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:

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

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

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

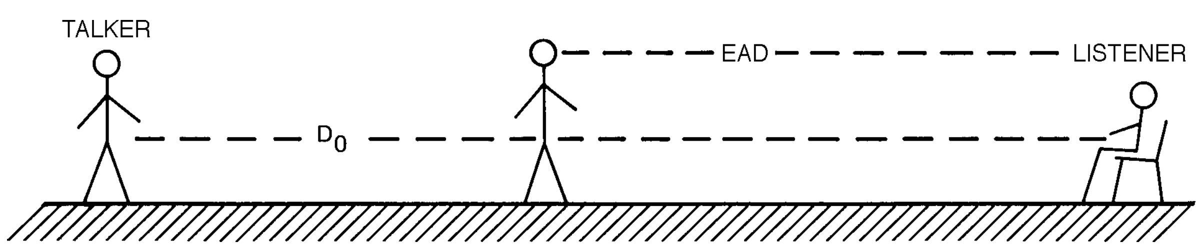


Figure 4-6. Concept of Effective Acoustical Dustance (EAD)

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.

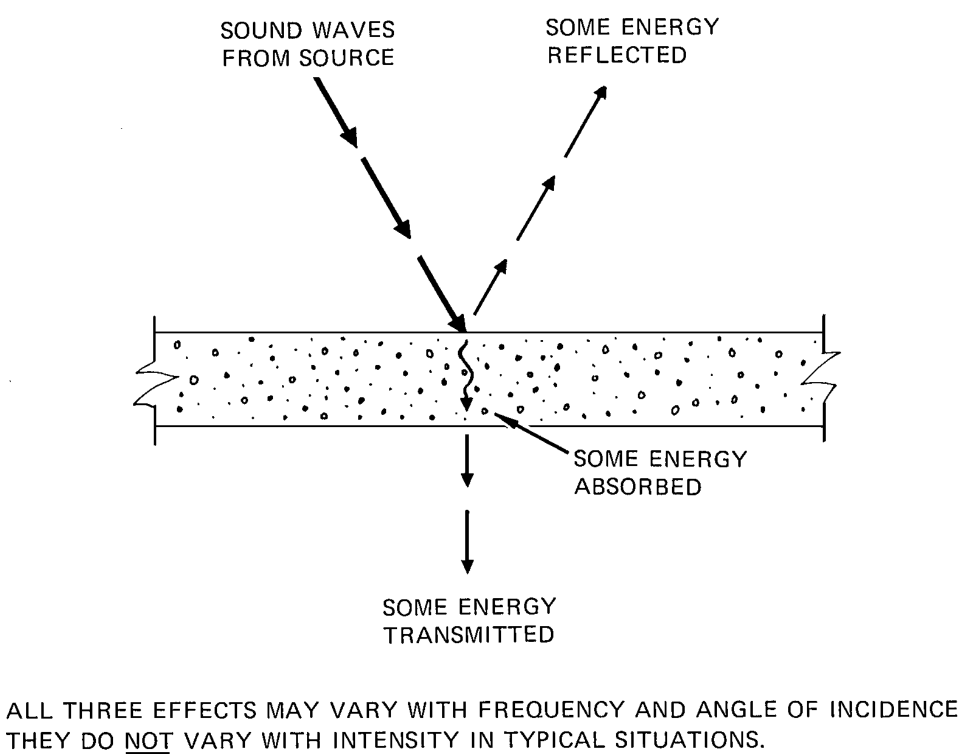


Figure 5-1. Sound impinging on a large boundary surface

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.)

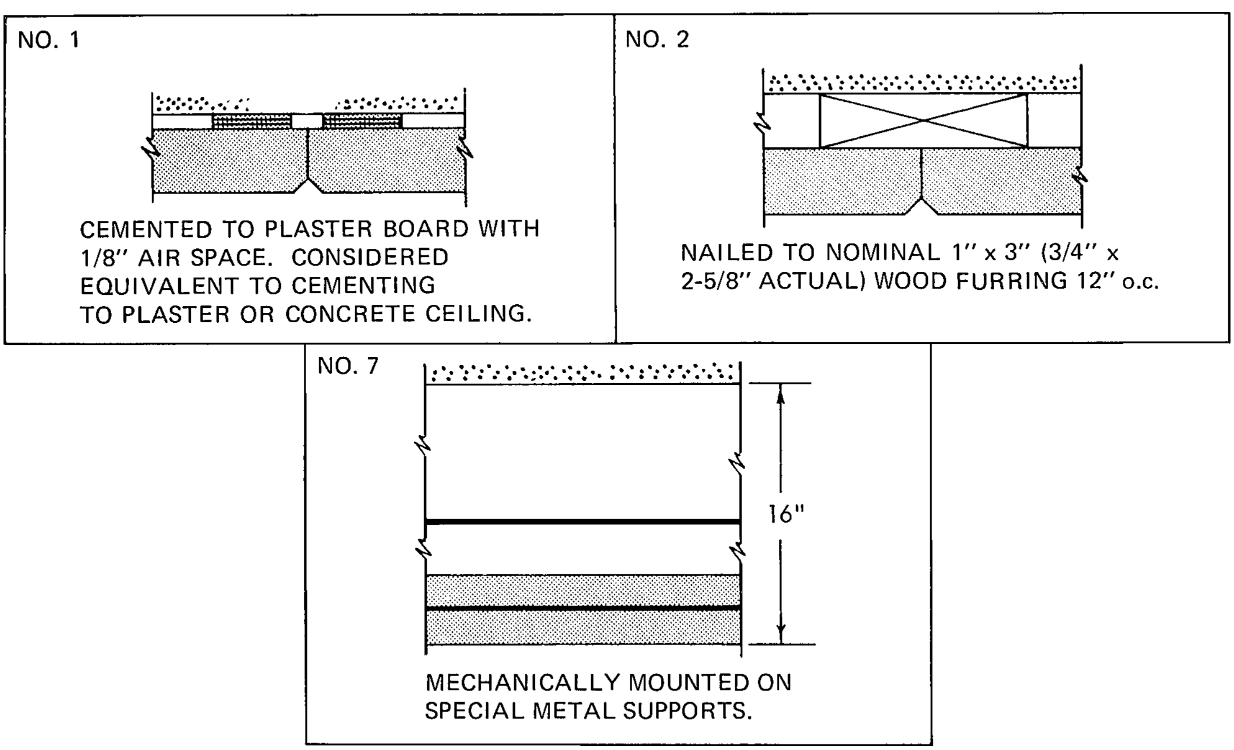


Figure 5-2. ASTM types of mounting (used in conducting sound absorption tests)

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

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

The use of the average absorption coefficient $\bar{\alpha}$ 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.

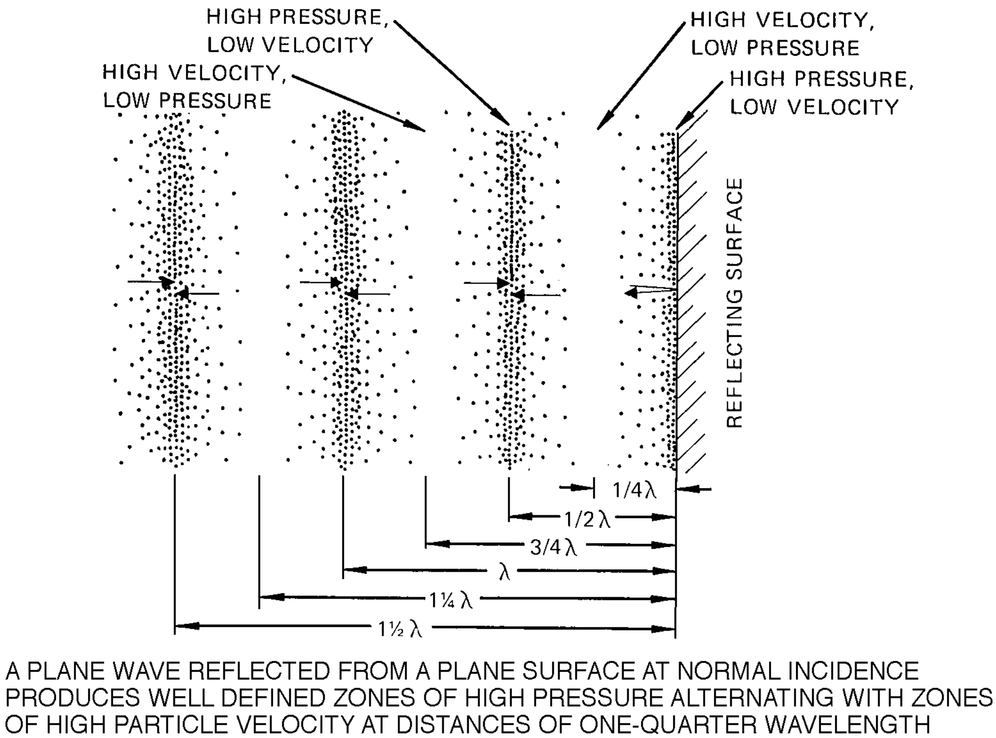


Figure 5-4. Interference pattern of sound reflected from a solid boundary

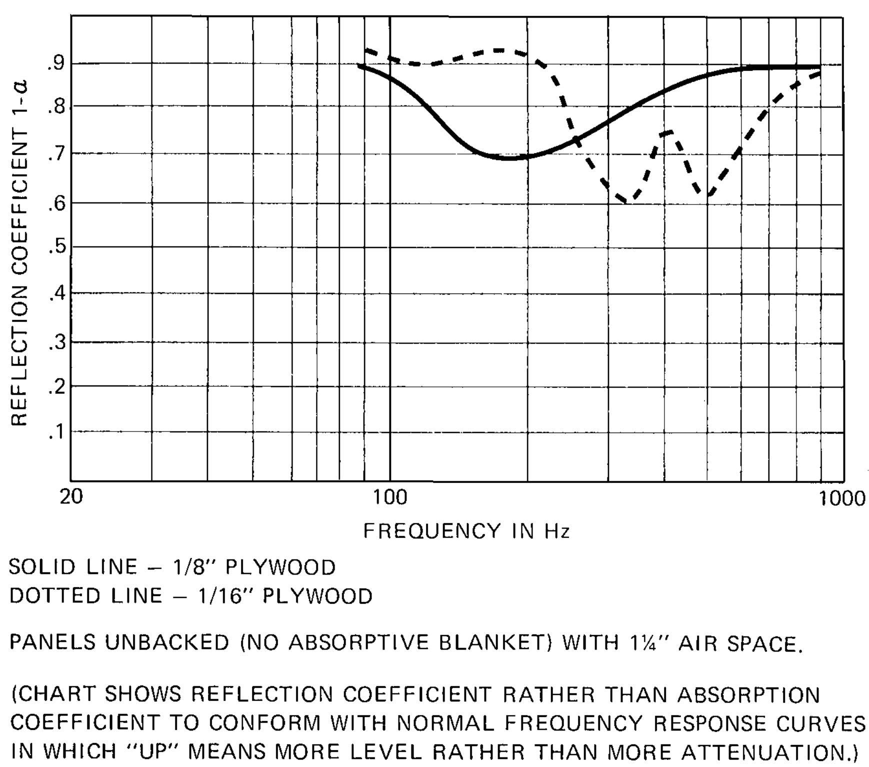


Figure 5-5. Reflectivity of thin plywood panels

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.

