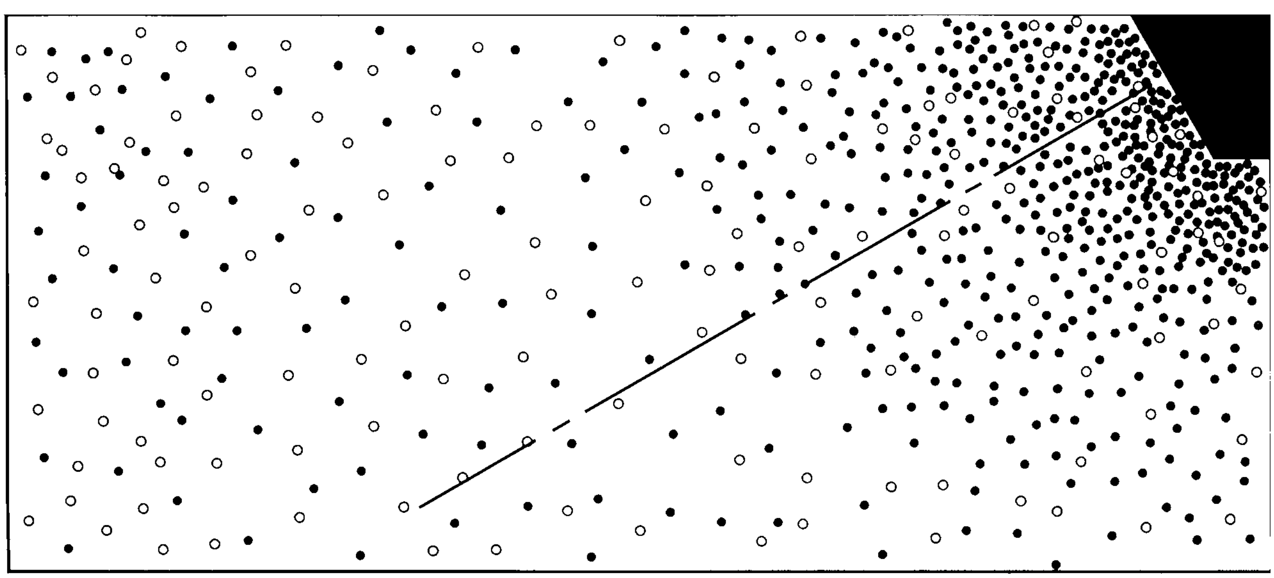


Figure 5-18. Direct and reverberant fields, dead room

Critical distance also is affected by the absorption coefficients of room boundary surfaces. Figures 5-17 and 5-18 illustrate the same sound source in the same size room. The difference is that in the first illustration the room surfaces are assumed to be highly reflective, while in the second they are more absorptive. The density of the black dots representing the direct field is the same in both illustrations. In the live room, because energy dissipates quite slowly, the reverberant field is relatively strong. As a result, the circle-black boundary is pushed in close to the sound source. In the second example sound energy is absorbed more rapidly, and the reverberant field is not so strong. Therefore, the circle-black boundary is farther from the source.

Even though the direct field and the reverberant field are produced by the same sound source, the sound is so well scrambled by multiple reflections that the two components are non-coherent. This being so, total rms sound pressure measured at the critical distance should be 3 dB greater than that produced either by the direct field or reverberant field alone.

Within the normal variations of statistical averaging, such is the case in actual rooms. The behavior of loudspeakers in rooms was described in great detail in 1948 by Hopkins and Stryker (6). Their calculations of average sound pressure level versus distance are illustrated in Figure 5-19. A great deal of useful information has been condensed into this single chart. Sound pressure is given in terms of the level produced by a point source radiating one acoustic watt. The straight diagonal line shows the decrease in sound pressure with distance that would be measured in open air.

The Room Constant (R)

The various shelving curves are labeled with numbers indicating a new quantity, the *room constant*. This will be defined in subsequent paragraphs. Essentially, R is a modified value of the total absorption in the room [$R = S\bar{a}/(1 - \bar{a})$]. A small room constant indicates a very live room, and a large room constant describes a room having a great deal of absorption.

