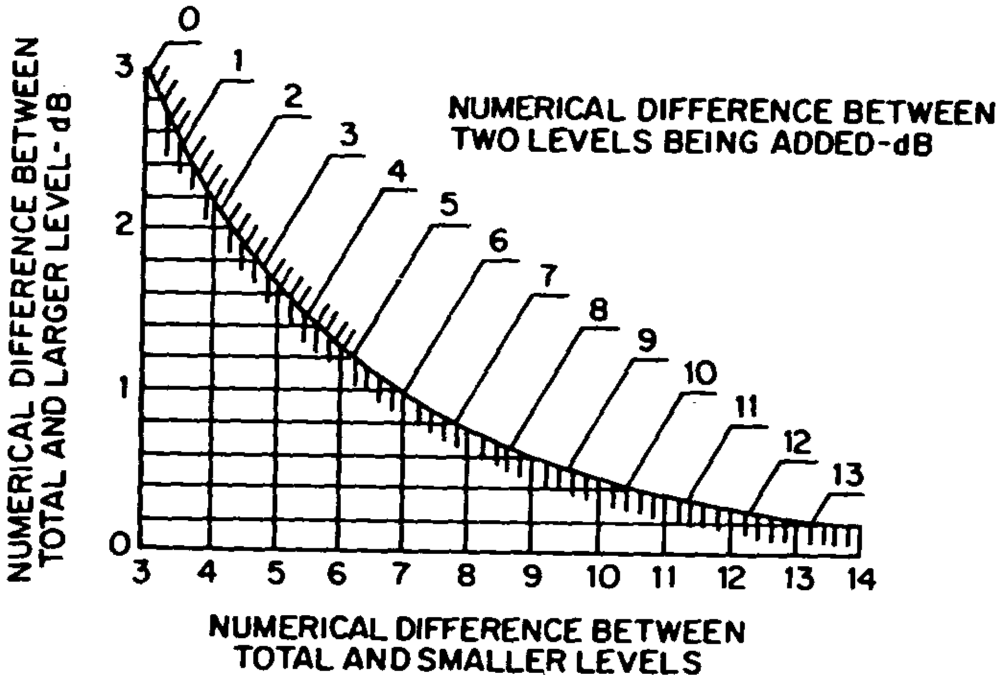



Figure 2.7. Chart for combining decibel levels*.

Table 2.3. Table for obtaining decibel sum of two decibel levels.

| DIFFERENCE BETWEEN TWO DECIBEL LEVELS TO BE ADDED (dB) | AMOUNT TO BE ADDED TO LARGER LEVEL TO OBTAIN DECIBEL SUM (dB) |
|---|---|
| 0 | 3.0 |
| 1 | 2.6 |
| 2 | 2.1 |
| 3 | 1.8 |
| 4 | 1.4 |
| 5 | 1.2 |
| 6 | 1.0 |
| 7 | 0.8 |
| 8 | 0.6 |
| 9 | 0.5 |
| 10 | 0.4 |
| 11 | 0.3 |
| 12 | 0.2 |

*From Handbook of Noise Measurement. 7th ed., A.P.G. Peterson and E.E. Gross, Jr. GenRad, Inc., Concord, MA 01742. This chart is based on one developed by R. Musa. Reprinted by permission of the publisher.

is appropriate for a particular machine, machine component, or process. This aspect of noise problem analysis is closely related to identifying where the noise is coming from: the topic of noise problem diagnosis. To perform even a simple noise problem diagnosis, you must be able to add decibels.

## Decibel Addition

The calculation involved in decibel addition is fundamental to noise control engineering. Suppose we know the sound levels of two separate sources, and we want to know their total when the two sources are operating simultaneously. We make the basic assumption that the noises are random and that they bear no relationship to each other (that is, they do not have the same strong pure tones). The formula for calculating the combined level, $L_c$, of two individual decibel levels $L_1$ and $L_2$, is

$$L_c = L_1 + 10 \log [10^{(L_2 - L_1)/10} + 1]. \quad (2.5)$$

As a practical example, you might have already measured or obtained (at a specified distance or location) the sound levels of two individual sound sources, each operating alone, and you now want to know the sound level (at the same distance) of the two together. For random sounds, the total measured on an SLM would agree (within measurement accuracies of about 1 dB) with the calculated total, using Equation 2.5. Figure 2.7 or Table 2.3 simplifies decibel addition without the formula.