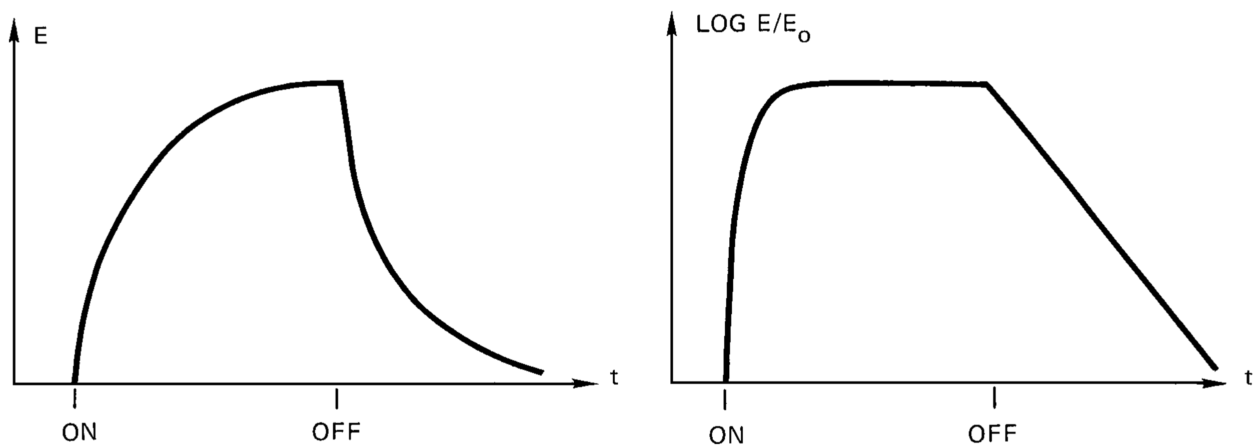


Figure 5-6. Idealized growth and decay of sound energy in an enclosure

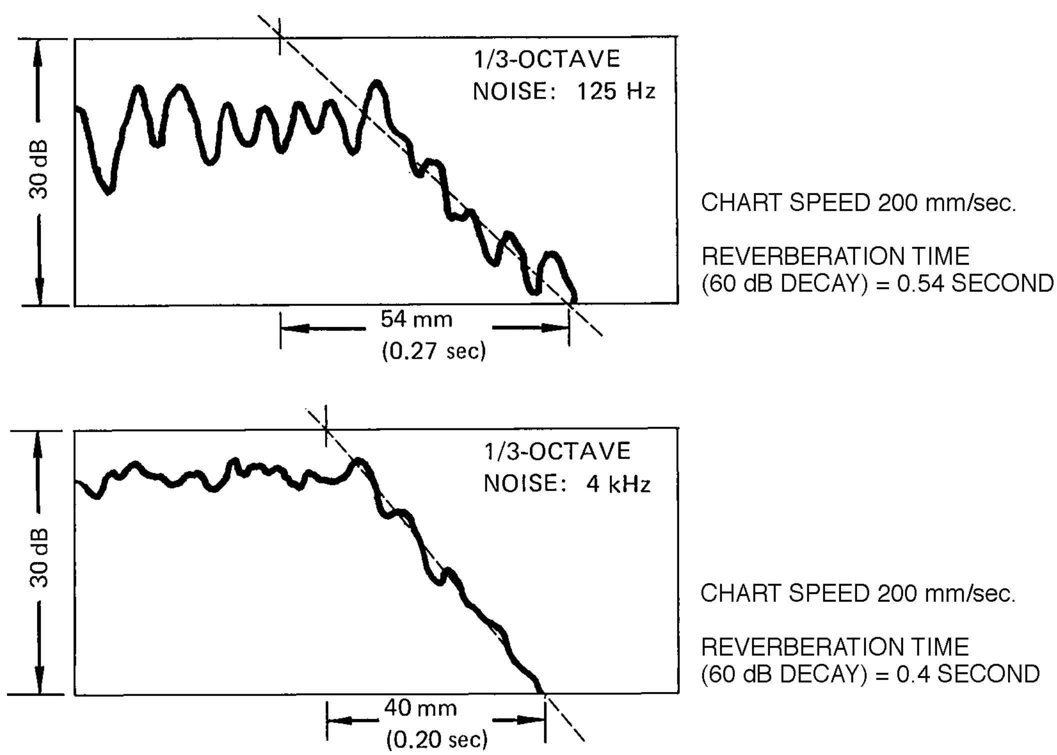


Figure 5-7. Actual chart recordings of decay of sound in a room

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 $\bar{\alpha}$ 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 = $66m^2 \times .09 = 5.9m^2$

$126m^2$ $\quad$ $24.9m^2$

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

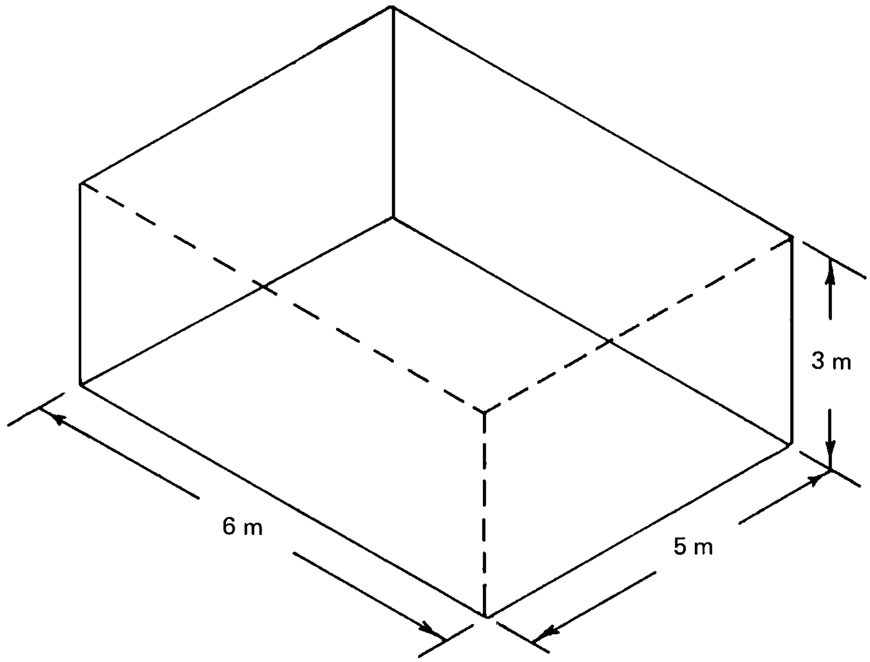


Figure 5-8. Calculating reverberation time

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, $\bar{\alpha}$. 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 $\bar{\alpha}$ 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 $\bar{\alpha}$. Each time sound is bounced off a new surface, its energy is decreased by a proportion determined by the average absorption coefficient $\bar{\alpha}$.

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 ($S\bar{\alpha}$) adds up to 24.9 absorption units. Therefore, the average absorption coefficient ($\bar{\alpha}$) 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%.

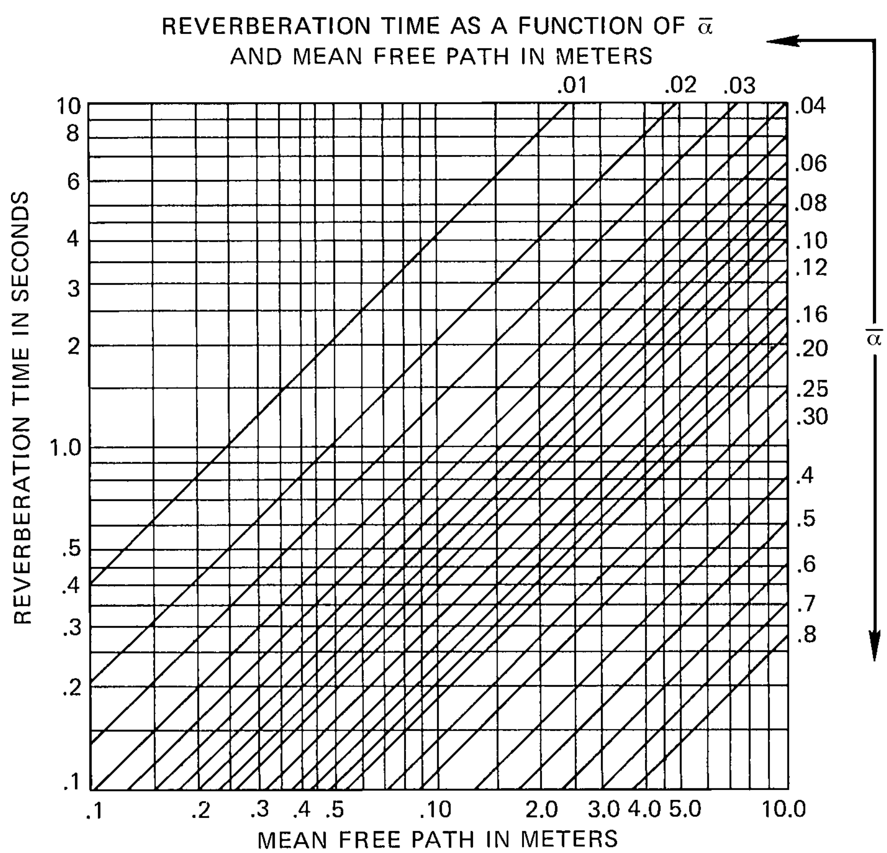


Figure 5-10. Reverberation time chart, SI units

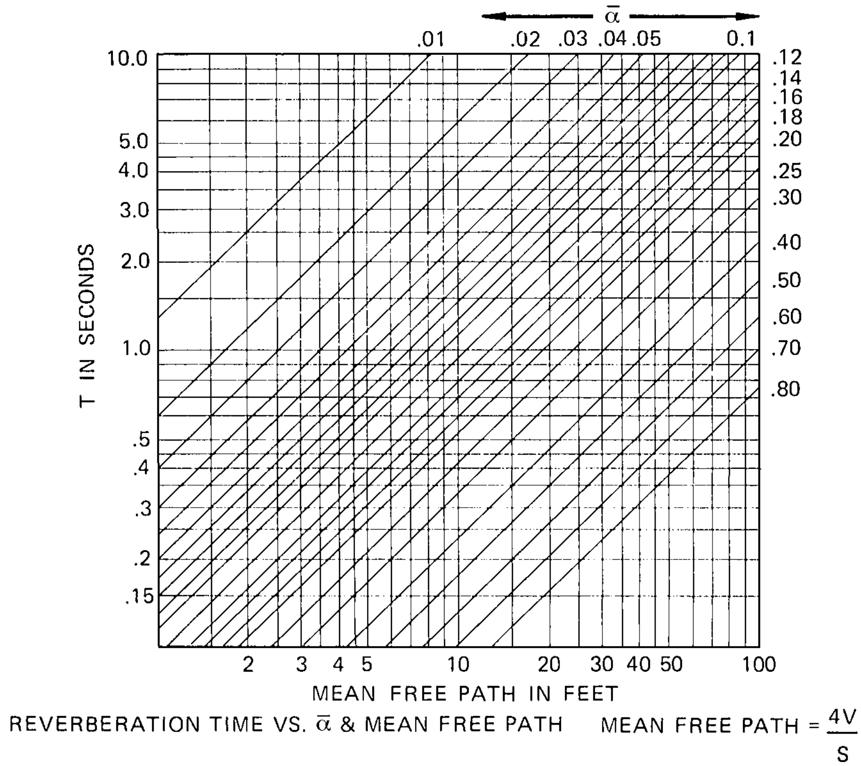


Figure 5-11. Reverberation time chart, English units

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.

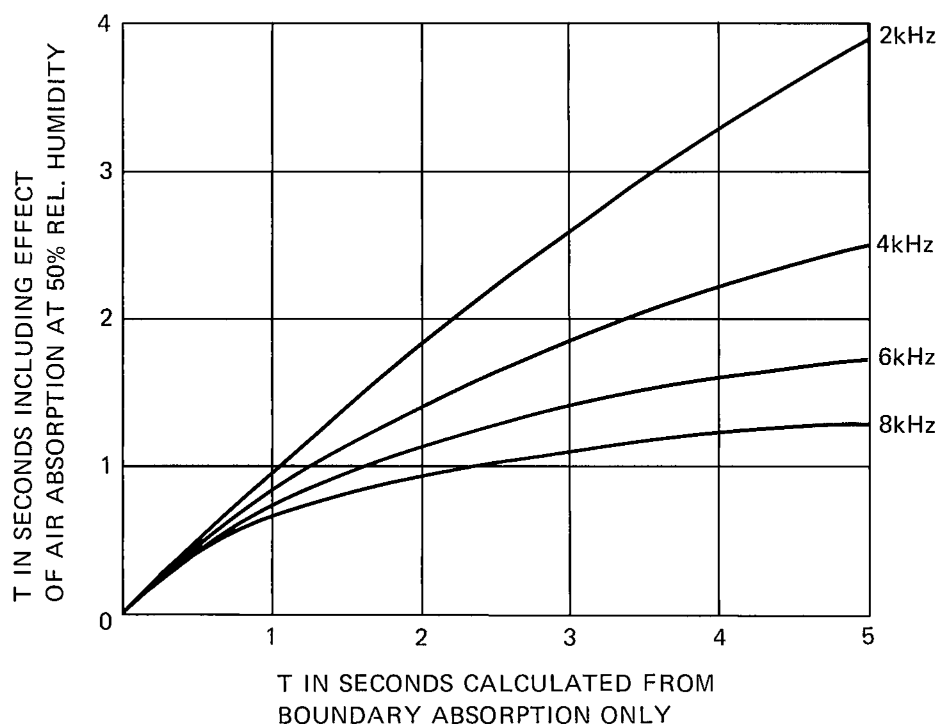


Figure 5-13. Effect of air absorption on calculated reverberation time

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*.

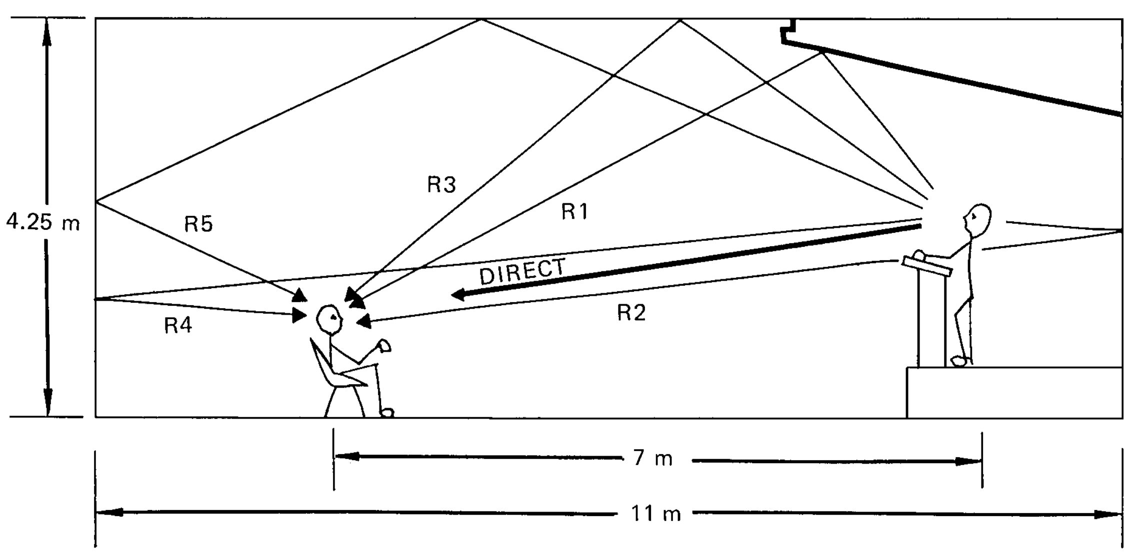
The distinction between early reflections and the later reverberation is usually made at some point between 20 and 30 milliseconds after the arrival of the direct sound. Most people with normal hearing find that early reflections are combined with the direct sound by the hearing mechanism, whereas later 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.