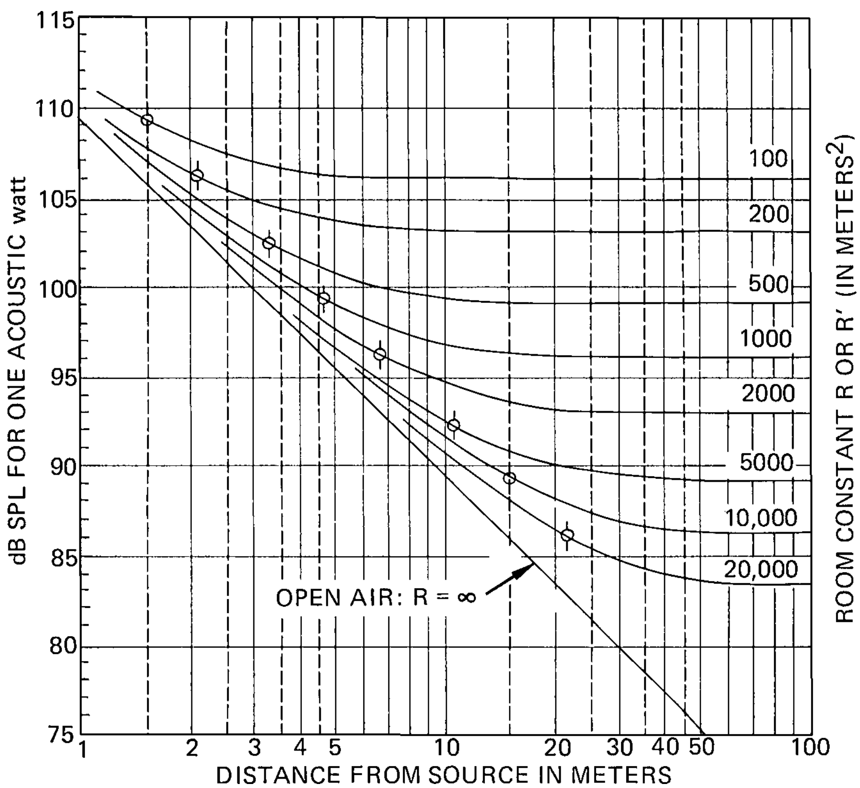


Figure 5-19. SPL (point source radiating one acoustic watt) vs. R and distance from source

Suppose we place a small non-directional sound source in a room having R = 200 m^2. If we measure the sound level at a distance 0.25 meter from the acoustic center and then proceed to walk in a straight line away from the source, the level will at first decrease as the square of the distance. However, about 1 meter from the source, the inverse square relationship no longer applies. At distances of 6 meters or more from the source, there is no substantial change in sound pressure at all because we are well into the reverberant field and the direct sound no longer has a perceptible effect upon our reading.

If we reverse our path and walk back toward the source from a distance of 12 or 15 meters, sound pressure at first remains unchanged and then gradually begins to climb until, at a distance about 2 meters from the source, it has increased 3 dB above the reverberant field reading. This position, indicated by the mark on the curve, is the critical distance.

The graph of Figure 5-20 is a universal relationship in which critical distance is used as the measuring stick. It can be seen that the effective transition zone from the reverberant field to the direct field exists over a range from about one-half the critical distance to about twice the critical distance. At one-half the critical distance, the total sound field is 1 dB greater than the direct field alone; at twice the critical distance, the total sound field is 1 dB greater than the reverberant field alone.

