- 8 -
Room Acoustics

The Small Room

As much as we would like to cover this subject in its entirety, alas we cannot. The subject is far too vast to cover completely in a text on transducers. The unfortunate thing is that from a subjective point of view, one of the most important aspects of the listening experience is the room. Technically, the loudspeaker design can be optimized to fit the room design, but seldom, if ever, do we optimize the room for the loudspeaker design. This is simply because the room usually already exits and even if it doesn't there are fundamental characteristics about rooms which simply cannot be alleviated; like room modes and wall reflections.

There are some important concepts in room acoustics that must be understood in order to understand what we are trying to achieve with a loudspeaker design. We will cover these fundamentals briefly with our focus mainly on small room acoustics since that is the venue where most loudspeakers are used. We do not intend to discount the importance of sound reinforcement in large spaces, which is indeed an important and interesting subject, but these spaces are well covered in numerous other texts on room acoustics. The small room, on the other hand, has hardly been examined anywhere. There are significant differences in small room acoustics when compared to larger rooms and we will highlight some of these, as well as some similarities.

8.1 The Rectangular Room[1]

The room response for a rectangular room is a simple extension of the solutions to the Wave Equation that we have already seen. The three spatial dimensions will have three independent solutions: $X(x)$, $Y(y)$ and $Z(z)$. The separation constants, which turn out to be the room's modes, are summed together, as required by the separation of variables technique

$$k^2 = k_x^2 + k_y^2 + k_z^2 \tag{8.1.1}$$

and the eigenfunctions are

1. See Kuttruff, *Room Acoustics*, Morse, *Vibration and Sound,* or Morse and Ingard, T*heoretical Acoustics.*

$$
\begin{aligned}
X(x) &= A e^{-ik_x x} + B e^{ik_x x} \\
Y(y) &= C e^{-ik_y y} + D e^{ik_y y} \\
Z(z) &= E e^{-ik_z z} + F e^{ik_z z}
\end{aligned}
\tag{8.1.2}
$$

There are a number of boundary conditions that we could apply, such as a rigid wall, a wall with an impedance on it, a driven wall, etc. These conditions could be in any perturbation of the three coordinates. Clearly, this is far too many examples to study here. We will start by looking at a perfectly rigid room, with a small amount of damping uniformly spread around the room.

By applying the boundary conditions

$$
\begin{aligned}
\frac{\partial X(x)}{\partial x} &= 0 \quad at \quad x = 0, L_x \\
\frac{\partial Y(y)}{\partial y} &= 0 \quad at \quad y = 0, L_y \\
\frac{\partial Z(z)}{\partial z} &= 0 \quad at \quad z = 0, L_z
\end{aligned}
\tag{8.1.3}
$$

Eq. (8.1.2) become

$$
\begin{aligned}
X(x) &= A\cos\left(k_x x\right) \\
Y(y) &= B\cos\left(k_y y\right) \\
Z(z) &= C\cos\left(k_z z\right)
\end{aligned}
\tag{8.1.4}
$$

and the complete solution for the pressure

$$
p(x,y,z) = ik\rho c \sum_{l,m,n} A_{lmn} \cos\left(k_x x\right)\cos\left(k_y y\right)\cos\left(k_z z\right) \tag{8.1.5}
$$

$$
k_x = \frac{l\pi}{L_x}, \quad k_y = \frac{m\pi}{L_y}, \quad k_z = \frac{n\pi}{L_z}
$$

$$
and \quad k_{lmn} = \sqrt{\left(\frac{l\pi}{L_x}\right)^2 + \left(\frac{m\pi}{L_y}\right)^2 + \left(\frac{n\pi}{L_z}\right)^2} + i\delta
$$

We will account for damping by letting the wavenumber k be complex, where δ is the damping constant. This approach is accurate so long as the damping is small and uniformly distributed. We will see a more accurate formulation in later sections.

We are interested in the response a point in the room x_p due to a point source at point x_0. The result will be the Green's Function for the room and from it we can calculate the response at any point due to a source placed anywhere in the room. In order to get the Greens function we need to solve the equation

$$
\Delta^2 p(x,y,z) - k^2 p(x,y,z) = \delta(x - x_0)\delta(y - y_0)\delta(z - z_0) \tag{8.1.6}
$$

Substituting Eq. (8.1.5) in Eq. (8.1.6) results in

$$\sum_{l,m,n} A_{l,m,n} \cos(k_x x)\cos(k_y y)\cos(k_z z)(k_{lmn}^2 - k^2) = \delta(x - x_0)\,\delta(y - y_0)\,\delta(z - z_0) \tag{8.1.7}$$

Multiplying both sides of this equation by the eigenfunctions (denoted k') and integrating over all three coordinates yields

$$\begin{aligned} &\iiint_V \sum_{l,m,n} A_{l,m,n} \cos(k_x x)\cos(k_y y)\cos(k_z z)\cdot \\ &\qquad \cos(k'_x x)\cos(k'_y y)\cos(k'_z z)(k_{lmn}^2 - k^2)\,dV \\ &= \iiint_V \delta(x - x_0)\delta(y - y_0)\delta(z - z_0)\cos(k'_x x)\cos(k'_y y)\cos(k'_z z)\,dV \end{aligned} \tag{8.1.8}$$

Using the fact that the modes are orthogonal and the properties of the delta function we can obtain a formula for the coefficients A_{lmn}

$$A_{lmn} = \frac{\cos(l\pi\, x_o)\cos(m\pi\, y_o)\cos(n\pi\, z_o)}{\Lambda_{lmn}(k_{lmn}^2 - k^2)} \tag{8.1.9}$$

$$\Lambda_{lmn} = \iiint_V \cos^2(k_x x)\cos^2(k_y y)\cos^2(k_z z)\,dV$$

Finally the Green's Function for a rectangular room

$$p(x, y, z) = i\,k\rho c \sum_{l,m,n} \frac{\cos(k_x x)\cos(k_y y)\cos(k_z z)\cos(k_x x_0)\cos(k_y y_0)\cos(k_z z_0)}{\Lambda_{lmn}(k_{lmn}^2 - k^2)} \tag{8.1.10}$$

A typical room response calculation using this equation is shown in Fig. 8-1. This calculation used 27,000 modes plotted along 400 log spaced frequency points, a very high resolution plot. The room is typical of a fairly large sized room that might be found in a home – i.e. a living room. It is three meters tall by six meters long and four meters wide and has a small amount of damping. The source is in one corner and the receiver is just under two meters from the source straight out into the room.

The falloff at about 2 kHz is not real, but it is deliberate. It is there to show where the modal calculation no longer has modes, i.e. above the resonance frequency of mode with the highest mode number used in the calculation. The response falls to zero above this frequency.

The first few modes are clearly seen at about 29 Hz, 40 Hz, 50 Hz, etc., but it is certainly obvious that there are not 27,000 peaks in this response. This means that, above about 100 Hz, we cannot say for certain if individual peaks in the response are actually modes or not. With this many modes it is inevitable that some mode(s) will exists at each and every peak; there could actually be many.