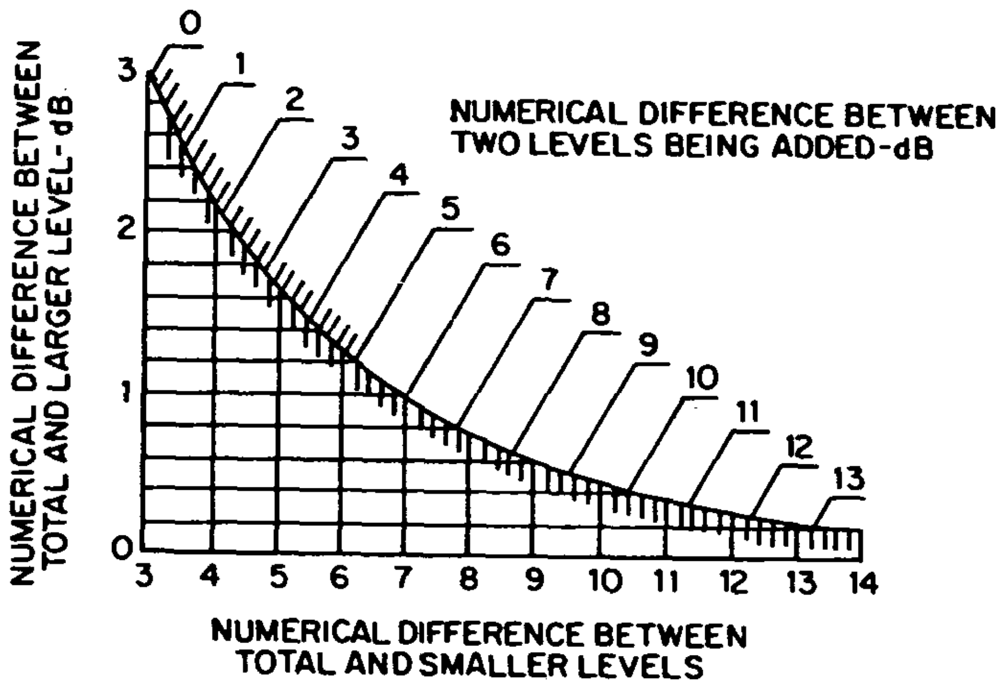
An alternative form of decibel addition, which relies on a few simple rules which can be learned (results accurate to ±1 dB) is:

(1) When two decibel levels are equal or within 1 dB of each other, their sum is 3 dB higher than the higher individual level. For example, 89 dBA + 89 dBA = 92 dBA, 72 dB + 73 dB = 76 dB.

(2) When two decibel levels are 2 or 3 dB apart, their sum is 2 dB higher than the higher individual level. For example, 87 dBA + 89 dBA = 91 dBA, 76 dBA + 79 dBA = 81 dBA.

(3) When two decibel levels are 4 to 9 dB apart, their sum is 1 dB higher than the higher individual level. For example, 82 dBA + 86 dBA = 87 dBA, 32 dB + 40 dB = 41 dB.

(4) When two decibel levels are 10 or more dB apart, their sum is the same as the higher individual level. For example, 82 dB + 92 dB = 92 dB.

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|---|---|
| 0 | 3.0 |
| 1 | 2.6 |
| 2 | 2.1 |
| 3 | 1.8 |
| 4 | 1.4 |
| 5 | 1.2 |
| 6 | 1.0 |
| 7 | 0.8 |
| 8 | 0.6 |
| 9 | 0.5 |
| 10 | 0.4 |
| 11 | 0.3 |
| 12 | 0.2 |

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When adding several decibel levels, begin with the two lower levels to find their combined level, and add their sum to the next highest level. Continue until all levels are incorporated.

Table 2.4 gives an example of how several levels can be added to find their decibel total.