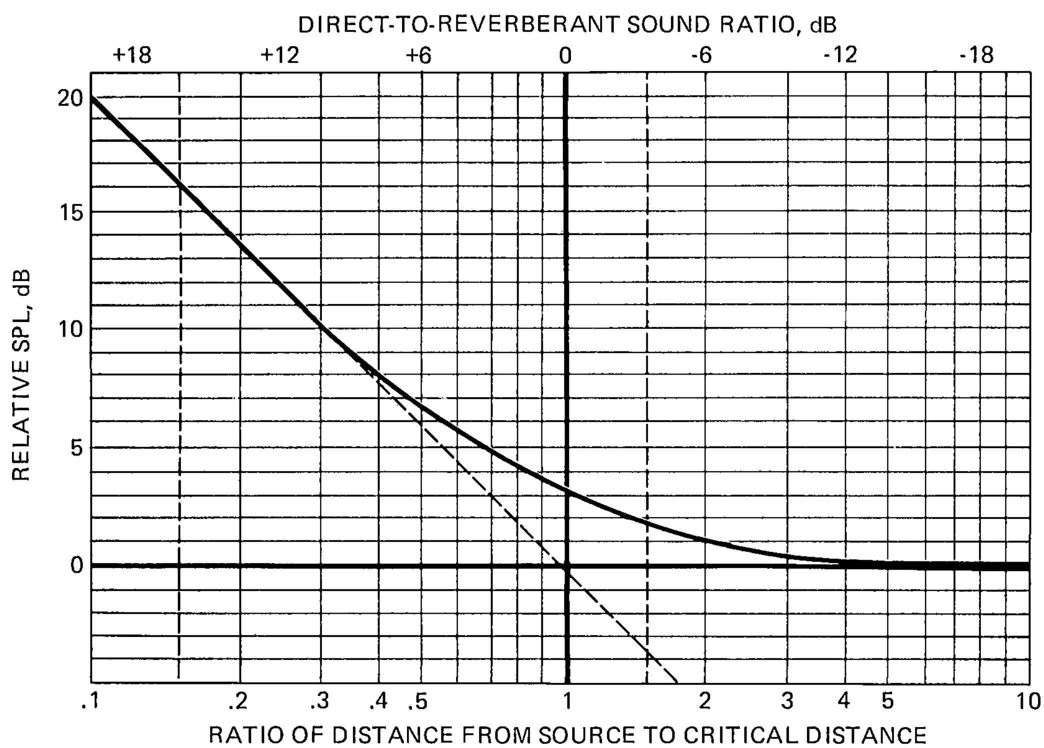
The ratio of direct to reverberant sound can be calculated from the simple equation shown below the chart, or estimated directly from the chart itself. For example, at four times D_C the direct sound field is 12 dB less than the reverberant sound field. At one-half D_C, the direct sound field is 6 dB greater than the reverberant sound field.

Remember that, although critical distance depends on the directivity of the source and the absorption characteristics of the room, the relationships expressed in Figure 5-19 remain unchanged. Once D_C is known, all other factors can be calculated without regard to room characteristics. With a directional sound source, however, a given set of calculations can be used only along a specified axis. On any other axis the critical distance will change and must be recalculated.

Let us investigate these two factors in some detail: first the room constant R, and then the directivity factor Q.

We have already mentioned that the room constant is related to the total absorption of an enclosed space, but that it is different from total absorption represented by $S\bar{a}$.

One way to understand the room constant is first to consider that the total average energy density in a room is directly proportional to the power of the sound source and inversely proportional to the total absorption of the boundary surfaces. This



CALCULATED FROM: $10\log(1+1/x^2)$ WHERE x IS RATIO OF DISTANCE FROM SOURCE TO CRITICAL DISTANCE

Figure 5-20. Relative SPL vs. distance from source in relation to critical distance

relationship is often indicated by the simple expression: $4W/cS\bar{a}$. *W* represents the output of the sound source, and the familiar expression *S*$\bar{a}$ indicates the total absorption of the boundary surfaces.

Remembering our statistical room model, we know that sound travels outward from a point source, following the inverse square law for a distance equal to the mean free path, whereupon it encounters a boundary surface having an absorption coefficient $\bar{a}$. This direct sound has no part in establishing the reverberant sound field. The reverberant field proceeds to build up only after the first reflection.

But the first reflection absorbs part of the total energy. For example, if $\bar{a}$ is 0.2, only 80% of the original energy is available to establish the reverberant field. In other words, to separate out the direct sound energy and perform calculations having to do with the reverberant field alone, we must multiply *W* by the factor $(1 - \bar{a})$.

This results in the equation:

$$E_{rev} = \frac{4W}{cR}*$$

This gives the average energy density of the reverberant field alone. If we let $R = S\bar{a}/(1 - \bar{a})$, the equation becomes:

$$E_{rev} = \frac{4W(1-\bar{\alpha})}{cS\bar{\alpha}}$$

Note that the equation has nothing to do with the directivity of the sound source. From previous examples, we know that the directivity of the source affects critical distance and the contour of the boundary zone between direct and reverberant fields. But power is power, and it would seem to make no difference whether one acoustic watt is radiated in all directions from a point source or concentrated by a highly directional horn.

Is this really true? The equation assumes that the porportion of energy left after the first reflection is equivalent to $W(1 - \bar{a})$. Suppose we have a room in which part of the absorption is supplied by an open window. Our sound source is a highly directional horn located near the window. According to the equation the energy density of the reverberant field will be exactly the same whether the horn is pointed into the room or out of the window! This obviously is fallacious, and is a good example of the importance of understanding the basis for acoustical equations instead of merely plugging in numbers.