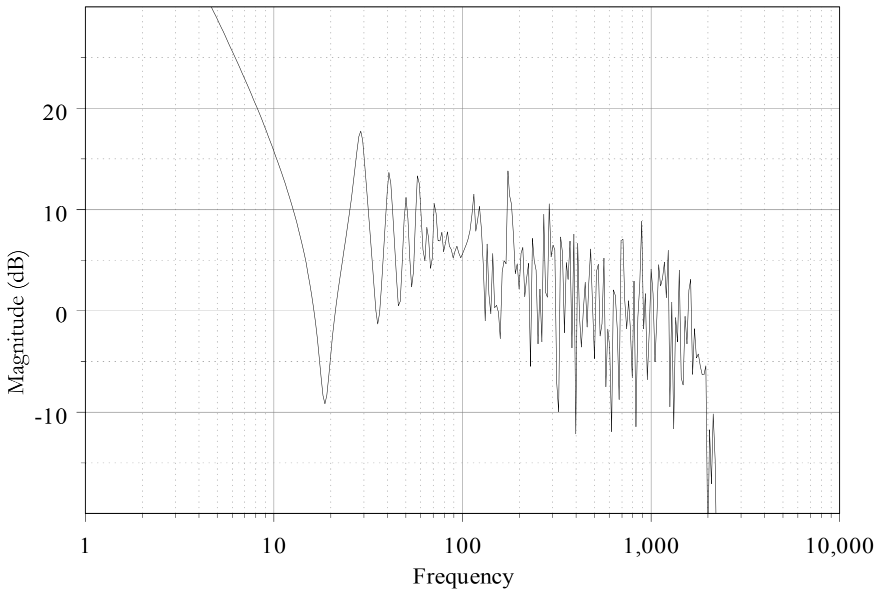

Figure 8-1 - Typical room response for a closed room with rigid walls

The corollary to this is that each mode does not necessarily have a corresponding peak.

Looking at Fig. 8-1 we can make the following observations.

- Above some frequency peaks in the response are no longer due to modes.
- The response actually rises at lower frequencies below the first non-zero mode (and the first anti-resonance). This is due to the 0,0,0 mode, i.e. a static pressure mode. Its level depends on the static pressure release resistance and only goes to infinity for a completely sealed (air tight) enclosure. Real enclosures, most notably cars, can have significant low frequency gain as a result of this static mode.[2]
- Above about 1.5 kHz, we no longer have enough modes in the calculation to get accurate results and hence the response falls off, indicating that it is the modes that carry the energy.
- The response flattens out above some frequency (about 100 - 200 Hz in this curve) and has a fairly constant response ripple amplitude. This is in stark contrast to the lower frequencies where the peak-to-trough (resonance to anti-resonance) ratio decreases with frequency and then widens again as the modal distribution gets larger.

2. See Blind and Geddes "The ESP Sound System", *JAES*

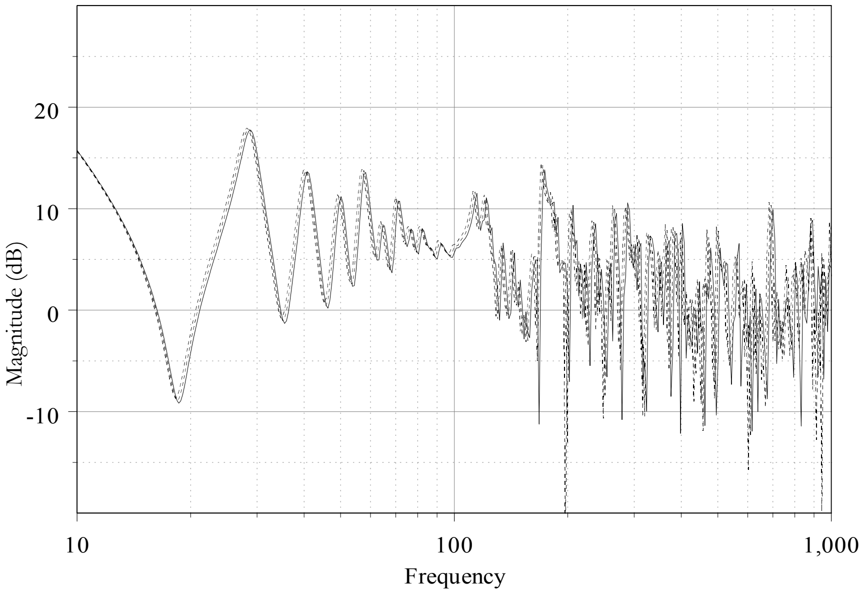

Figure 8-2 - A room calculation with a very small perturbation of the wave speed

Consider now another computer experiment, the results of which are shown in Fig. 8-2. In this figure we have calculated the response for the same room as above, but now we have varied the sound velocity by ±1%. A 1% change in the speed of sound can occur with less than a 1° change in room temperature or a correspondingly minor change in atmospheric pressure. It is a virtual certainty that the sound speed will change by this amount in any room over a period of a few minutes or quite possibly a substantially less period of time. A small air current could even change the relative local sound speeds in different parts of the room by this amount. At low frequencies, this speed variation has little effect, but at the higher frequencies it appears to make a great deal of difference. By enlarging a section of the plot from 700 Hz to 750 Hz and plotting the calculations with a higher resolution, we will get the curves shown in Fig. 8-3.

This is an important result, for it indicates that the frequency response of a room is not at all a stable quantity. Similar to the concepts in chaos theory, the summation over an extremely large number of modes, which exist in rooms at higher frequencies, creates a hypersensitivity to even minimal changes in the parameters. Very large changes in the results stem from very small changes in the underlying properties. There is little to no recognizable similarity in the three curves shown in Fig. 8-3 although we might naïvely have expected them to be identical.

The interesting thing is that it turns out that we will get this same result for a small displacement of the source or receiver, in any frequency band (above some frequency, which we will define shortly), for any source or receiver locations and

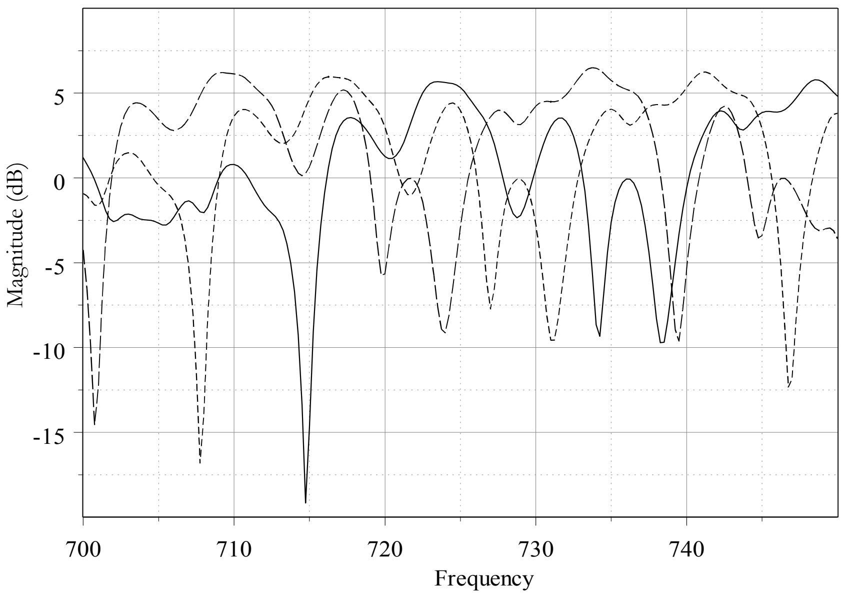

Figure 8-3 - the same calculation as in Fig. 8-2 but expanded to show the detailed frequency response

most importantly, for any room, of any size, shape or reverberation time. This is a remarkable situation.