Table 2.4. Example of decibel addition

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Original decibel levels | 85 | 92 | 90 | 84 | 93 | 87 | |
| Rearranged | 84 | 85 | 87 | 90 | 92 | 93 | |
| 84 + 85 = 88 | 88 | | | | | | |
| 88 + 87 = 91 | | 91 | | | | | |
| 91 + 90 = 94 | | | 94 | | | | |
| 92 + 93 = 96 | | | | | 96 | | |
| 94 + 96 = 98 | | | | | | | 98 |

Signals that are *not random* do not follow any of the addition procedures described above. If two identical sources emit strong pure-tone signals at exactly the same frequency, they would be termed *coherent* sources, not random sources. Their total could add up to as much as 6 dB above either single signal, if both sources are exactly equal in level and exactly in phase with each other at the measurement position. If the signals are not exactly in phase, they could interfere destructively with each other, and the measured tones could appear to vanish at the specific measurement position. The occurrence of truly coherent sources is so unlikely in practical plant problems that decibel addition of pure tones exactly in phase at one specified location is almost never considered and can be ignored.

## Identifying Noisy Equipment: Simple Cases

At this point, you are ready to perform some simple evaluations to determine where a noise problem really lies, as a preliminary step in performing noise control. A truly simple, but most illuminating, technique is to turn individual pieces of equipment on and off and to measure and observe the resulting sound levels at the position of interest. Such measurements and observations may reveal the one or two machines that are exceptionally noisy. As an example of how this technique works, assume these measurements are made at an operator position:

- With all equipment running 92 dBA
- With only machine A turned off 92 dBA
- With only machine B turned off 89 dBA
- With only machine C turned off 88 dBA.

These data reveal that machine A is insignificant relative to the total sound level measured (machine A must contribute less than about 83 dBA, otherwise the 92-dBA level would have changed when it was turned off). Machines B and C dominate the noise exposure; the 92-dBA sound level is fully accounted for by the sum of their contributions (88 dBA + 89 dBA = 92 dBA).

When you evaluate noise conditions in this fashion, it is preferable to take octave-band sound pressure level data as well as sound level data. The extra detailed information may be of immediate benefit. Following the above example, you may find the spectra of the 88-dBA and 89-dBA noise to be, respectively, primarily low-frequency and high-frequency in nature. Knowing that high-frequency noise is easier to reduce, you can begin to search for a treatment which will reduce the 89 dBA from machine C by enough so that the contributions from that machine and machine A would total no more than 86 dBA. (Then, 86 dBA + 88 dBA would equal 90 dBA.) You may even estimate a spectrum for the 86-dBA noise which, when combined with the 88-dBA noise spectrum, will produce a 90-dBA total. This can then be used to determine exactly how much noise reduction is required on an octave-band basis. Noise control details can then be considered and designed to enable the reduction to be met.

Other simple measurements may be used to pinpoint important noise contributors of a complex machine. In some cases, a machine may be studied in detail during periods of scheduled downtime. The machine could be operated in various modes, possibly revealing noisy aspects of its operation. You might find, for example, that the noise problem disappears when the pneumatic system is deactivated or that the noise problem is alleviated when a particular component is removed.

The noise control problem is compounded when it is found that several sound sources (either separate pieces of equipment or different components of one piece of equipment) contribute about equally to the total sound level (e.g., three machines, each contributing 96 dBA to a 101-dBA noise environment). When such a situation is encountered, several design alternatives may occur. For the example of the three 96-dBA machines just cited, assume that you want to reduce the 101-dBA level to 90 dBA, an 11-dBA reduction. First, this reduction could be achieved by reducing the noise emission of each machine by 11 dB. Hence, by decibel addition, 85 dBA + 85 dBA + 85 dBA = 90 dBA.

Or, two machines could be reduced by 13 dB, and one machine could be reduced only 8 dB. Thus,

83 dBA + 83 dBA + 88 dBA = 90 dBA.

Or, one machine could be reduced by 19 dB, one by 12 dB, and one by 7 dB. Thus,

77 dBA + 84 dBA + 89 dBA = 90 dBA.

In each case, the result would be 90 dBA for the sum of the three treated machines. Clearly, the amount of noise reduction needed for each machine is not a fixed quantity, and the noise control engineer has some latitude in choosing which equipment to treat and to what degree.

## General Procedure

In the previous section, we discussed identification of the source of a problem noise in situations where it is possible to turn production equipment on and off. Often, the noise control engineer is faced with the task of making the necessary identification without the luxury of equipment being operated to his convenience. How does he do it?