* With room dimensions in meters and acoustic power in watts, the reverberant field level in dB is: $L_{rev} = 10 \log W/R + 126$ dB. See Figure 5-21.

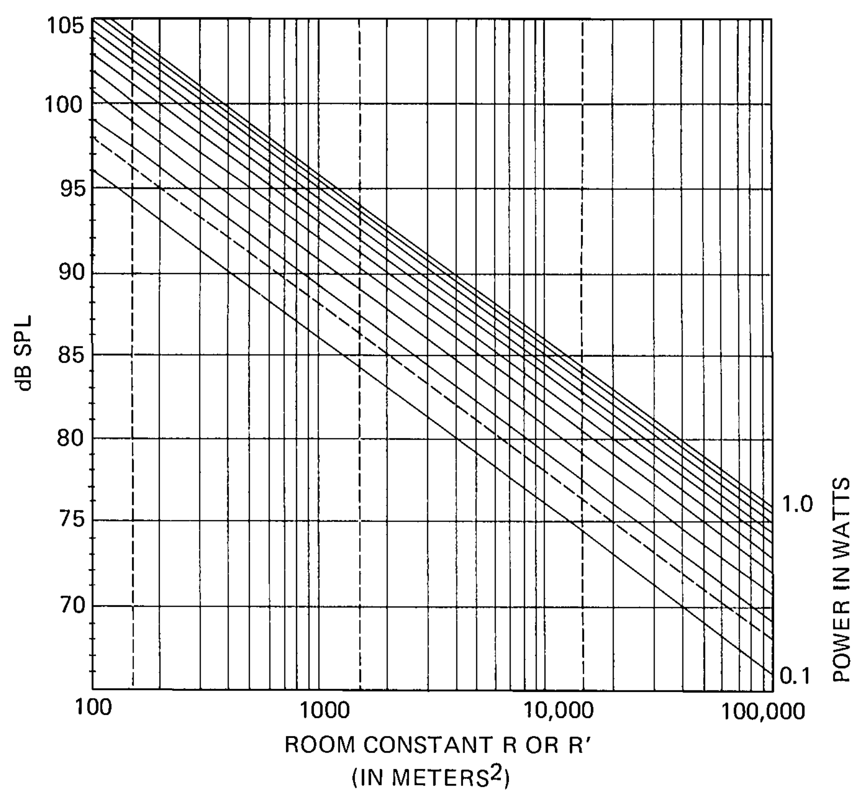


Figure 5-21. Steady-state reverberant field SPL vs. acoustic power and room constant

We can agree that if the source of sound in a given room is non-directional, the equation for *R* is probably accurate for all practical purposes. It would also seem that the equation could be used for a room in which absorption was uniformly distributed on all boundary surfaces, regardless of the directivity of the source. Where we run into trouble is the situation of a directional source and absorption concentrated in restricted areas. The description is exactly that of a classical concert hall in which almost all absorption is provided in the audience area and in which the sound system designer has endeavored to concentrate the power from the loudspeakers directly into the audience.

One could go through laborious calculations to arrive at the intensity of the reverberant field by taking reflections one by one. In practice, however, it is usually sufficient to make an educated guess as to the amount of energy absorbed in the first reflection. We can denote the absorption coefficient of this first reflection as *a'*. The energy remaining after the first reflection must then be proportional to (1 - a'). This allows us to write an expression for the effective room constant designated by the symbol *R'*:

$$R' = S\bar{a}/(1 - a').$$

The importance of determining the room constant as accurately as possible lies in the fact that it not only allows us to calculate the maximum level of a given sound system in a given room, but also enters into our calculations of critical distance and direct-to-reverberant sound ratio.

Although not explicitly stated, *R'* can be used in any of the equations and charts in which the room constant appears, Figures 5-19, 21, and 22, for example. In most situations, the standard equation for *R* will seem to be a reasonable approximation of the condition that exists. In each case, however, an examination of the room geometry and source directivity should be made, and the designer should try to estimate what will really happen to the sound energy after the first reflection.