The only way that we can even begin to discuss the frequency response of rooms is to consider this response to be a random variable. We can then consider its statistical properties, etc. but we cannot consider it to be a deterministic quantity, i.e. a stable curve. We must never talk about any single measurement of sound in a room with any level of assurance as to its validity.

Along these lines it would be good to know how the pressure response (in dB) levels are distributed, the probability distribution function. Fig. 8-4 shows the probability distribution, actual (bar) and as a fitted function (line), for the pressure response calculations in the previous figures. This distribution is a classical case of a Poisson distribution. (It is important to remember that this distribution is in dB values of the response, not linear ones.) An identical result was shown by Schroeder in 1954 using an entirely different technique than that used here.[3] This distribution shows that the response is almost Gaussian about the zero dB line with a standard deviation of about ±5 dB, Schroeder found 5.5 dB theoretically. The mean here is not zero dB only because the absolute scale in the calculations was arbitrarily chosen. If it were normalized to the RMS pressure throughout the room then the mean would be zero dB.

3. See Schroeder, “On the Statistical Response of Sound in Rooms”, *JAES*

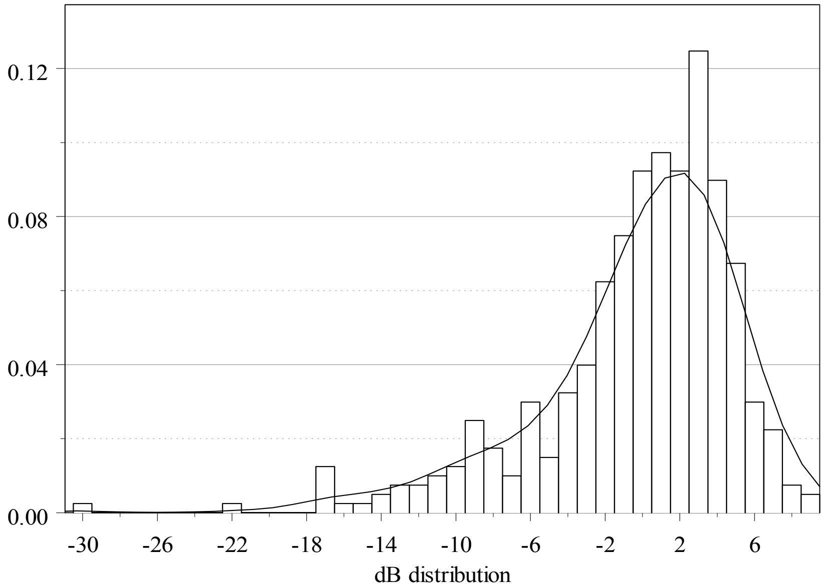

Figure 8-4 - The probability distribution in dB level for the pressure response in a room

These results imply that a measurement of a tone or tones in a room will have an expected error of 5.5 dB – a virtually useless measurement. As with any statistical quantity, this variance can be reduced by using a greater number of degrees of freedom in the measurement. We can achieve this by either frequency averaging, spatial averaging or typically both. Frequency averaging alone requires a fairly wide bandwidth, about an octave to get the expected error down to about 1 dB. By performing a spatial averaging we can get a much narrower bandwidth measure to the same expected error of 1 dB. Understanding the implications of the statistical nature of the sound field in rooms is paramount to the understanding of room acoustics.

As we stated earlier, the statistical nature of the sound field in a room is valid only above a certain frequency. Kuttruff[4] states that frequency is

$$f_s \approx \frac{5000}{\sqrt{V\delta_0}} \quad Hz \tag{8.1.11}$$

f_s = *the Schroeder frequency*

α = *the damping constant in* s^{-1}

V = *the room volume in* m^3

The Schroeder frequency is the lower limit of the applicability of the statistical assumption. Eq. (8.1.11) has been verified experimentally. In the room whose

4. See Kuttruff, *Room Acoustics*

modeled response we have shown in this chapter, the statistical region appears to start somewhere between 100-200 Hz. Applying Eq. (8.1.11) to this room (using a damping constant of about $10\,s^{-1}$ and a volume of $76\,m^3$) we would expect f_s to be at about 181 Hz. We can see that this equation is reasonable even for small rooms.

For comparison purposes

$$T = \frac{6.91}{\delta_0} \tag{8.1.12}$$

T = reverberation time in s

δ_0 = the mean damping coefficient in s^{-1}

Fig. 8-5 shows a graph of f_s versus room volume at several values of the reverberation time. Residential rooms hover around the $100\,m^3$ point, commercial

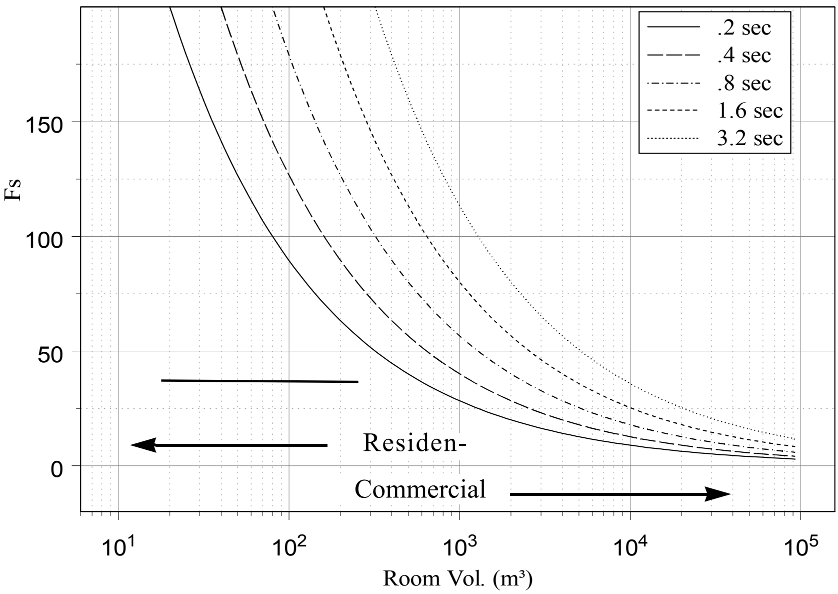

Figure 8-5 - Schroeder frequency versus room volume and reverberation time

rooms the $1000\,m^3$ point and large theatres etc. around $10{,}000\,m^3$. Clearly, the geometrical acoustics assumption is valid at any audible frequency in any theater and it is also valid for most commercial rooms, except perhaps smaller reverberant ones, but it is certainly not valid in residential rooms at low frequencies.