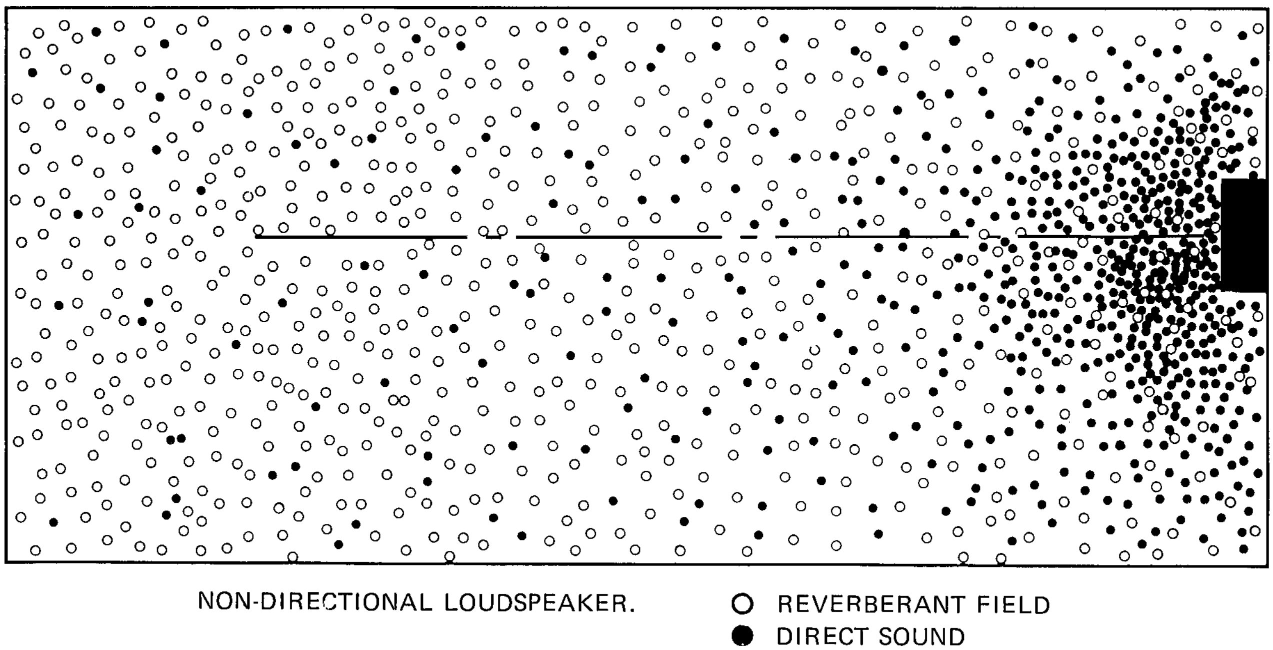


Figure 5-15. Direct and reverberant fields, non-directional loudspeaker

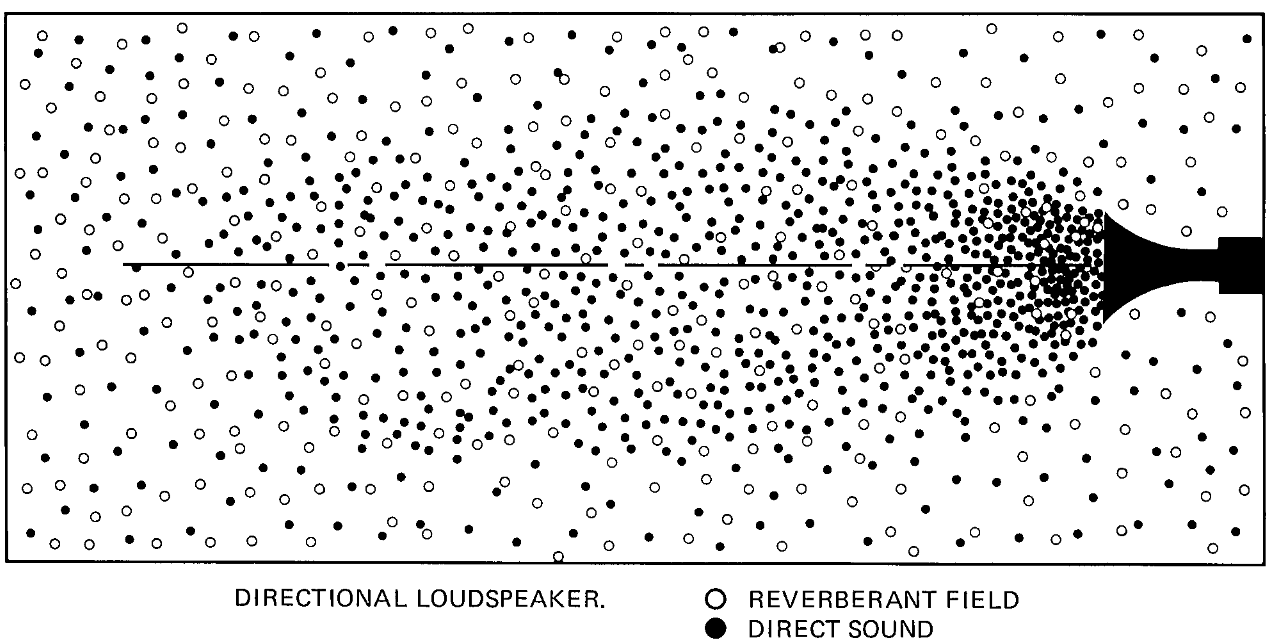


Figure 5-16. Direct and reverberant fields, directional loudspeaker

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.

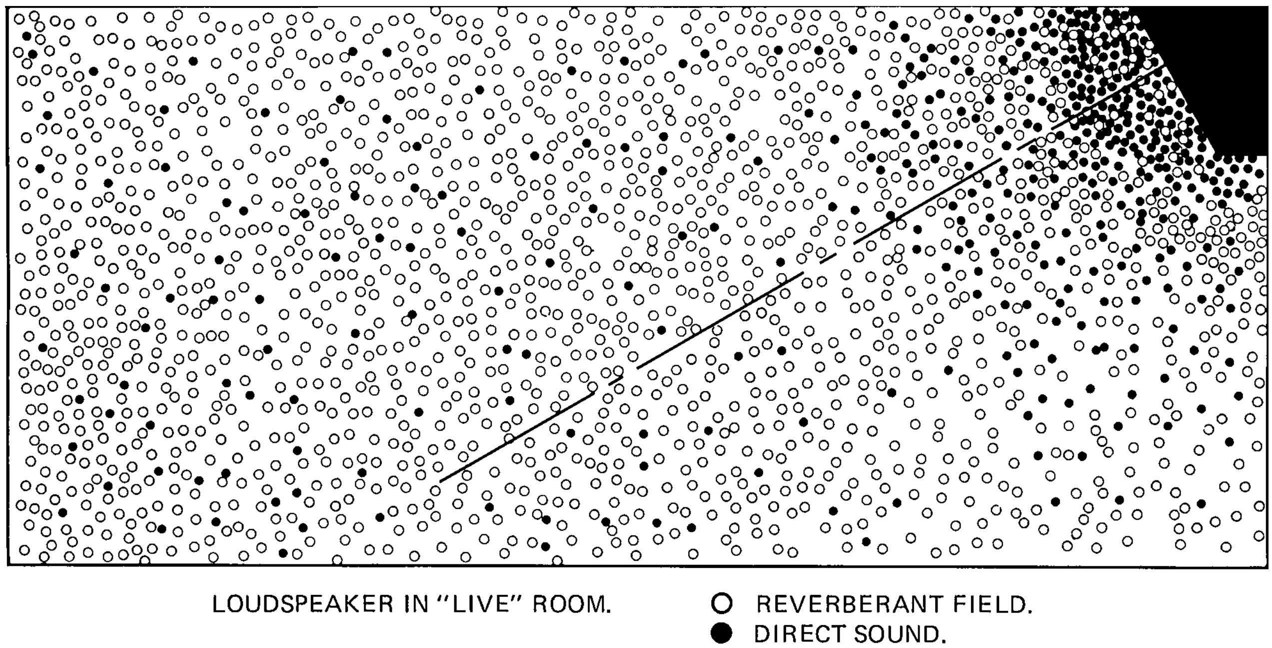


Figure 5-17. Direct and reverberant fields, live room

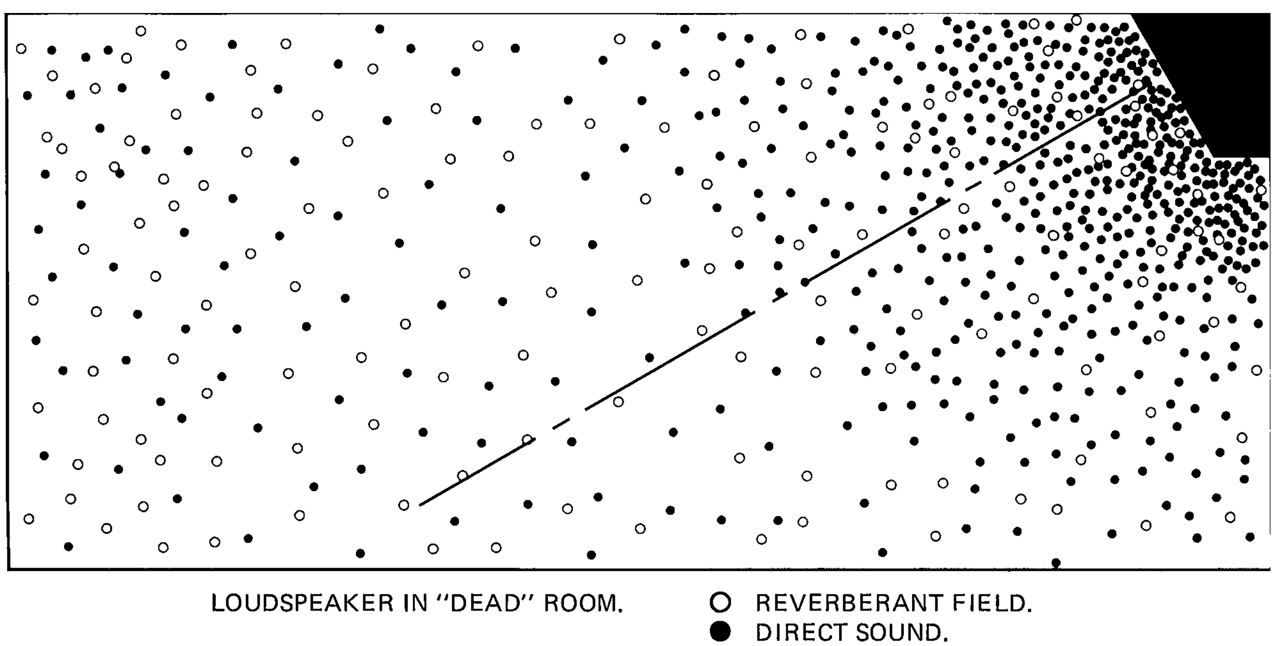


Figure 5-18. Direct and reverberant fields, dead room

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 [$R = S\bar{\alpha}/(1 - \bar{\alpha})$]. A small room constant indicates a very live room, and a large room constant describes a room having a great deal of absorption.

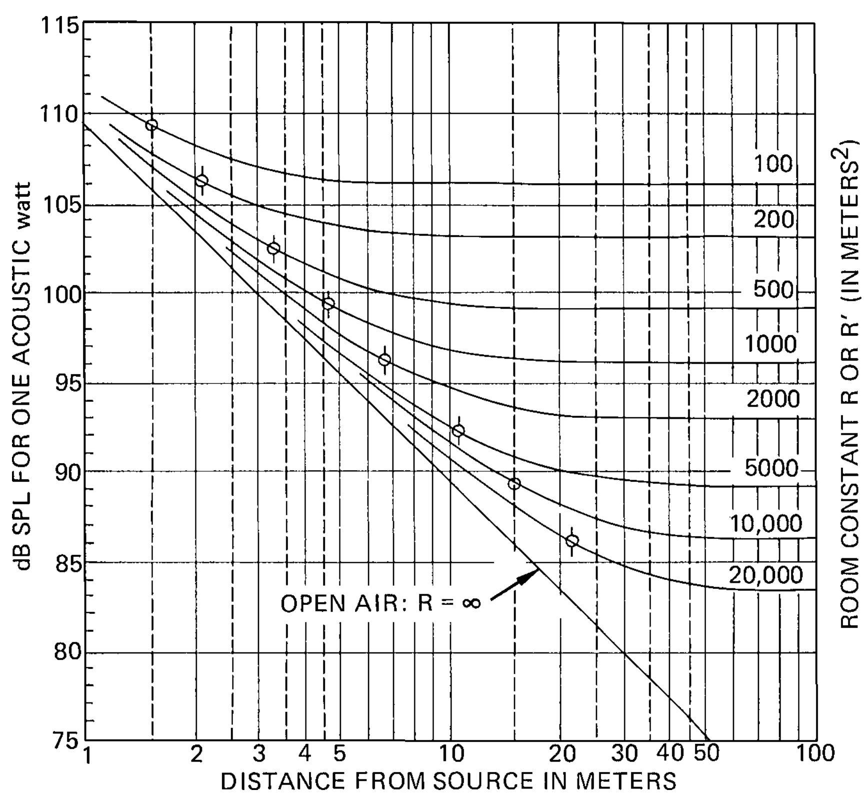


Figure 5-19. SPL (point source radiating one acoustic watt) vs. R and distance from source

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 $S\bar{\alpha}$.