Figures 5-21 and 5-22 present some reverberant field relationships in graphical form. For example, if we know the efficiency of a sound source, and hence its acoustical power output in watts, we can measure the sound pressure level in the reverberant field and determine the room constant directly. Or, if the room is not accessible to us, and a description of the room enables us to estimate the

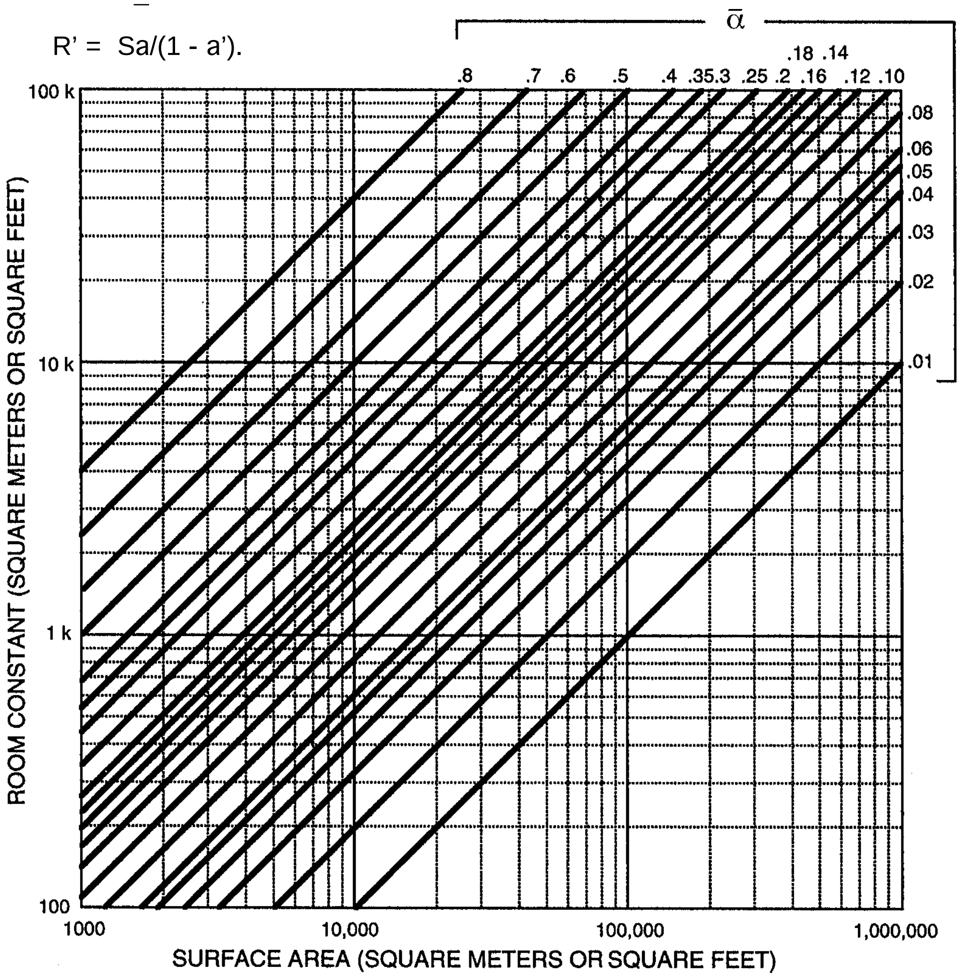


Figure 5-22. Room constant vs. surface area and ā

room constant with some confidence, then we can estimate the sound pressure level that will be produced in the reverberant field of the room for a given acoustical power output.

Figure 5-22 enables us to determine by inspection the room constant if we know both $\bar{a}$ and the total surface area. This chart can be used with either SI or English units.

If both room constant and directivity factor of a radiator are known, the critical distance can be solved directly from the following equation:

$$D_C = .14\sqrt{QR}$$

This equation may be used with either SI or English units, and a graphical solution for it is shown in Figure 5-23. It is helpful to remember that the relationship between directivity index and critical distance is in a way very similar to the inverse square law: an increase of 6 dB in directivity (or a "times-four" increase in *Q*) corresponds to a doubling of the critical distance. One might think of this as the "direct square law".

A second useful factor to keep in mind is that the directivity index of a person talking, taken in the 1 kHz range along the major axis, is about 3 dB. For convenience in sound system calculations, we normally assume the Q of the talker to be 2.