It is interesting to note that the assumptions of geometrical acoustics, where sound rays reflect off of walls, obeying optical laws of reflection etc., are only valid and can only be applied above f_s. Below this frequency, one can no longer think of sound in a space as traveling in a ray-like fashion and only modal approaches to analysis can be used. This result has significant implications to concepts such as directivity, damping placement, etc. with which we have become so familiar.

In the geometric acoustics region, the sound field is well adapted to analysis with computer programs using concepts such as specular reflections, phase gratings, etc. which are all well defined and applicable. Owing to the fact that there is so much literature in the field, which utilizes geometrical acoustics, we will not cover this subject here. We will also discuss some aspects of early reflections as found in small rooms which are, of course, above f_s. What we will look at in some detail is the region below f_s and the vicinity of the transition point, where very little literature is available on room acoustics in this region.

8.2 A Wall With an Impedance

One important aspect of the modal problem is the way finite wall impedances affects the modes. We will now show how the fundamental equations of the previous section are modified to account for these different boundary conditions. This will be done only for a single dimension, the rationale being that the analysis and conclusions are identical for the other two dimensions.

The new boundary condition is one where we specify the relationship between the pressure and the velocity – the pressure gradient. The new conditions become

$$\beta \left.\frac{\partial p(x)}{\partial x}\right|_{x=L_x} = ikp(x)\big|_{x=L_x} \tag{8.2.13}$$

β = the specific acoustic impedance at the $x = L_x$ wall

This condition must be applied at each boundary where there is an impedance value other than infinite. The basic solution is still

$$p(x) = Ae^{-ik_x x} + Be^{ik_x x} \tag{8.2.14}$$

Inserting this equation into the two boundary conditions (the other, for $x=0$, is the same as in the previous section) yields

$$\begin{aligned} &A - B = 0 \\ &-i(\beta k_x + k)e^{-ik_x L_x} A + i(\beta k_x - k)e^{ik_x L_x} B = 0 \end{aligned} \tag{8.2.15}$$

Which after some algebraic manipulation leads to

$$e^{i2k_x L_x} = \frac{\beta k_x + k}{\beta k_x - k} \tag{8.2.16}$$

This is a transcendental equation in k_x which can be solved (albeit with some difficulty) for the eigenvalues. We must remember that both k_x and β are, in general, complex quantities.

Taking the natural log of both sides of Eq. (8.2.16) we get

$$i2k_xL_x = \ln\left(\frac{\beta k_x + k}{\beta k_x - k}\right)$$
$$k_x = \frac{\ln\left(\frac{\beta k_x + k}{\beta k_x - k}\right)}{i2L_x} \tag{8.2.17}$$

which is still a transcendental equation and we seem to be no better off. If, however, we now consider the fact that we are primarily interested in approximate values of k_x in the vicinity of each of the resonance frequencies then $k \approx k_x$ and the above equation becomes much simpler and more readily solvable. The resulting form is

$$k_x \approx \frac{l\pi}{L_x} + i\frac{\ln\left(\frac{\beta-1}{\beta+1}\right)}{2L_x}$$
$$\approx \frac{l\pi}{L_x} + i\frac{\ln\left(\frac{\sqrt{(\varepsilon^2+1-2\delta+\delta^2)(\varepsilon^2+1+2\delta+\delta^2)}}{(\delta-1)^2+\varepsilon^2}\right)}{2L_x} + \frac{\tan^{-1}\left(\frac{-2\varepsilon}{(\delta+1)(\delta-1)+\varepsilon^2}\right)}{2L_x} \tag{8.2.18}$$

$\beta = \delta + i\varepsilon =$ the complex wall specific acoustic impedance

We can see that for large values of β, we will get eigenvalues equal to the original values $k_x = l\pi / L_x$

Lets examine what happens when we have a wall with only a reactive part, i.e. $\delta = 0$, then Eq. (8.2.18) becomes

$$k_x \approx \frac{l\pi}{L_x} + \frac{\ln\left(\frac{\sqrt{(\varepsilon^2+1)(\varepsilon^2+1)}}{1+\varepsilon^2}\right)}{i2L_x} + \frac{\tan^{-1}\left(\frac{2\varepsilon}{1-\varepsilon^2}\right)}{2L_x} = \frac{l\pi + \frac{1}{2}\tan^{-1}\left(\frac{-2\varepsilon}{\varepsilon^2-1}\right)}{L_x} \tag{8.2.19}$$

For negative ε (compliance-like) the eigenvalues are raised and for positive values (mass-like), the eigenvalues are lowered (ε must be greater than one for validity of these results).

For a purely dissipative wall $\varepsilon = 0$ and we have

$$k_x \approx \frac{l\pi}{L_x} + i\frac{\ln\left(\frac{(\delta+1)}{(\delta-1)}\right)}{2L_x} \approx \frac{l\pi}{L_x} + i\frac{2}{\delta L_x} \tag{8.2.20}$$

A value of $\delta > 1.0$ will result in a decay of energy over time, but no change in the frequency of the mode.

When we compare this result to the results shown in Kuttruff[5] for an impedance on both walls we see that we have exactly the same results. This is, of course, good (that we agree), but we should remember that the two problems are slightly different. Here we had only one wall with an impedance and Kuttruff assumed two. The equivalence of the results implies that, in the modal region, it makes little difference where the impedance is placed. It will act as a combined impedance independent of its placement. This is good to know! It means that if we want low frequency damping in a room to smooth out the modes then we only need to do this on one wall of each opposing pair, a real savings in construction cost. Of course in the geometrical acoustical region it makes a lot of difference on which wall the damping is placed!