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

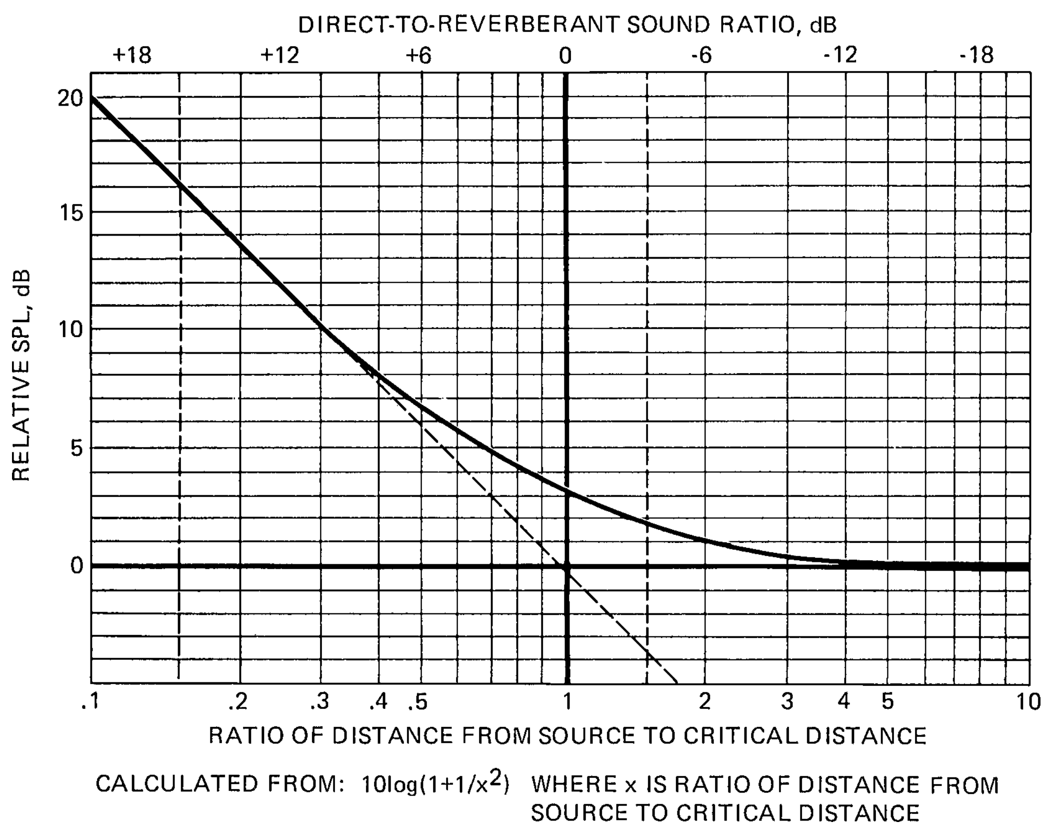


Figure 5-20. Relative SPL vs. distance from source in relation to critical distance

relationship is often indicated by the simple expression: $4W/cS\bar{\alpha}$. *W* represents the output of the sound source, and the familiar expression $S\bar{\alpha}$ 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 $\bar{\alpha}$. 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 $\bar{\alpha}$ 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 - $\bar{\alpha}$).*

This results in the equation:

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

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 $W(1 - \bar{\alpha})$. 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.

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

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

* 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.

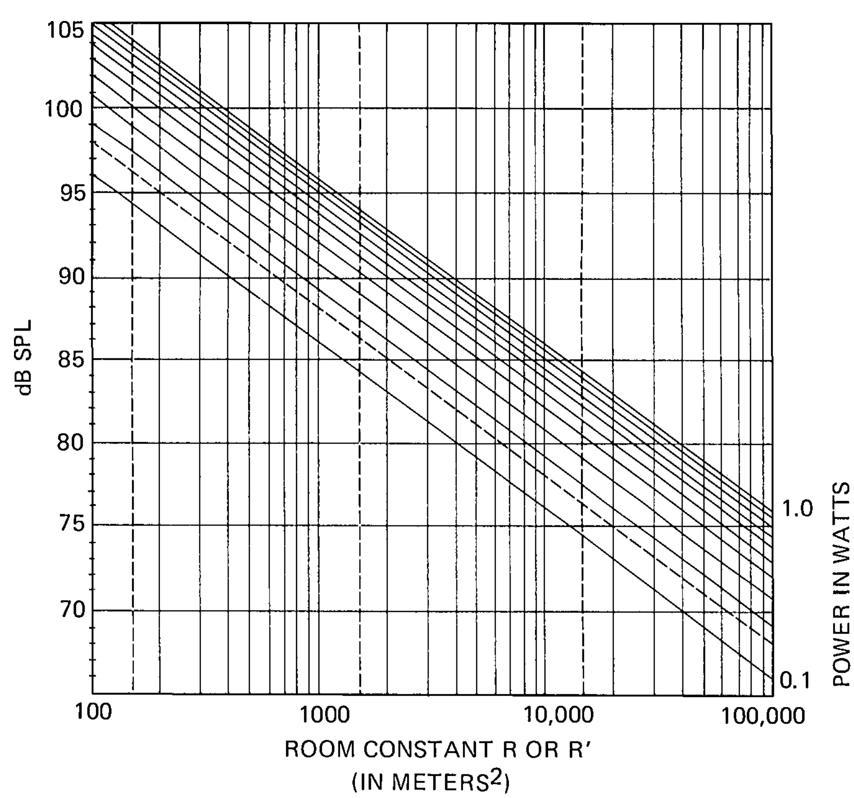


Figure 5-21. Steady-state reverberant field SPL vs. acoustic power and room constant

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 α'. The energy remaining after the first reflection must then be proportional to $(1 - \alpha')$. This allows us to write an expression for the effective room constant designated by the symbol *R'*:

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

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

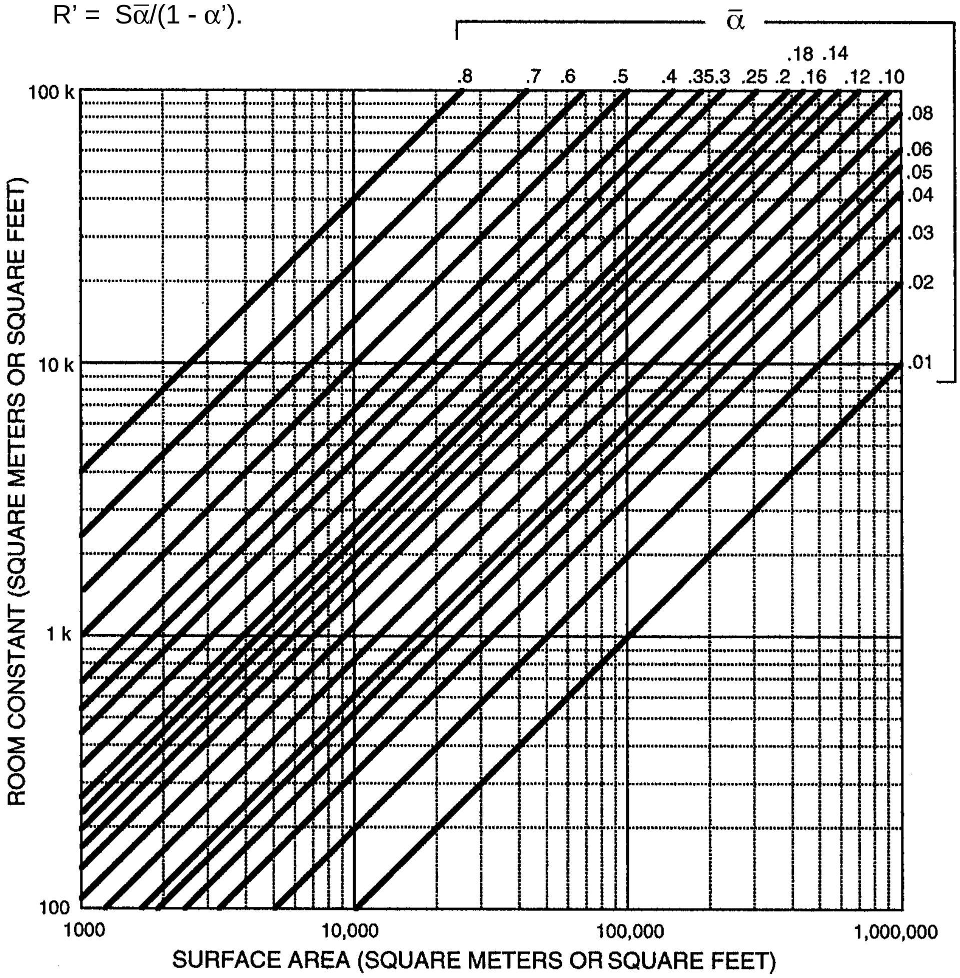


Figure 5-22. Room constant vs. surface area and $\bar{\alpha}$

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 $\bar{\alpha}$ 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.

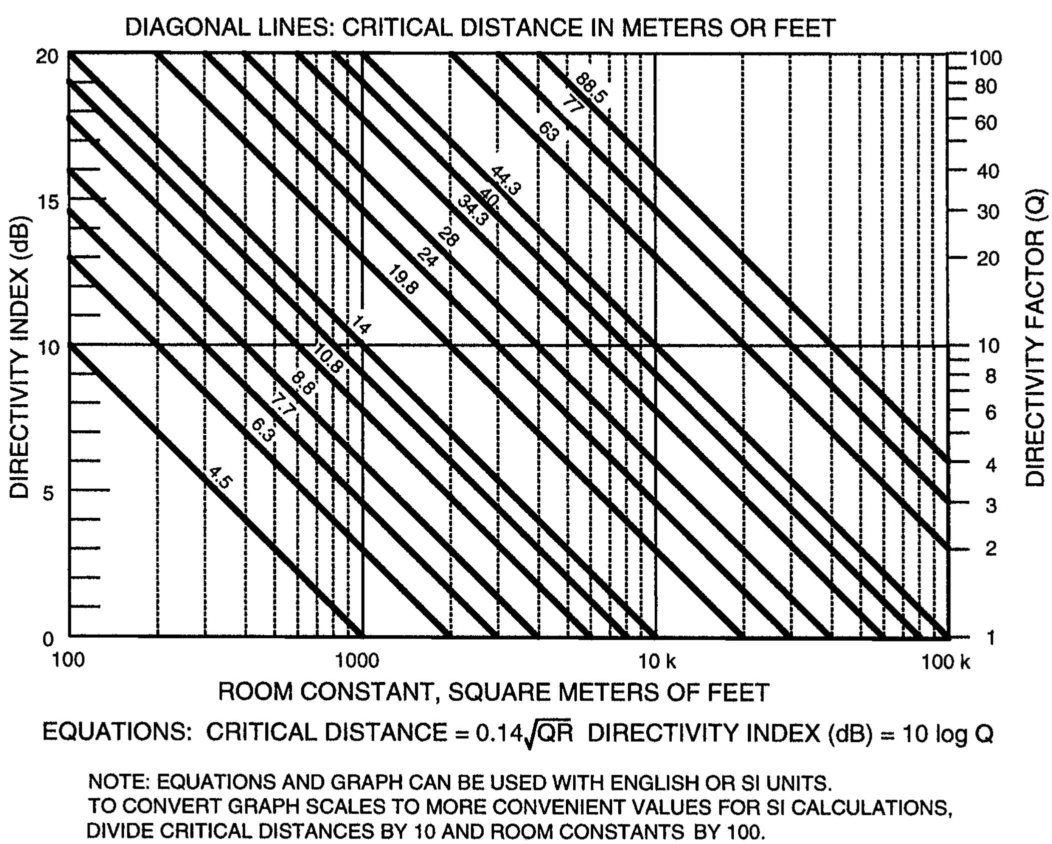


Figure 5-23. Critical distance as a function of room constant and directivity index or directivity factor

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.

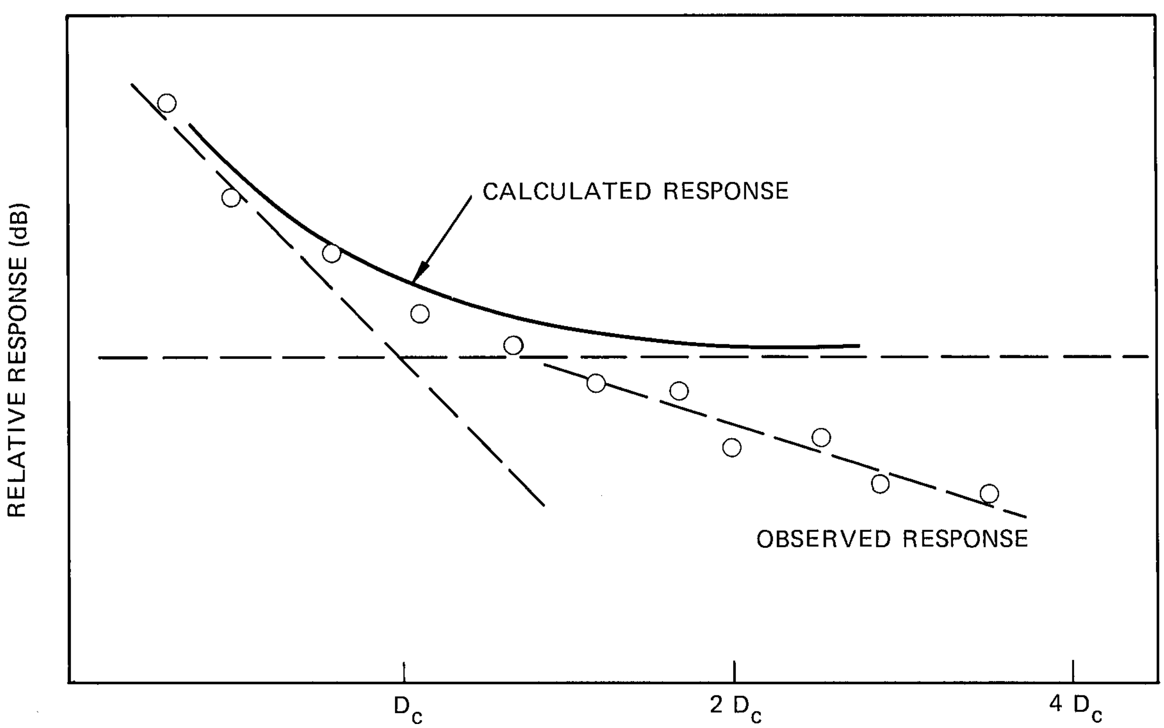


Figure 5-24. Attentuation with distance in a relatively dead room

Chapter 6: Behavior of Sound Systems Indoors

Introduction

The preceding five chapters have provided the groundwork on which this chapter is built. The “fine art and science” of sound reinforcement now begins to take shape, and many readers who have patiently worked their way through the earlier chapters will soon begin to appreciate the disciplines which have been stressed.