These two facts can be used to make reasonably accurate acoustical surveys of existing rooms without equipment. All that is needed is the cooperation of a second person — and a little experience. Have your assistant repeat a word or count slowly in as even a level as possible. While he is doing this, walk directly away from him while carefully listening to the intensity and quality of his voice. With a little practice, it is easy to detect the zone in which the transition is made from the direct field to the reverberant field. Repeat the experiment by starting at a considerable distance away from the talker, well into the reverberant field, and walking toward him. Again, try to zero in on the transition zone.

After two or three such tries you may decide, for example, that the critical distance from the talker in that particular room is about 4 meters. You know that a loudspeaker having a directivity index of 3 dB will also exhibit a critical distance of 4 meters along its major axis in that room. To extend the critical distance to 8 meters, the loudspeaker must have a directivity index of 9 dB.

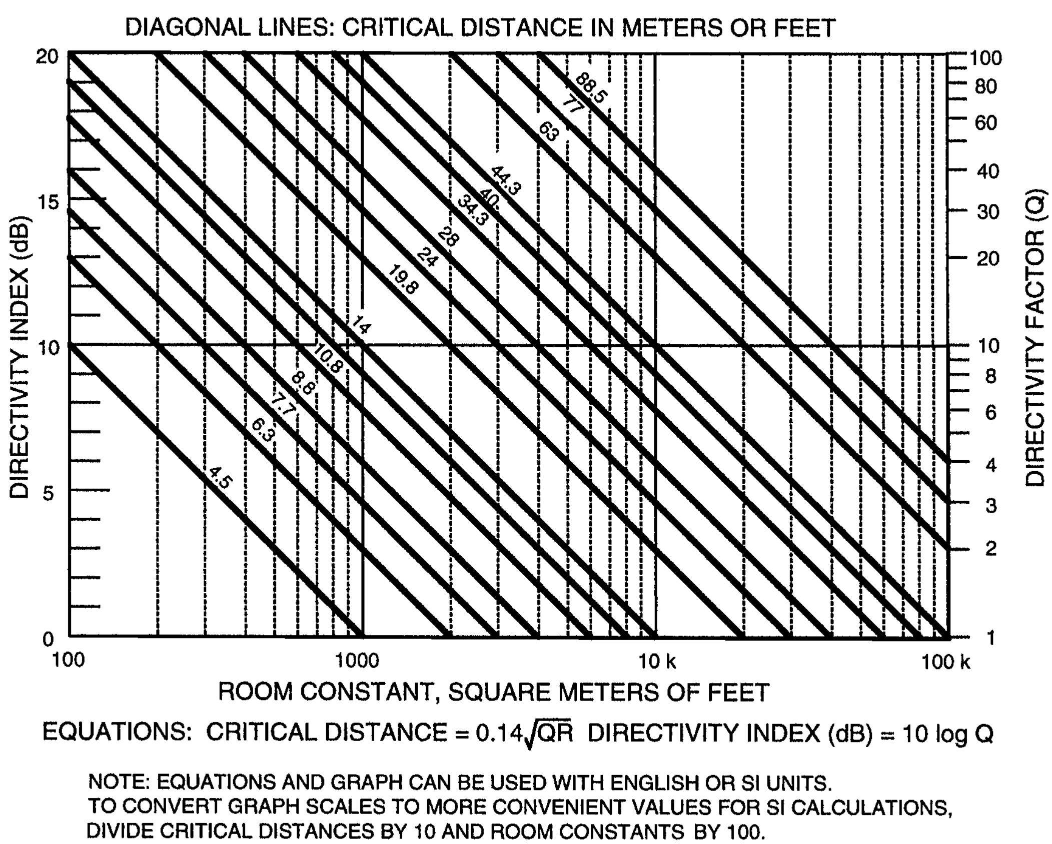


Figure 5-23. Critical distance as a function of room constant and directivity index or directivity factor

Once the critical distance is known, the ratio of direct to reverberant sound at any distance along that axis can be calculated. For example, if the critical distance for a talker is 4 meters, the ratio of direct to reverberant sound at that distance is unity. At a distance of 8 meters from the talker, the direct sound field will decrease by 6 dB by virtue of inverse square law, whereas the reverberant field will be unchanged. At twice critical distance, therefore, we know that the ratio of direct to reverberant sound must be -6 dB. At four times D_C, the direct-to-reverberant ratio will obviously be -12 dB.