The mode functions (Eigenfunctions) can be determined from Eq. (8.2.15) which yields the result that $A = B$ and so the functions are simply

$$p(x) = A\cos(k_x x) \tag{8.2.21}$$

where the complex value of k_x carries the information about the wall impedance. These functions are shown in Fig. 8-6 for the cases of a rigid wall, a damped wall, a wall with a mass like impedance and a wall with a compliance like impedance. The standing wave ratio is constant for all but the dissipative case, which decreases near the absorbing wall for the absorbing case. The wall at $x = 0$ is rigid, so no changes take place there

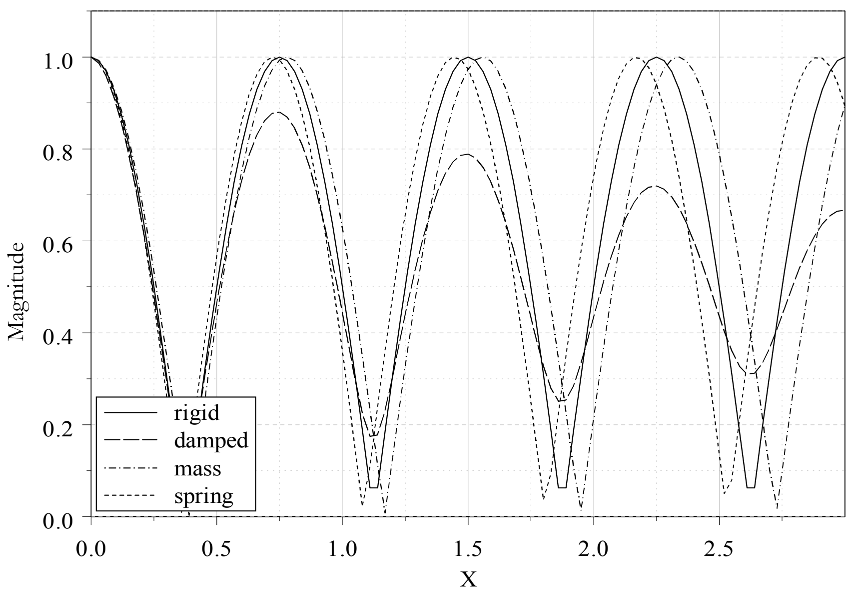

Figure 8-6 - modal functions for mode four with various wall impedances

5. See KuttRuff, *Room Acoustics*

8.3 Fundamental Concepts in Small Room Acoustics

We should first mention a factor that we have ignored up to this point, regarding sources in rooms, which is important to understand. This factor concerns the characteristics of the *direct* and *reverberant* sound fields. The direct field is often confused with the nearfield, a concept that we have already talked about.

Consider the following experiment: place a source somewhere in a room emitting a continuous signal and measure this sources output level as we move away from it along a straight line. Let us consider this to be an ideal source in that it does not have a significant near-field component to worry about. We would see that as we moved away from this source the sound level would fall off at the rate of 6dB per doubling of distance, that is, until we reached the point at which the level of the reverberation field, the energy continuously bouncing around the room, equals that of the sound directly emitted from the source. Beyond this point, the sound level will be stationary, at least as stationary as the random nature of the sound field allows. The *direct* field is that portion of the sound field which surrounds the source and has a level greater than the reverberation level. The rest of the room is in the *reverberant* field where the sound is usually considered to be diffuse or random in nature. This brings us to a fundamental consideration when talking about sound sources in rooms – the random nature of the sound field applies only to the *reverberant* field, not the *direct* field.

The direct field depends on the reverberant characteristics of the room. As the reverberation level goes up, the direct field recedes and vice versa. Fig. 8-7 shows this effect for four rooms with increasing amounts of reverberation. The direct field is the solid line (the field in an anechoic chamber) and the reverberant field is where the lines become horizontal. Note that no clear cut transition is apparent, much like cutoff in a filters response. For higher reverberation the sound field is never truly direct.

We should make another important point regarding the direct and reverberation fields and that is that the first sound arrival from any source is direct, in that there has not been sufficient time for the reverberation to build up. Subjectively, this is a crucial point because it means that if there is sufficient time before the reverberation field has built up, then the ear will hear the direct sound free from reflection effects due to the room (nearby diffraction effects, cabinet edges etc. will still occur) and the reverberant sound will be a subjective event separate from the direct sound. Note that the direct sound will have the same frequency response as the of the loudspeaker system, which highlights the importance of the anechoic response. But, when talking about subjective impression, we must also consider the reverberation field. The importance of this field depends on several factors, which can be quite complicated. The first is the frequency response of the reverberant field. This response depends on the "power" response of the loudspeaker, the integrated polar response over all angles, as well as the room's acoustic response. For this reason, the ideal system would have a flat axial response plus a flat power response. The frequency response of the reverberation field is a combination of two responses: the loudspeakers power response and the rooms frequency response. We should always consider the frequency response of the sound absorbing material placed throughout the room as this affects the frequency response of the rooms reverberant field.

The importance of the power response further depends on where the listener is relative to the direct and reverberant fields. In the reverberant field the power response is very important because the reverberant response depends so heavily on the loudspeakers power response, but also the room absorption, as we said. If the listener is in the direct field of the source then the power response is not as important since it is a small factor in the total sound that arrives at the listener's ear.

It would appear that we want is to always be in the direct field, since then, only the axial response is important. We would not need to worry about the polar response, the room response, etc. This situation is easily achieved in practice by making the listening space as nearly anechoic as possible, and/or by sitting close to the loudspeakers. The problem with this approach is that there is then no natural acoustic contributed by the room itself. For good "spaciousness", (a term meaning the subjective feeling of space in the sound) it has been shown that the reverberant sound arrival should come from many directions, but principally in the horizontal plane. The characteristics of the ear make the reasons for this obvious. Being in the horizontal plane our ears are most acute to sound localization in this plane, which again is logical since the world we live in is principally in the horizontal plane.

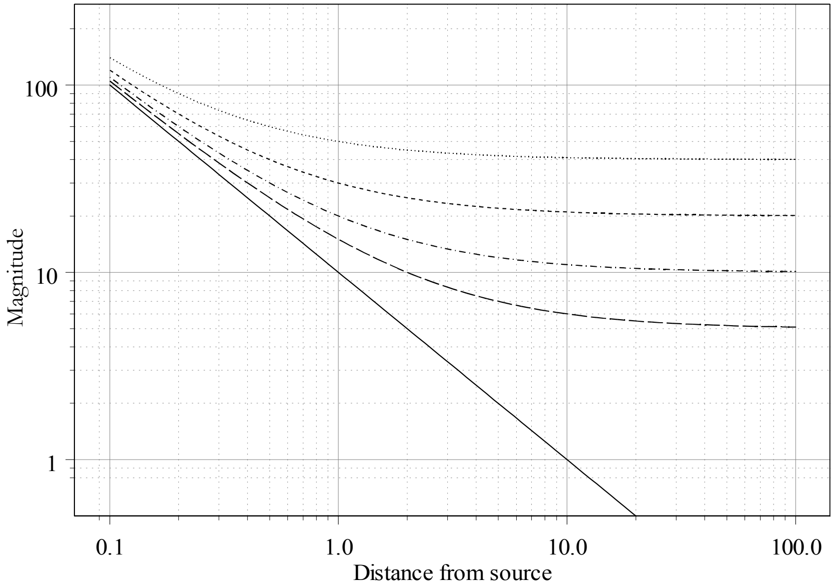

Figure 8-7 - Falloff of sound level in a reverberant field for several values of reverberation

In a dead room, there cannot be any perception of spaciousness from the room and the source cannot provide sound arrival from multiple directions. (There are technologies which can trick the ear into perceiving this effect, as well as multi-channel technologies which can create multiple arrivals from multiple directions, but they are not widely available. These techniques lie more in the domain of signal processing than loudspeaker or room design.) In some schools of thought, a dead room is ideal because then the only acoustics are in the recording itself. The listening room does not add any acoustic of its own. We do not subscribe to this belief.