The date at which sound reinforcement grew from “public address by guesswork” to a methodical process in which performance specifications are worked out in advance was marked by the publication in 1969 of a paper titled “The Gain of a Sound System,” by C. P. and R. E. Boner (4). It describes a method of calculating potential sound system gain, and that method has since become a fundamental part of modern sound system design. The following discussion is based on the Boner paper. Certain points are expanded, and examples are given that require calculations more complicated than those in the original study. Also discussed is the relation between theoretically achievable system gain and practical operating parameters of typical indoor sound systems.

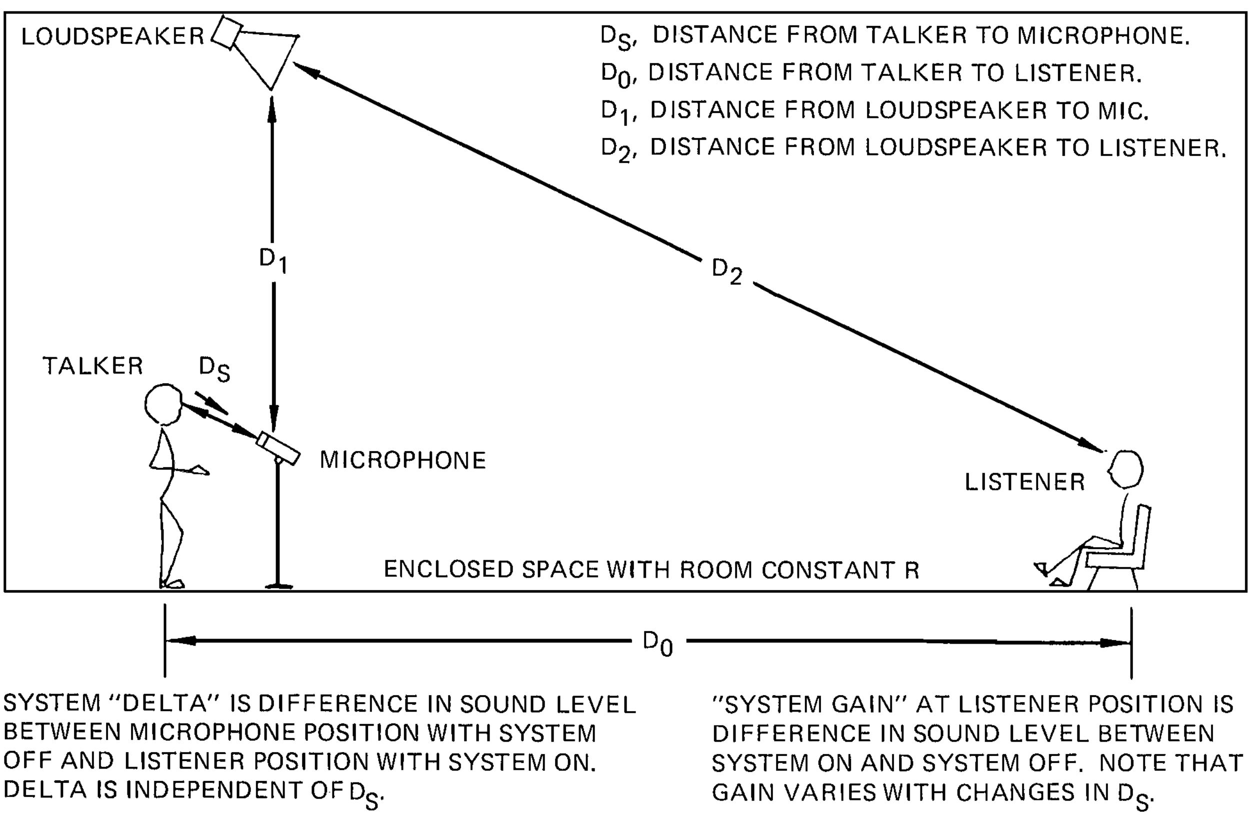


Figure 6-1. An indoor sound system

Acoustical Feedback and Potential System Gain

Just as in the outdoor case studied earlier, if we have a microphone/amplifier/loudspeaker combination in the same room and gradually turn up the gain of the amplifier to a point approaching sustained feedback, the electrical frequency response of the system changes with the gain setting. The effect results from an acoustic feedback path between the loudspeaker and the microphone. As a person talks into the microphone, the microphone hears not only the direct sound from the talker, but the reverberant field produced by the loudspeaker as well.

The purpose of using high-quality loudspeakers and microphones having smooth response characteristics, and sound system equalization (apart from achieving the desired tonal response) is to smooth out all of the potential feedback points so that they are evenly distributed across the audible frequency range. When this has been done, there should be as many negative feedback points as positive feedback points, and the positive feedback points should all reach the level of instability at about the same system gain.

We might expect this to average out in such a way that the level produced by the loudspeaker reaching the microphone can never be greater than that produced by the talker without causing sustained oscillation. In other words, we assume that the extra gain supplied by all the positive feedback spikes is just balanced out by the loss caused by all the negative feedback dips.

If the Boner criteria for optimum system geometry are followed, the microphone will be close to the talker so that it hears mostly direct sound from the talker. It will be far enough from the loudspeaker to be well into the reverberant field of the loudspeaker, so that direct sound from the loudspeaker is not an appreciable factor in triggering system feedback. Assuming that listeners are also in the reverberant field of the loudspeaker, it follows that the sound level in the listening area with the system turned on cannot be greater than that of the unaided talker at the microphone position with the system turned off. Using the Boner concept of system *delta*, the situation at maximum gain corresponds to a delta of unity. (Delta is defined as the difference in decibels between sound level at the system microphone with system off and the level in the audience area with system on. See Figure 6-1).

Although we have described these as conditions of maximum potential system gain, it is possible in practice to achieve a delta greater than unity. For example, if a directional microphone is used it can discriminate against the reverberant field and allow another 3 to 4 dB of system gain. Another possibility is to place the listener in the direct field of the loudspeaker, allowing a further increase in system gain. If the level of the reverberant field is lower in the performing area than in the listening area, additional system gain also results. This situation is described by the Boners as a room constant in the microphone area different from that in the seating area. Similar results may be noted in rooms having large floor areas, relatively low ceilings, and substantial sound absorption. In such rooms, as we have seen, sound from a point source tends to dwindle off beyond D_C at a rate of 2 or 3 dB for each doubling of distance rather than remaining constant in level.

Still another way to increase gain is to electrically suppress the positive feedback frequencies individually with very narrow bandwidth filters. If one could channel all energy into the negative feedback frequencies, the potential system gain would theoretically become infinite! Unfortunately, the acoustic feedback path is not stable enough to permit this degree of narrow-band equalization.

In all other situations, a gain setting is reached at which sustained oscillation occurs. By definition, maximum system gain is reached just below this point. However, the system cannot be operated satisfactorily at a point just below oscillation because of its unpleasant comb-filter response and the prolonged ringing caused by positive feedback peaks. To get back to reasonably flat electrical response and freedom from audible ringing, it usually is recommended that a properly equalized system be operated about 6 dB below its maximum gain point. Even an elaborately tuned system using narrow-band filters can seldom be operated at gains greater than 3 dB below sustained oscillation.

Sound Field Calculations for a Small Room

Consider the room shown in Figure 6-2. This is a typical small meeting room or classroom having a volume less than 80 m^3. The average absorption coefficient $\bar{\alpha}$ is 0.2. Total surface area is 111 m^2. The room constant, therefore, is 28 m^2.

From the previous chapter, we know how to calculate the critical distance for a person talking (nominal directivity index of 3 dB). In the example given, D_C for a source having a directivity index of 3 dB is 1 meter.

The figure also shows geometrical relationships among a talker, a listener, the talker's microphone and a simple wall-mounted loudspeaker having a directivity index of 6 dB along the axis pointed at the listener. The microphone is assumed to be omnidirectional.

Step 1: Calculate relative sound levels produced by the talker at microphone and listener.

We begin with the sound system off. Although the calculations can be performed using only relative levels, we will insert typical numbers to get a better feel for the process involved.

The microphone is .6 meter from the talker, and at this distance, the direct sound produces a level of about 70 dB. Since D_C for the unaided talker is only 1 meter, the microphone distance of .6 meter lies in the transition zone between the direct field and the reverberant field of the talker. By referring to Figure 6-3, we note that the combined sound levels of the reverberant field and the direct field at a distance of .6 meter must be about 1 dB greater than the direct field alone. Therefore, since we have assumed a level of 70 dB for the direct field only, the total sound level at the microphone must be 71 dB.

Next, we use a similar procedure to calculate the sound level at the listener's position produced by an unaided talker:

The listener is located 4.2 meters from the talker, more than 3 times the critical distance of 1 meter, and therefore, well into the reverberant field of the talker. We know that the sound level anywhere in the reverberant field is equal to that produced by the direct field alone at the critical distance. If the level produced by direct sound is 70 dB at a distance of .6 meter, it must be 4.6 dB less at a distance of 1 meter, or 65.4 dB, and the level of the reverberant field must also be 65.4 dB. The sound level produced by the unaided talker, at the listener's position, therefore is 65.4 dB.

At this point, let us consider two things about the process we are using. First, the definition of critical distance implies that sound level is to be measured with a random-incidence microphone. (For example, we have chosen a non-directional system microphone so that it indeed will "hear" the same sound field as that indicated by our calculations). Second, we have worked with fractions of decibels to avoid confusion, but it is important to remember that the confidence limits of our equations do not extend beyond whole decibel values, and that we must round off the answer at the end of our calculations.

Step 2:The sound field produced by the loudspeaker alone.

Now let us go back to our example and calculate the sound field produced by the loudspeaker. Our system microphone is still turned off and we are using an imaginary test signal for the calculations. We can save time by assuming that the test signal produces a sound level at the microphone of 71 dB — the same previously assumed for the unaided talker.

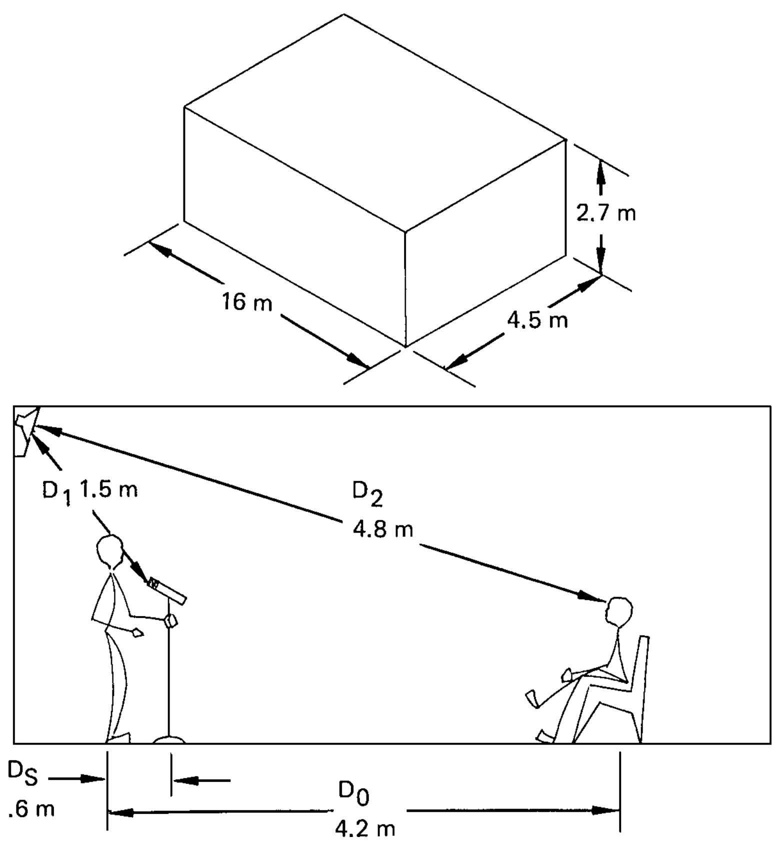


Figure 6-2. Indoor sound system gain calculations

The loudspeaker is mounted at the intersection of wall and ceiling. Its directivity index, therefore, is assumed to be 6 dB. In this room, the critical distance for the loudspeaker is 1.4 meters. This is almost the same as the distance from the loudspeaker to the microphone. Since the microphone is located at the loudspeaker's critical distance, and since we have assumed a level of 71 dB for the total sound field at this point, the direct field at the microphone must equal 71 dB minus 3 dB, or 68 dB.

The listener is 4.8 meters from the loudspeaker (more than 3 times the critical distance) and therefore, well into the reverberant field of the loudspeaker. We know that the level in the reverberant field must equal the level of the direct field alone at the critical distance. The sound level at the listener's position produced by the loudspeaker must, therefore, be 68 dB.

Step 3: Potential acoustic gain is now considered.