Statistical Models vs. the Real World

We stated earlier that a confidence level of about 10% allowed us to simplify our room calculations significantly. For the most part, this is true; however, there are certain environments in which errors may be quite large if the statistical model is used. These are typically rooms which are acoustically dead and have low ceilings in relation to their length and width. Hotel ballrooms and large meeting rooms are examples of this. Even a large pop recording studio of more regular dimensions may be dead enough so that the ensemble of reflections needed to establish a diffuse reverberant field simply cannot exist. In general, if the average absorption coefficient in a room is more than about 0.2, then a diffuse reverberant field will not exist. What is usually observed in such rooms is data like that shown in Figure 5-24.

Peutz (9) has developed an empirical equation which will enable a designer to estimate the approximate slope of the attenuation curve beyond D_C in rooms with relatively low ceilings and low reverberation times:

$$\Delta \approx \frac{0.4\sqrt{V}}{h\,T_{60}}\ \text{dB}$$

In this equation, *D* represents the additional fall-off in level in dB per doubling of distance beyond D_C. *V* is the volume in meters^3, *h* is the ceiling height in meters, and T_{60} is the reverberation time in seconds. In English units (*V* in ft^3 and *h* in feet), the equation is:

$$\Delta \approx \frac{0.22\sqrt{V}}{h\,T_{60}}\ \text{dB}$$

As an example, assume we have a room whose height is 3 meters and whose length and width are 15 and 10 meters. Let us assume that the reverberation time is one second. Then:

$$\Delta \approx \frac{0.4\sqrt{450}}{3\,(1)} = 2.8\ \text{dB}$$

Thus, beyond D_C we would observe an additional fall-off of level of about 3 dB per doubling of distance.

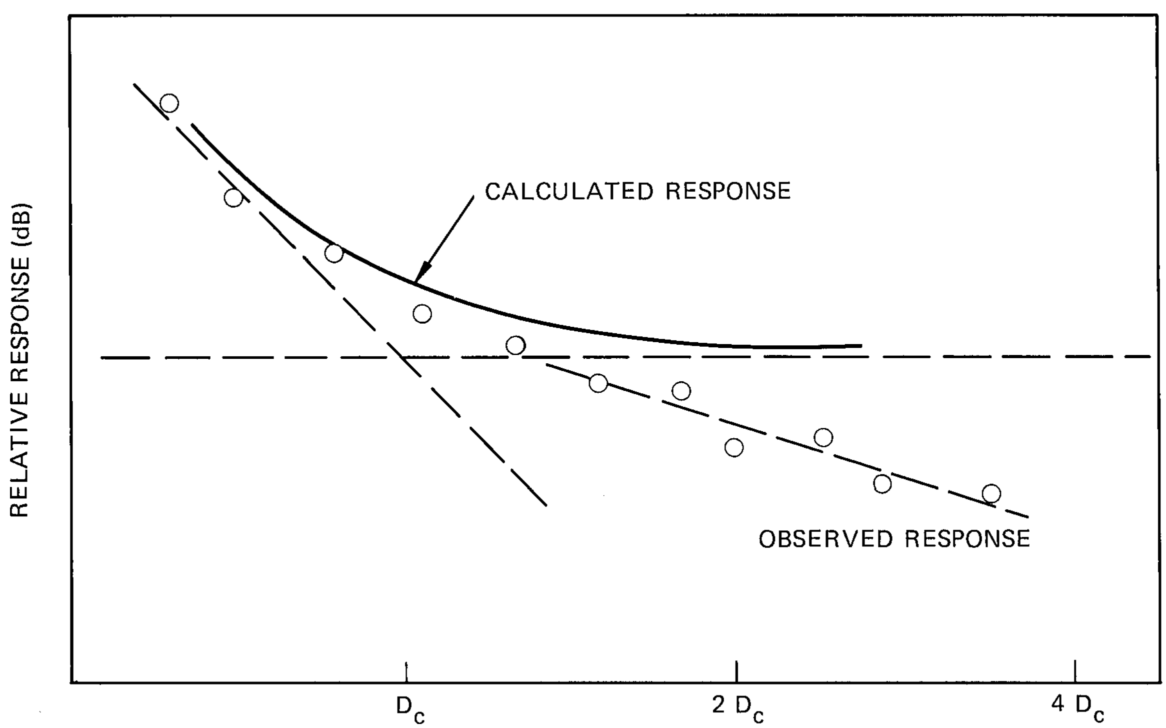


Figure 5-24. Attenuation with distance in a relatively dead room