In order to have a feeling of being in a large room with good acoustics, i.e. spaciousness, one must have a perceptible amount of reverberation, which can be difficult to achieve in a small room. Our experience has shown that with proper attention to the room design we can have a small room with a substantial reverberant field component and the associated feeling of spaciousness. In this approach, however, the polar/power response of the system becomes of paramount importance.

We should also consider the directivity of the system from another perspective. If the source is omni-directional, then the reverberation field is established much faster than it would be for a system with a narrow directivity. In other words, the directivity has a pronounced effect on the time delay at which the reverberation field over takes the direct field. This delay also exists in large rooms where the direct sound arrives well before the build up of the reverberant field and it is usually quite large. An omni-directional source cannot achieve any significant amount of delay in the build-up of the reverberant field. It happens almost instantaneously. A properly designed narrow directivity system can have a substantial delay in the reverberation buildup. An omni-directional sources almost instantaneous effect on the reverberant field and its predominance in the marketplace is, perhaps, why there is a significant contingent of practitioners who prefer dead rooms.

Some paragraphs above we mentioned that “if there is sufficient time before the reverberation field has built up” without really quantifying what we meant. In Chap. 13 we will learn that the ear integrates sound over about a time window (see Sec.13.3 on page 284). If there is reverberant sound (reflections) after the arrival of the direct sound but within the time period of integration then the ear will integrate the two sounds, creating the comb filtering effect, which results from the summation of two signals time delayed from one another. Early reflections change the perception of the timbre and localization of the sound presentation. Reflections in larger rooms almost always occur outside of the ears' integration time and have a completely different subjective effect than they do in small rooms.

We can see that if there are no reflections and hence no reverberation buildup in the first 20 ms (or there about), then the direct sound from the loudspeaker will be heard without coloration from the room. Remember that in a small room the reverberation build up can be extremely fast and once the first reflections begin to arrive, they are quickly followed by others and the reverberant field has been established. A 20 ms delay is difficult to achieve in a small room. Delays of this magnitude amount to about seven meters of additional travel for the reflected wave, which in a room of typical dimensions is virtually impossible to achieve. It is extremely important to minimize these early reflections without the use of significant amounts of sound absorbing material because a reduction in the reverberation field will reduce the spaciousness quality. Once again, source directivity can be a critical advantage in dealing with this problem. A narrow sound beam can be aimed to prevent the sound from hitting nearby walls until well after it has arrived at the listener, once again pointing us in the direction of using narrow source directivity for small rooms.

Consider the schematic drawing shown in Fig. 8-9. Two identical 90º coverage

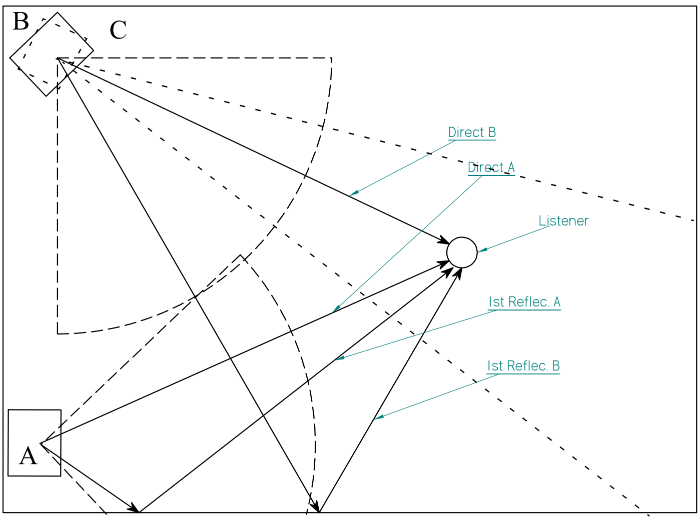

Figure 8-8 - Room layout showing coverage of directive sources in two different configurations

loudspeakers are placed in or near the front corners of the room. However, they point in different directions. Speaker A has a very small delay between the direct sound and the first reflection while for speaker B the delay is much larger. In addition, the direct sound and delayed sound in setup A both arrive at, principally, the same ear, while for B they arrive at opposite ears. The timbre shift will be significantly lower in setup B because there will be less interference between the direct and reflecting waves. Further, the ears' ability to minimize the negative effects of two sound waves tends to work better when the sound is binaural rather than monaural. A small amount of sound absorption placed at the location of the early reflections will further reduce the degradation due to these reflections.

There are, of course, even earlier reflections from the floor and ceiling, but these can also be minimized by having a narrow vertical directivity. This also has the added advantage of creating the maximum amount of sound energy is the horizontal plane where we want it.

It should be obvious that if we had the narrow directivity loudspeaker labeled C that the early reflections would be minimized and the onset of the reverberation would be significantly delayed. If this room were live, then there would be a significant amount of lateral reflections delayed in time from the direct sound. While the directivity of C is perhaps a little extreme, we clearly prefer it over an omni-directional source in a live room.

This has been a simplified analysis of some of the problems of small room acoustics, but it does point out one thing very clearly. What we want in source directivity is not the omni-directional response that we get from the extremely common small loudspeaker systems that dominate today's marketplace. We want a narrow directivity such as can only be achieved with waveguides and/or arrays of transducers. This is one of the primary rationales behind our detailed analysis of directivity control in this text. Controlled source directivity is essential to good sound in almost any venue, but it can be critical in a small room. We feel that this area of loudspeaker design has been grossly overlooked in the vast body of literature on loudspeaker design.

8.4 Source Placement[6]

We have seen that small rooms can have significant problems with modes at low frequencies (see Fig. 8-1). Most rooms will not enjoy the significant boost that comes from the zero order mode as shown in the figures since most rooms have doors and ventilation systems which negate this effect or, at least, substantially reduce it. By now, the reader should also have been convinced that the steady state room response, above f_s, must be treated as a random variable (even though it appears to be deterministic). Further, we should now realize that our principal concern is with the mean response in rooms, since we must expect any single response to have errors, relative to this mean, of about 6 dB. From our studies thus far it should be clear that the low frequency modes will generate, on the average, an even larger frequency response variation than what we would expect to occur at the higher frequencies. The question as to the proper type and location for a low frequency source is natural.

A concept that we hear a lot about in small room acoustics is that of modal distribution. One of the authors did his dissertation on this topic.[7] Using FEA, the sound field statistics below f_s were investigated for several different room configurations, source locations and damping levels. A major result of this study was

6. See Geddes, "Sources in Real Rooms", *JAES*

7. See Geddes, *An Analysis of the Low Frequency Sound Field in Non-rectangular Enclosures …*

that the modal distribution was not a significant factor once some damping was achieved at these frequencies. The modal overlap and the damping distribution were found to be the important parameters. The modal overlap is basically controlled by the amount of damping that is present. The greater the damping, the greater the modal overlap. The net result being that given a sufficient amount of uniform damping (i.e. not modal dependent), all rooms have basically the same low frequency statistics regardless of their shape.