Since we deliberately set up the example to represent the condition of maximum theoretical gain for a properly equalized system, we can use these same figures to calculate the difference in level at the listener's position between the unaided talker and the talker operating with the system turned on. We have calculated that the unaided talker produces a level at the listener's position of 65.4 dB. We have also calculated that the level produced by the loudspeaker at the listener's position is 68 dB. The acoustic gain of the system for this specific set of conditions must be the difference between the two, or only 2.6 dB. Obviously such a sound reinforcement system is not worth turning on in the first place.

Note that system acoustical gain is dependent upon the distance from the microphone to the talker. A more general concept is that of system delta. According to the Boner paper, the maximum theoretical Δ of a properly equalized system is unity. In our example, Δ works out to be -3 dB. Why?

The Boners emphasize that for maximum system gain the microphone must be in the direct field of the talker and in the reverberant field of the loudspeaker. But in our example, the microphone is not quite in the direct field of the talker and is located at the critical distance of the loudspeaker! To achieve more gain, we might move the microphone to a distance .3 meter from the talker and use a more directional loudspeaker. This would result in a 3 dB increase in Δ and a potential acoustic gain at the listener's position of about 9 dB.

In practice, however, we cannot operate the system at a point just below sustained feedback. Even if we modify the system as described above, our practical working gain will only be about 3 dB. Our calculations merely prove what we could have guessed in advance: in a room this small, where an unaided talker can easily produce a level of 65 dB throughout the room, a sound system is unnecessary and of no practical benefit.

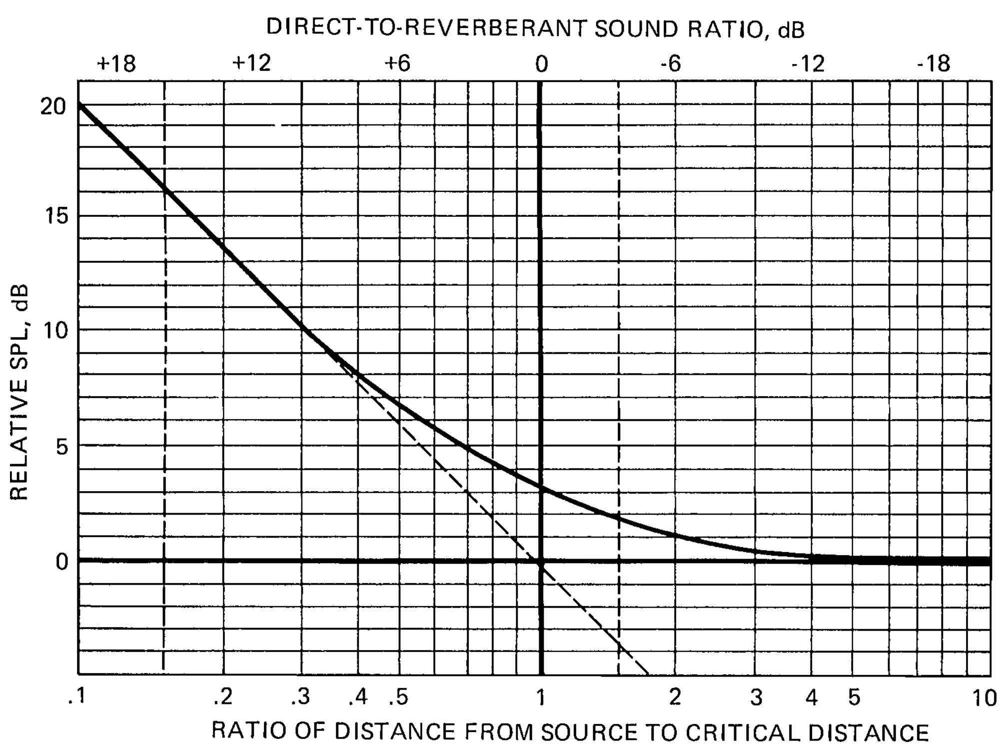


Figure 6-3. Relative SPL vs. distance from source in relation to critical distance

Calculations for a Medium-Size Room

Consider a more typical (and more complicated) situation in which the sound system is used in a larger room and in which a directional microphone is employed. Figures 6-4 and 6-5 show a room having a volume of 918 m^3, a total surface area of 630 m^2 and $\bar{\alpha} = 0.15$.

The first step is to calculate the room constant, and we would do well to examine the actual distribution of absorptive material in the room. Chapter 5 explains why the effective room constant R’ in a particular situation may vary substantially from the figure given by the equation $R = S\bar{\alpha}/(1 - \bar{\alpha})$. Rather than complicate the example, however, assume that the equation really does work and that the room constant is about 110 m^2.

The next step is to calculate critical distances for the talker and the loudspeaker. Since the loudspeaker does not have a uniform radiation pattern, we must calculate its critical distance at the particular angle in which we are interested. Figure 6-5 shows the distances involved and the geometrical relationships between talker, microphone, loudspeaker and listener.

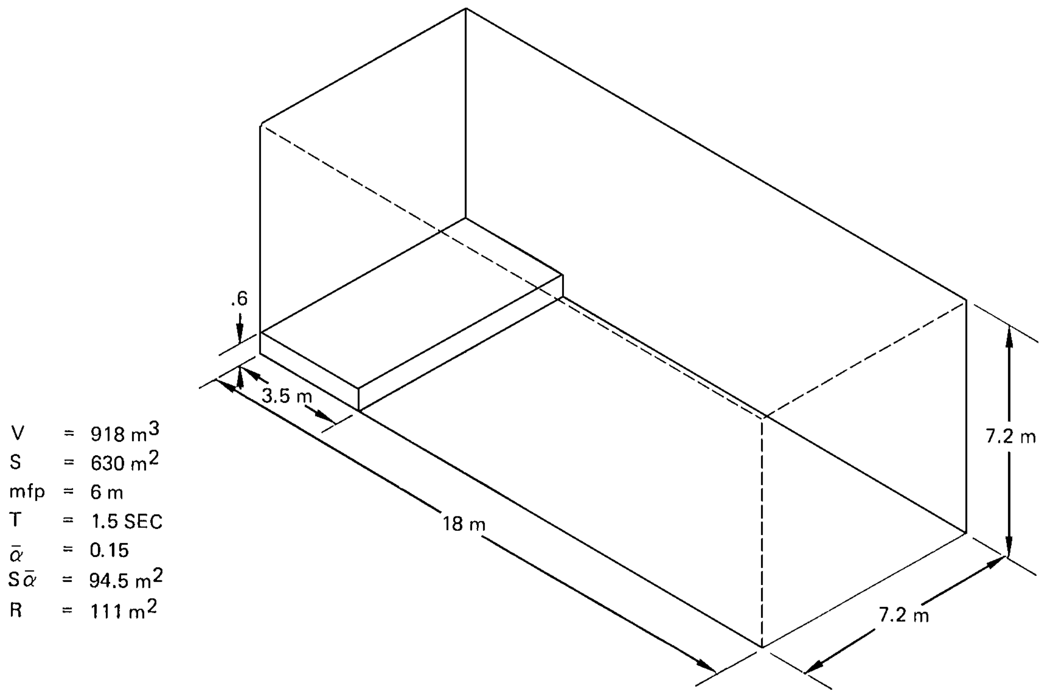


Figure 6-4. A sound system in a medium-size room

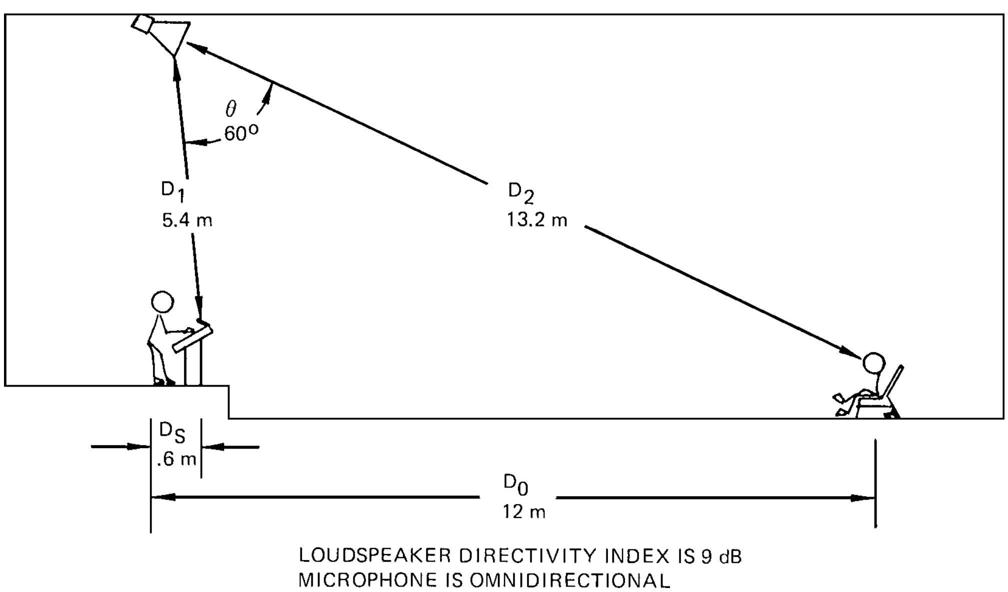


Figure 6-5. Sound system in a medium-size room, gain calculations

In the frequency range of interest, the loudspeaker is assumed to have a directivity index along its primary axis of 9 dB. From Figure 6-6 we find the corresponding critical distance of 4.2 meters. The loudspeaker's directivity index at a vertical angle of 60° is assumed to be -3 dB, with a corresponding critical distance of 1 meter. The unaided talker has a directivity index of 3 dB and his critical distance must therefore be 2 meters.

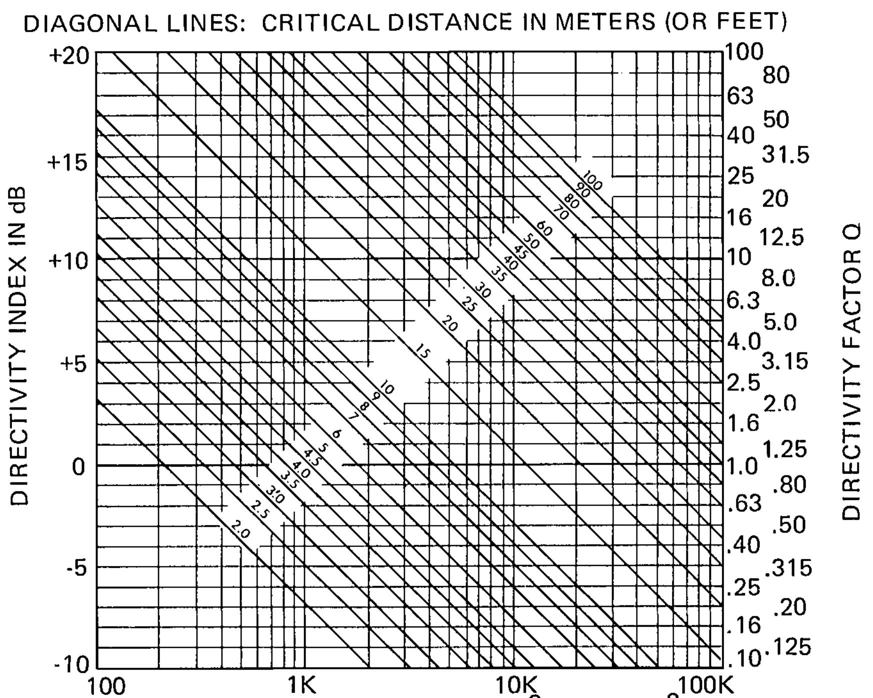
Our next step in calculating system gain is to find the difference in level produced by an unaided talker at the listener position as contrasted with that at the microphone position. In this example the listener is 12 meters from the talker and the microphone again is .6 meters away.

The talker's critical distance of 2 meters is more than 3 times the microphone distance. Therefore, the microphone is well in the direct field of the talker. The listener is more than 3 times the critical distance and is well into the reverberant field of the unaided talker. Setting the level produced by the unaided talker at 70 dB for a distance of .6 meters, we calculate that the direct field at D_C must be 60 dB, and since the reverberant field must also equal 60 dB, the level produced by the unaided talker at the listener's position is 60 dB.

The third step is to make similar calculations for the loudspeaker alone. The listener is located on the major axis of the loudspeaker and is more than 3 times the critical distance of 4.2 meters. The microphone is located at a vertical angle of 60 degrees from the loudspeaker's major axis, and also is more than 3 times the critical distance (at this angle) of 1 meter. Both the listener and the microphone are located in the reverberant field of the loudspeaker.

If the sound level produced by the loudspeaker at the microphone can be no greater than 70 dB (the same level as the talker) then the level produced by the loudspeaker at the listener's position must also be 70 dB, since both are in the reverberant field.

Having established these relationships we know that the talker produces a level at the listener's position of 60 dB with the sound system off and 70 dB with the sound system on, or a maximum potential gain of 10 dB. Allowing 6 dB headroom in a properly equalized system, we still realize 4 dB gain at the listener's position, and the sound system can be said to provide a small but perceptible increase in sound level.



EQUATIONS:

CRITICAL DISTANCE = $0.14\sqrt{QR}$

DIRECTIVITY INDEX IN dB = $10 \log Q$

NOTE: EQUATIONS AND CHART CAN BE USED WITH ENGLISH OR METRIC UNITS. TO CONVERT CHART SCALES TO MORE CONVENIENT VALUES FOR METRIC CALCULATIONS, DIVIDE CRITICAL DISTANCES BY 10 AND ROOM CONSTANTS BY 100.

Figure 6-6. Critical distance as a function of room constant and directivity index or directivity factor

However, all of the preceding calculations have assumed that the microphone is an omnidirectional unit. What happens if we substitute a directional microphone? Figure 6-7 shows the additional geometrical relationships needed to calculate the increase in gain produced by a directional microphone.

Note that the distance from talker to microphone is still .6 meters and that the talker is assumed to be located along the major axis of the microphone. The loudspeaker is located 5.4 meters from the microphone along an angle of 75° from the major axis.

Figure 6-7 also shows a typical cardioid pattern for a directional microphone. The directivity index of such a microphone along its major axis is about 5 dB.

Since the talker is located on the major axis of the microphone, it "hears" his signal 5 dB louder than the random incidence reverberant field. In theory this should increase potential system gain by a factor of 5 dB.

But we must also consider the microphone's directional characteristics with relation to the loudspeaker. If the directivity index of the microphone at 0° is 5 dB, the polar pattern indicates that its directivity index at 75° must be about 3 dB. This tells us that even though the loudspeaker is 75° off the major axis of the microphone, it still provides 3 dB of discrimination *in favor* of the direct sound from the loudspeaker.

We know that the loudspeaker's directivity index is -3 dB along the axis between the loudspeaker and the microphone. We also know that the microphone's directivity index along this axis is +3 dB. The combined directivity indices along this axis must therefore, be 0 dB and we can find the equivalent critical distance from Figure 6-6.

The combined critical distance of loudspeaker and microphone along their common axis is about 1.3 meters. Since the distance between the two is more than 3 times this figure, the microphone still lies within the reverberant field of the loudspeaker. Using the directional microphone should therefore allow an increase in potential system gain before feedback of about 5 dB. (In practice, little more than 3 or 4 dB of additional gain can be achieved.)

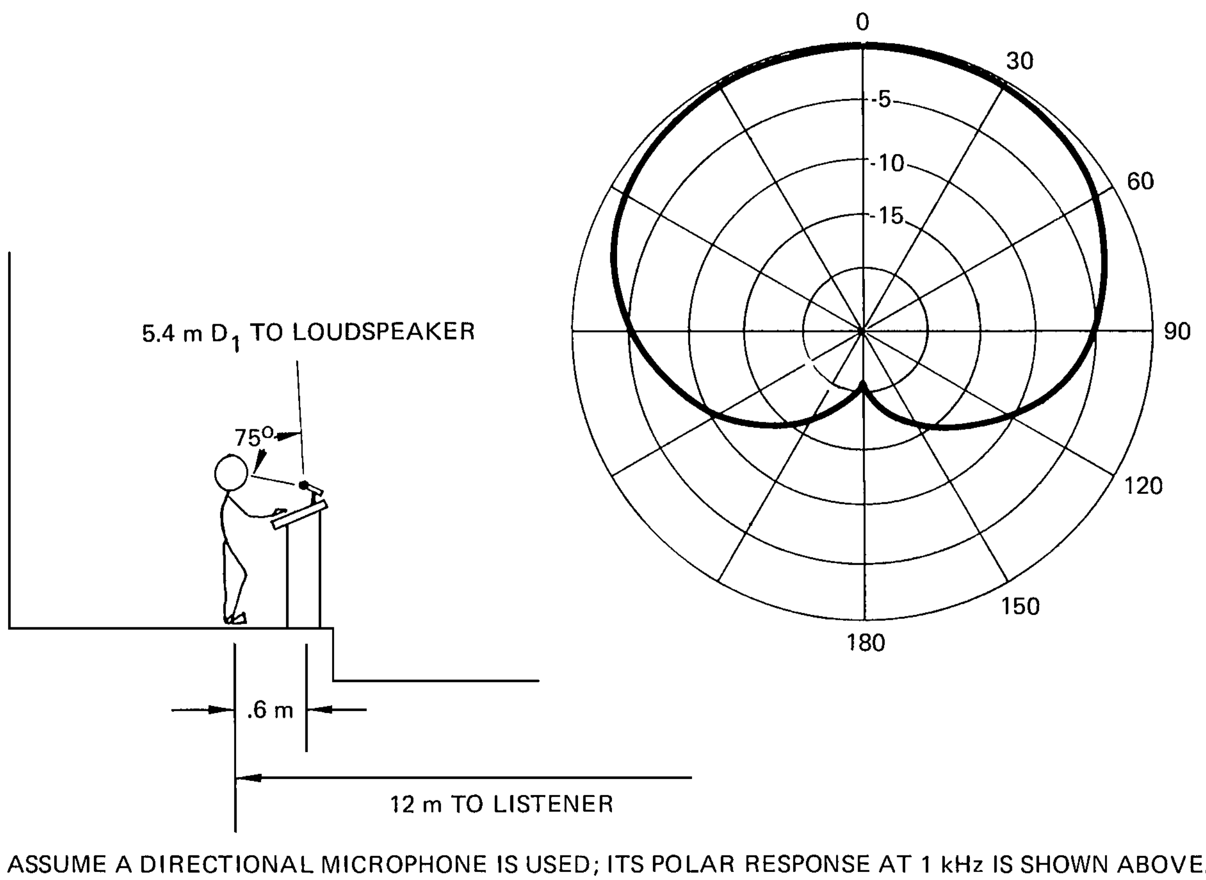


Figure 6-7. Characteristics of a cardioid microphone

Calculations for a Distributed Loudspeaker System

Figure 6-8 shows a moderate-size meeting room or lecture room. Its volume is 485 m^3, surface area is about 440 m^2, and $\bar{\alpha}$ is 0.2 when the room is empty. For an unaided talker in the empty room, R is 110 m^2. However, when the room is fully occupied, $\bar{\alpha}$ increases to 0.4 and the corresponding room constant is 293 m^2. We calculate the critical distance for the unaided talker (directivity index of 3 dB) to be 2 meters in an empty room and 3.4 meters when the room is full.

The room is provided with a sound system diagrammed in Figure 6-9. Forty loudspeakers are mounted in the ceiling on 1.5 meter centers to give smooth pattern overlap up into the 4 kHz region. Coverage at ear level varies only 2 or 3 dB through the entire floor area.

The usual definitions of critical distance and direct-to-reverberant ratio are ambiguous for this kind of loudspeaker array. Here, however, we are interested only in potential acoustic gain, and the ambiguities can be ignored. We already have stated that the loudspeaker array lays down a uniform blanket of sound across the room. The relative directional and temporal components of the sound field do not enter into gain calculations.

An omnidirectional microphone is located .6 meters from the talker, less than 1/3 D_C. No matter how many people are present, the microphone is in the direct field of the talker.

The farthest listener is 9 meters from the talker, more than three times D_C when the room is empty, and just about three times D_C when the room is full.

If the unaided talker produces 70 dB sound level at the microphone with the system off, and if the amplified sound level can be no greater than 70 dB at the microphone with the system on, then the maximum level is 70 dB everywhere in the room.

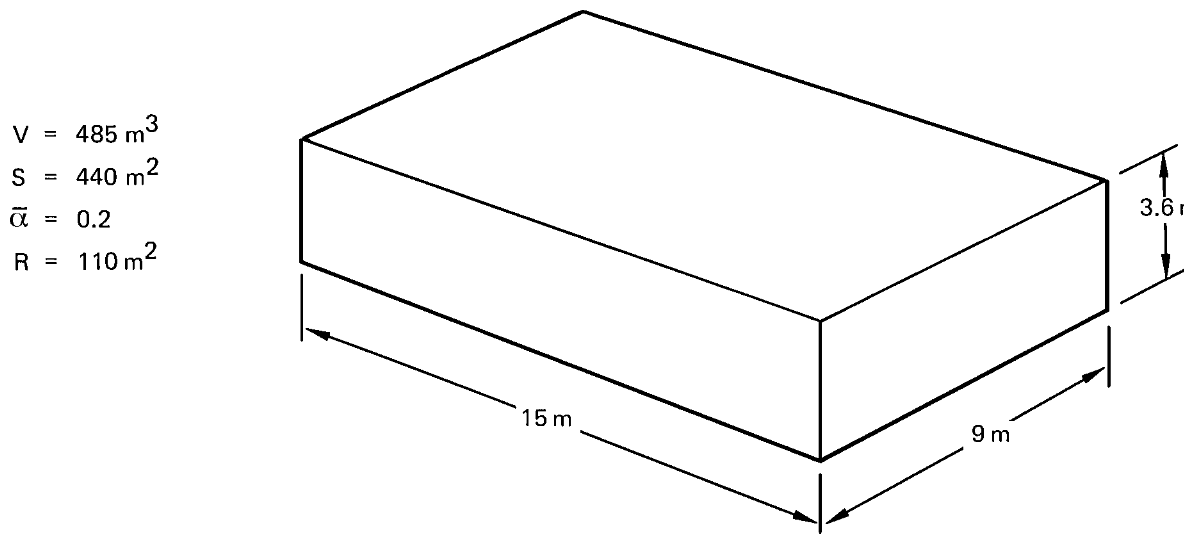


Figure 6-8. A moderate-size lecture room

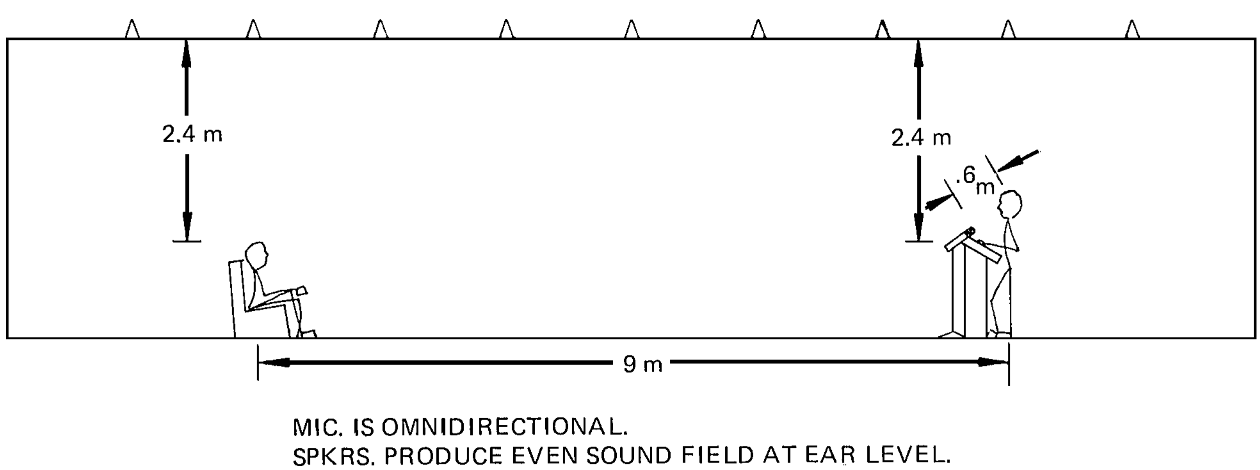


Figure 6-9. Sound system in a medium-size lecture room

From our calculations of critical distances, we see that the unaided talker will produce a sound level at the listener of 59 dB in an empty room and about 55 dB with a full audience. For a usable working delta of -6 dB, the calculated acoustic gain at the listener's position is about 5 dB in an empty room and about 9 dB when full.

Can we get more gain by turning off the loudspeaker directly over the microphone? Not in a densely packed array such as this. The loudspeakers are mounted close together to produce a uniform sound field at ear level. As a result, the contribution of any one loudspeaker is relatively small. However, by turning off *all* the loudspeakers in the performing area and covering only the audience, some increase in system gain may be realized.

In the example just given, each loudspeaker is assumed to have a directivity index in the speech frequency region of +6 dB at 0°, +3 dB at 45°, and 0 dB at 60°. Suppose we use only the 25 loudspeakers over the audience and turn off the 15 loudspeakers in the front of the room. In theory, the increase in potential gain is only 1 dB with a single listener or 2 dB when the audience area is filled. Even if we allow for the probability that most of the direct sound will be absorbed by the audience, it is unlikely that the gain increase will be more than 3 dB.

The calculations required to arrive at these conclusions are tedious but not difficult. The relative direct sound contribution from each of the loudspeakers at microphone and listener locations is calculated from knowledge of polar patterns and distances. By setting an arbitrary acoustic output per loudspeaker, it is then possible to estimate the sound level produced throughout the room by generally reflected sound (reverberant field) and that produced by reflected plus quasi-direct sound.

System Gain vs. Frequency Response

In the preceding examples we have not defined the frequency range in which gain calculations are to be made. In most sound systems the main reason for worrying about system gain is to make sure that the voice of a person talking can be amplified sufficiently to reach a comfortable listening level in all parts of the seating area. Therefore, the most important frequency band for calculating gain is that which contributes primarily to speech intelligibility: the region between 500 and 4000 Hz.

Below 500 Hz the response of the system can be gradually shelved, or attenuated, without seriously degrading the quality of speech. Above 4 kHz sound systems tend to take care of themselves, due to the increase in overall acoustical sound absorption. At very high frequencies, most environments are substantially absorptive, the air itself contributes considerable acoustical absorption and loudspeaker systems tend to become directional. These factors make it highly unusual to encounter feedback frequencies much above 2500 Hz.

To make sure that a sound reinforcement system will successfully amplify speech, it is a good idea to make gain calculations in at least two frequency bands. In a well-designed system, if calculations are made for the regions centered at 1 kHz and 4 kHz, chances are that no unforeseen problems in achieving desired system gain will be encountered.

However, the region below 500 Hz cannot simply be ignored. The room constant and the directivities of the loudspeaker system and the microphone should be checked in the 200 - 500 Hz range to make sure that there are not substantial deviations from the calculations made at 1 and 4 kHz. If the room has very little absorption below 1 kHz, and if the loudspeaker system becomes nondirectional in this region, it may be impossible to achieve satisfactory system gain without severely attenuating the mid-bass region. The result is the all too familiar system which provides satisfactory speech intelligibility, but which sounds like an amplified telephone.