The second important consideration to come out of this work was the importance of the distribution of the damping in the room. Non-rectangular rooms do have a clear advantage here since a room with one wall slanted (angled) by at least 15º in two dimensions (i.e. the normal does not lie along any coordinate direction) will "mix-up" the modes so that <u>all</u> sound waves will hit <u>all</u> walls. This inherently causes a uniform distribution of the damping no matter where the damping takes placed. The effect that the slanted wall had on the modes was of secondary importance, although, admittedly, there was a modest improvement in the statistics for lightly damped rooms (i.e. reverberation chambers). No residential room would ever fit into this later category, however.

From what we found in section 8.2, we should realize that a rectangular room with a reasonable amount of low frequency damping on at least one of each opposing wall pairs will have a low frequency sound field which is "as good as it gets".

There has also been much discussion about the proper location and/or source type for the low frequency source so as to smooth out the response irregularities that we have noted. Some have argued that a dipole is a better low frequency source than a monopole because it excites fewer modes. We should emphasize, however, that a dipole – a pressure gradient (or velocity) source – does not necessarily excite fewer modes; it simply excites different modes. This is because a dipole excites pressure nodes (velocity anti-nodes) while monopoles excite pressure anti-nodes (velocity nodes). In general, there is no reason to expect a different numbers of modes to be excited by a dipole than a monopole. We should also realize by now that exciting fewer modes also does not necessarily reduce the response variations, in fact it may very well increase them. It is not the *room modes* that cause room response problems, but the *lack of modes*, typical of low frequencies in small rooms, which cause problems. This is a widely misunderstood concept. Perhaps the Greeks understood this when they placed multiple large urns (Helmholtz resonators) around the dead outdoor spaces that they used for theatre. We might want to consider doing something along these same lines in our small rooms.

Fig. 8-9 shows a typical response curve generated by placing a monopole, (top plot) and a dipole (bottom) in a two dimensional room. We have only used two dimensions because the third dimension does not add anything to the study (we ignored the floor to ceiling dimension). The room in this figure was five meters wide by six meters long and the source was one meter out from a corner. The room has a medium amount of damping. The response curves at several random

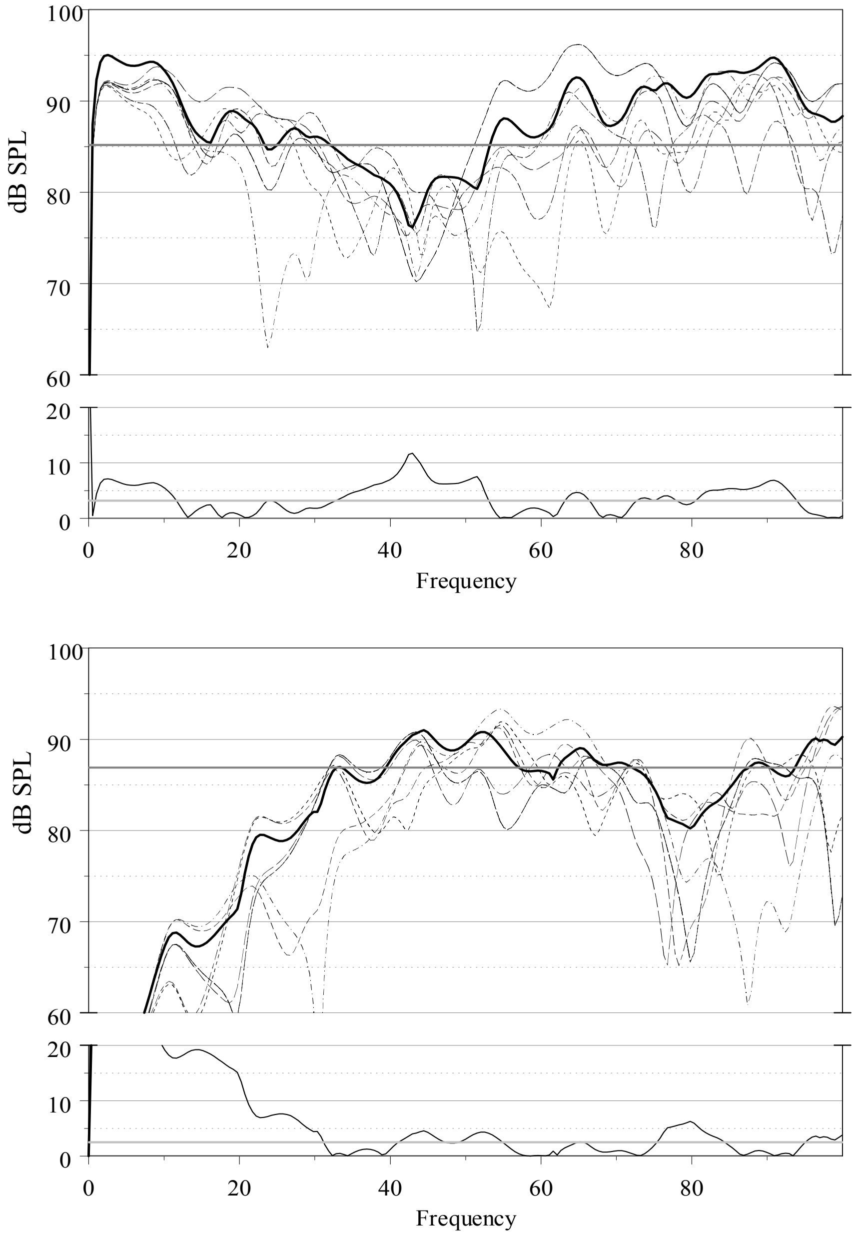

Figure 8-9 - Room responses for a monopole and a dipole locations in the other half of the room (away from the source) are also shown in these figures. The average frequency response curve (what we would get from a spatial average of the individual frequency response curves) is shown as the solid line. The mean level over all receiver positions and frequencies is shown as the horizontal line. The solid line at the bottom of the plot is the deviation of the average frequency response curve from the mean level and the horizontal line is the frequency average of the deviations. We will call the curves at the bottom of the plots the *roughness* of the response.

The first thing that we notice about these two plots is the lack of low frequency response for the dipole. This, of course, is to be expected as one goes lower in frequency since the dipole cannot generate a static pressure, while a monopole can. So at the lowest frequencies the monopole will always have the greater response. The extreme low frequency falloff of the dipole is ignored in the data calculations to follow. This is because no real source could drive a room at these frequencies anyway. In the plots we have shown the response as if the source could go down to DC, but in the data calculations we have excluded the responses below 20 Hz.

Table 1 shows the results for three sources: a monopole, and two dipoles with different dipole moments (the dipole moment is the distance between the positive and negative sources which make up the dipole). Several source locations were investigated. The location parameter is the distance out from the corner to the source. For any single source location, the error is probably 1dB, while the averages would have an expected error of about one-third of that.

Source location	Monopole	Dipole (moment)		Avg.
		.1m	.2m	
Mean Sound Pressure Level (dB SPL)				
.5	88.2	83.3	89.5	87.0
1.0	88.0	80.7	86.5	85.1
1.5	86.7	80.2	86.3	84.4
2.0	87.9	79.2	85.3	84.1
2.5	89.0	78.9	85.2	84.4
3.0	87.7	79.6	85.9	84.4
Avg.	87.9	80.3	86.5	84.9

Table 8.1: Response averages and roughness numbers for three sources in six locations

Source location	Monopole	Dipole (moment)		Avg.
		.1m	.2m	
Roughness (dB)				
.5	3.2	6.3	6.0	5.2
1.0	3.7	2.9	2.8	3.1
1.5	4.0	5.1	4.9	4.7
2.0	3.4	5.1	4.8	4.4
2.5	2.8	5.6	5.3	4.6
3.0	2.7	5.3	4.9	4.3
Avg.	3.3	5.1	4.8	4.4

Table 8.1: Response averages and roughness numbers for three sources in six locations

It is evident from this table that the dipole does not always have smaller roughness, as hypothesized (although not by us). In fact, on the average, the monopole would win the roughness contest. Although, when the dipoles are placed "near" the corner they show a remarkably small roughness. Fig. 8-9 showed the frequency response for this location and it can be seen to be very smooth. We should, however take note of the low frequency falloff and the rising roughness which results. The roughness jumps up as the dipole is moved further into the corner principally because the low frequency response falls off to such an extent that the roughness numbers become very high. The detailed response is still fairly smooth. It appears that if one can live with the extremely low output of a dipole in a corner then this is a good source configuration at that location. It also appears that there may be some truth to minimizing the modes since a dipole in a corner would excite few modes – hence its low output. As an interesting comparison for the dipole consider Fig. 8-10. This is the plot for a mid room location.

One can pretty much put a monopole any place without creating a problem, but we must be careful where we put a dipole. Remember that all rear ported enclosures act as a dipole below cutoff.

One solution to the roughness problem is to use multiple sources. In general, the statistics will go as any statistical function when more degrees of freedom are added, namely $1/n$ where n is the number of sources. Adding a second source would be expected to cut the variations in half (on the average) and adding a third to a third of the results shown. There is much to be gained from the first two or three sources, but the benefits would diminish quickly.

The output level definitely rises as we move any of these sources into the corner. This is a well know phenomena and it is generally acknowledged that the best

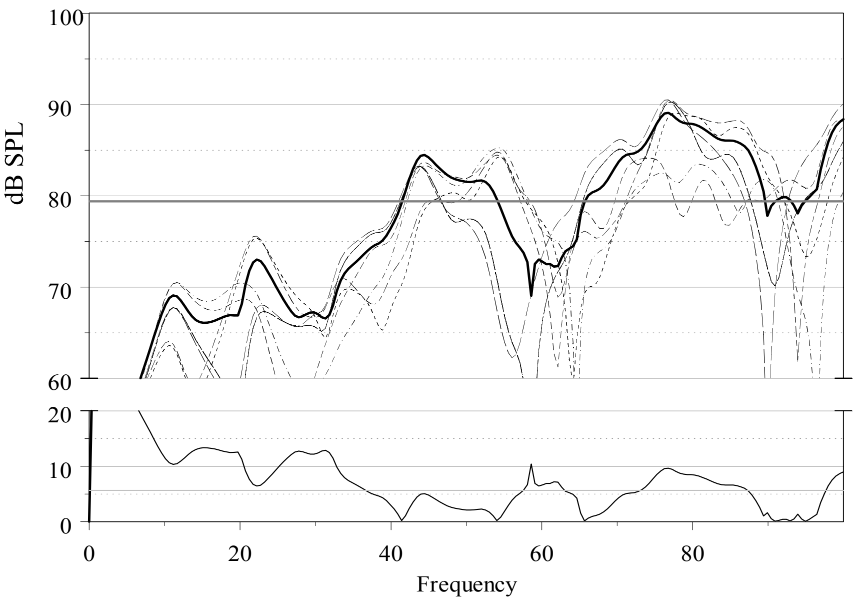

Figure 8-10 - A monopole source in the middle of the room – a worst case

place to put a "subwoofer" is at a junction of boundaries such as a corner. In this way, the *maximum number* of modes will be excited.

No matter how many modes we excite there are still not enough modes at the lowest frequencies to provide for a smooth response. As we have pointed out, the use of significant amounts of damping helps to smooth out these response irregularities (at the sake of reverberation field level). This presents us with a significant problem. In the previous section we discussed the desire for a small room to minimize the use of damping material and yet we have now concluded that for a good low frequency response, we want just the opposite. To make matters worse nearly all of the materials that are typically used in modern room construction are effective dampeners at higher frequencies but not at all effective at low frequencies. It would appear then that typical room treatments in a small rooms are exactly the wrong thing to do.

What we want is substantial low frequency absorption with minimal high frequency absorption, but is this possible? In fact, it is. Without going into the details, it is possible to construct a room such that it has massive walls with hard surfaces but which are in fact suspended on springs (resilient channel) and internally damped with constrained layer damping (high-loss viscoelastic material between two sheets of stiff material). These panels will yield a wall that is flexible and well damped at low frequencies, yet it is rigid and reflective at higher frequencies.

We clearly don't want the kind of construction usually used in auditoriums with an extremely rigid structure and walls and the proficient use of heavy drapery, absorptive seats and thick carpeting. Large rooms and small rooms simply do not behave the same and there is no reason to expect that they should have the same acoustic treatments. Consider the following facts. The damping treatment in a room is seen by a sound wave each time it impinges on this material. Since the "mean free path" (the average path length between successive reflections) for a small room is typically orders of magnitude smaller than that for a large room, any material used in a small room will become that much more effective than in the large room. A patch of absorbing material on a wall will be impinged upon by a sound wave perhaps a thousand times more often in a small room than a large one. This is easy to see by just considering the volume to area ratio of a small volume compared to a large one. One could almost say that the use of sound absorbing material in a small room should be kept to an absolute minimum, especially at higher frequencies.

The take-away point of this chapter is that small rooms behave very differently than larger rooms. One must question each and every assumption used in the theory of room acoustics as applied to large spaces when using these theories in small spaces. From what we have seen, the odds are that the assumptions will not hold up and any theory developed from them must be seriously scrutinized for applicability to small room acoustics.

8.5 Summary

We have shown how the sound field in a room is hypersensitive to small changes in the rooms parameters. This lead us to the conclusion that the room response must be considered as a random variable. We have tried to show how different the acoustics are for a small room when compared to a large room. We have attempted to replace the large room concepts with a small room design goal of high values of low frequency absorption combined with low values of high frequency absorption (and pointed out that this is usually the opposite of one gets). When this is done, the need for increased directivity from the sound source becomes evident.

Finally we have shown that the low frequency source type does make a difference, but it is a difference that depends on several other factors. More low frequency sources are better and for higher output, all sources work better in a corner.