The Indoor Gain Equation

From the foregoing discussions, we can appreciate the complexity of indoor system gain analysis and the need for accurately calculating the attenuation of sound along a given path, from either talker or loudspeaker, noting when we leave the direct field and make the transition into the reverberant field. If we were to attempt to establish a general system gain equation, we would have a very difficult task. However, in the special case where the microphone is in the talker's direct field, and both microphone and listener are in the loudspeaker's reverberant field, then the system gain equation simplifies considerably.

Let us consider such an indoor system, first with the system turned off, as shown in Figure 6-10. If the talker produces a level L at the microphone, then the level produced at the listener will be:

Level at listener = L - 20 log (D_{ct}/D_s), where D_{ct} is the critical distance of the talker. The assumption made here is that the level at the listener is entirely made up of the talker's reverberant field and that that level will be equal to the inverse square component at D_{ct}.

Now, the system is turned on, and the gain is advanced until the loudspeaker produces a level *L* at the microphone. At the same time, the loudspeaker will produce the same level *L* at the listener, since both microphone and listener are in the loudspeaker's reverberant field.

Subtracting the levels at the listener between the system on and the system off, we have:

$$\text{Difference} = L - \left[L - 20 \log (D_{ct}/D_s)\right]$$

or:

$$\text{Gain} = 20 \log D_{ct} - \log D_s$$

Finally, adding a 6 dB safety factor:

$$\text{Gain} = 20 \log D_{ct} - 20 \log D_s - 6$$

Note that there is only one variable, D_s, in this equation; D_{ct} is more or less fixed by the directivity of the talker and the acoustical properties of the room.

Of course there are many systems in which the microphone may be placed in the transition zone between the talker's direct and reverberant fields, or where the listener is located in the transition region between the loudspeaker's direct and reverberant fields. In these more complicated cases, the foregoing equation does not apply, and the designer must analyze the system, both on and off, pretty much as we went stepwise through the three examples at the start of this chapter.

Measuring Sound System Gain

Measuring the gain of a sound system in the field is usually done over a single band of frequencies. It is normally specified that system gain shall be measured over the octave-wide band centered at 1 kHz. Another common technique is to use pink noise which is then measured with the A-weighted scale. A typical specification for sound system gain might read as follows:

"The lectern microphone shall be used in its normal position. A small loudspeaker shall be mounted on a stand to simulate a person talking approximately .6 meters from the microphone. The response of this test loudspeaker shall be reasonably flat over the range from 250 - 4000 Hz.

"With the system turned off, the test loudspeaker shall be driven with a pink noise signal to produce a sound level of about 80 dB(A) at the system microphone. This level shall be measured with a precision sound level meter, using the "A" scale, with its microphone immediately adjacent to the sound system microphone.

"After noting the sound level at the system microphone with the sound system turned off, the sound system shall be turned on and its gain advanced to a point just below sustained oscillation. The amplified sound level shall be measured with the same sound level meter in the central part of the auditorium.

"The Δ of the sound system shall be calculated by subtracting the measured SPL at the microphone (system off) from the measured SPL in the auditorium (system on)."

The gain of the system is of course measured at some point in the auditorium and is the level difference at that point produced by the test loudspeaker before and after the system has been turned on. Details of the measurements are shown in Figure 6-11.

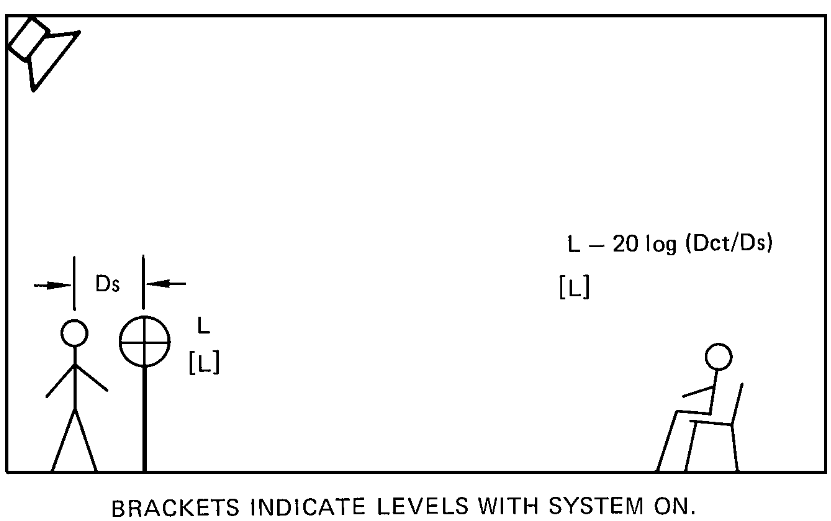


Figure 6-10. Conditions for the indoor system gain equation

General Requirements for Speech Intelligibility

The requirements for speech intelligibility are basically the same for unamplified as for amplified speech. The most important factors are:

1. Speech level versus ambient noise level. Every effort should be made to minimize noise due to air handling systems and outside interferences. In general, the noise level should be 25 dB or greater below the lowest speech levels which are expected. However, for quite high levels of reinforced speech, as may be encountered outdoors, a noise level 10 to 15 dB below speech levels may be tolerated.

2. Reverberation time. Speech syllables occur three or four times per second. For reverberation times of 1.5 seconds or less, the effect of reverberant overhang on the clarity of speech will be minimal.

3. Direct-to-reverberant ratio. For reverberation times in excess of 1.5 seconds, the clarity of speech is a function of both reverberation time and the ratio of direct-to-reverberant sound.

In an important paper (8), Peutz set forth a method of estimating speech intelligibility which has found considerable application in sound system design. The Peutz findings were compiled on the basis of data gathered over a period of years. The data and the method used to arrive at the published conclusion are clearly set forth in the paper itself. The conclusions can be summarized as follows:

1. In practice, the articulation loss of consonants can be used as a single indicator of intelligibility. Although the original research of Peutz was in Dutch speech, the findings seem to be equally applicable to English.

2. As would be expected, the researchers found wide variations in both talkers and listeners. However, a 15% articulation loss of consonants seems to be the maximum allowable for acceptable speech intelligibility. In other words, if articulation loss of consonants exceeds 15% for the majority of listeners, most of those people will find the intelligibility of speech to be unacceptable.

3. Articulation loss of consonants can be estimated for typical rooms. Articulation loss of consonants is a function of reverberation time and the direct-to-reverberant sound ratio.

4. As a listener moves farther from a talker (decreasing the direct-to-reverberant sound ratio) articulation loss of consonants increases. That is, intelligibility becomes less as the direct-to-reverberant ratio decreases. However, this relationship is maintained only to a certain distance, beyond which no further change takes place. The boundary corresponds to a direct-to-reverberant ratio of -10 dB.

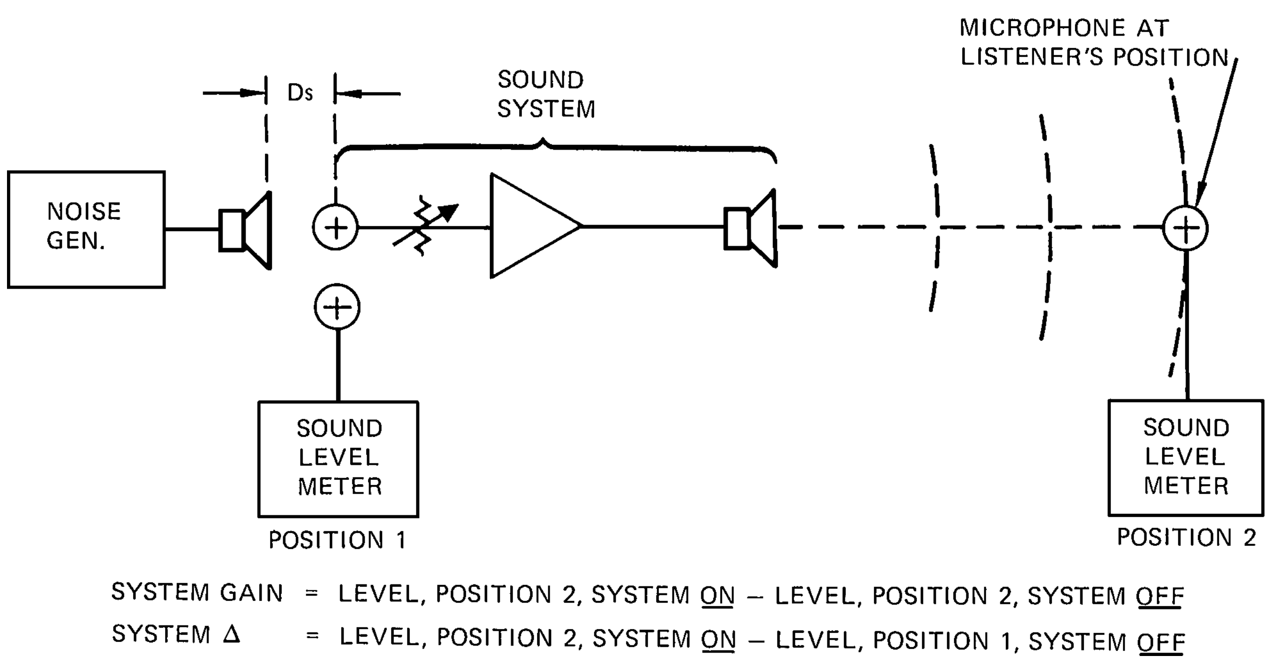


Figure 6-11. Measurement of sound system gain and delta (Δ)

The last point is illustrated graphically in Figure 6-12, adapted from the Peutz paper. Each of the diagonal lines corresponds to a particular reverberation time. Each shelves at a point corresponding to a direct-to-reverberant sound ratio of -10 dB. Note that the shelf may lie above or below the 15% figure depending upon the reverberation time of the room. This agrees with other published information on intelligibility. For example, Rettinger points out that in rooms having a reverberation time of 1.25 seconds or less, direct sound and early reflections always make up the greater portion of the total sound field. Intelligibility in such rooms is good regardless of the direct-to-reverberant sound ratio at any given listening position. Conversely, anyone who has worked in extremely large reverberant spaces such as swimming pools or gymnasiums knows that intelligibility deteriorates rapidly at any point much beyond the critical distance. According to the chart, a 15% articulation loss of consonants in a room having a reverberation time of 5 seconds corresponds to a direct-to-reverberant sound ratio of only - 5.5 dB.

Problems associated with speech intelligibility in enclosed spaces have received a great deal of attention prior to the publication of the Peutz paper. The virtue of Peutz' method for estimating speech intelligibility is its simplicity. It must be remembered, however, that a number of contributing factors are ignored in this one simple calculation. The chart assumes that satisfactory loudness can be achieved and that there is no problem with interference from ambient noise. It also postulates a single source of sound and a well behaved, diffuse reverberant sound field.

The data from the Peutz paper have been recharted in a form more convenient for the sound contractor in Figure 6-13. Here we have arbitrarily labeled the estimated intelligibility of a talker or a sound system as "satisfactory", "good", or "excellent", depending upon the calculated articulation loss of consonants.

There often is a dramatic difference in the acoustical properties of a room depending upon the size of the audience. Calculations should be made on the basis of the "worst case" condition. In some highly reverberant churches particularly, it may turn out that there is no practical way to achieve good intelligibility through the entire seating area when the church is almost empty. The solution may involve acoustical treatment to lessen the difference between a full and an empty church, or it may involve a fairly sophisticated sound system design in which reinforced sound is delivered only to the forward pews when the congregation is small (presuming that a small congregation can be coaxed into the forward pews).

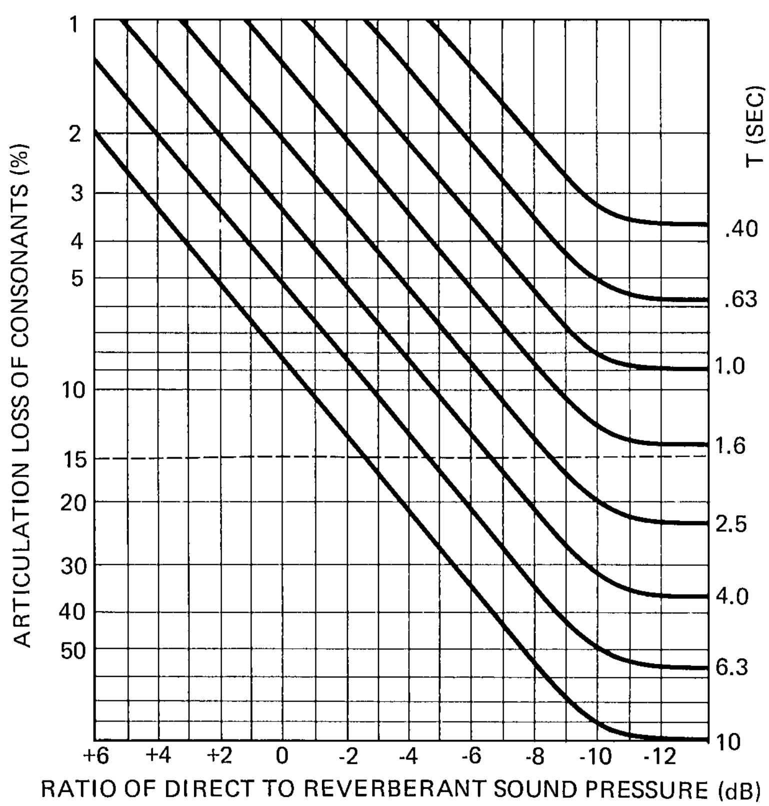


Figure 6-12. Probable articulation loss of consonants vs. reverberation time & direct-to-reverberant sound ratio