The noise control engineer will turn to his knowledge of sound fields and sound behavior. (These topics are discussed in detail later.) Essentially, the noise control engineer will couple (1) his knowledge about how sound propagates from one location to another with (2) data obtained at or near a suspected noise contributor to verify whether his suspicions are correct. The sound level around a noise source, if that source is significant, is almost invariably higher near it, or, to put it another way, noise makers are louder close by. You can usually learn something about the strength of the noise source – how much sound it radiates – by measuring the sound field near the source.

## Source Strength: Sound Power Level

The amount of sound radiated by a source is determined by its sound power, somewhat analogous to the power rating of electric light bulbs – 40 W, 75 W, 100 W, etc. In fact, sound power is also expressed in units involving watts. To relate sound power to familiar subjects, a mosquito emits a sound power of about $10^{-11}$ W, and a clap of thunder radiates a peak instantaneous sound power somewhere over a million watts. The average sound power of human speech at normal voice level is about $10^{-4}$ W, a symphony orchestra playing loud passages radiates about 10 W of sound power, and a 4-engine jet airliner during takeoff has a sound power of about $10^4$ W.

With such a large range of power for the many commonplace sound sources, it is convenient to use decibels here, too, to compress the range into manageable numbers. The reference sound power base is $10^{-12}$ W, and the sound power *level* ($L_w$, in dB) of a source relative to this base is

$$L_w = 10 \log \left(\frac{\text{Power radiated, watts}}{10^{-12}\text{W}}\right)$$

The mosquito then has a sound power level of about 10 dB (re $10^{-12}$ W), and the jet aircraft has a takeoff sound power level of about 160 dB (re $10^{-12}$ W).

Since decibels are used both with sound pressure level and sound power level, it is always necessary to indicate clearly which unit is being used. Because, as mentioned earlier, it is awkward and inconvenient to refer sound pressure levels repeatedly to the sound pressure reference base of 20 micropascals, it is usual to reference the power level base $10^{-12}$ W to assure that sound power levels are being used. Hence, the term "(re $10^{-12}$ W)" is used in the expressions above for the sound power levels of the mosquito and the jet.

There is another practical reason to reference the quantity $10^{-12}$ W. Before the United States joined the International Standards Organization in the use of common terminology in acoustics, the sound power level base used in this country was $10^{-13}$ W. Before 1963 to 1965, acoustics literature in the United States regularly referred to the $10^{-13}$ W base for sound power level data. If data from those earlier periods are used in current studies, determine positively the power base of the data. Subtract 10 dB from sound power levels relative to the $10^{-13}$ W base to convert them to values relative to the $10^{-12}$ W base.

How can sound power data be used in source diagnosis? The sound power level radiated by an "ideal point source" (a source radiating sound uniformly in all directions) is related to the sound pressure level at a distance r by the following equations:

$$L_w = L_p + 10 \log 4\pi r^2, \tag{2.6}$$

where r is expressed in meters, or

$$L_w = L_p + 10 \log 4\pi r^2 - 10, \tag{2.7}$$

where r is expressed in feet. For these two equations, the source is assumed to radiate its sound with no nearby reflecting surfaces. This would be known as spherical radiation in a free field, a relationship fundamental to source diagnosis. To preview its use, note that if we measure $L_p$ at a location close-in to the noise source, we can calculate $L_w$ for that source and then determine $L_p$ due to that source at a more distant location, such as at a nearby residence. In practice, many sound sources do not radiate sound uniformly in all directions, and reflecting surfaces can be nearby.

For an ideal point source located on or close to a large-area floor or at or near the ground in a large open area, the sound radiates hemispherically, and the above equations become: for r in meters,

$$L_w = L_p + 10 \log 2\pi r^2, \tag{2.8}$$

and, for r in feet,

$$L_w = L_p + 10 \log 2\pi r^2 - 10. \tag{2.9}$$

In the more general case, the source is not a point source; instead, it has finite values of length, width, and height. In this case, sound power and sound pressure levels are interrelated by the equations:

$$L_w = L_p + 10 \log S \tag{2.10}$$

for S expressed in square meters, or

$$L_w = L_p + 10 \log S - 10 \tag{2.11}$$

for S expressed in square feet. In these last two equations, S is the area of an imaginary shell all around the source, and $L_p$ is the sound pressure level that exists at any point on that imaginary shell.

In a further extension of Equations 2.10 and 2.11, suppose that the source does not radiate its sound uniformly through all portions of the shell. Perhaps one part of a large, complex sound source radiates higher sound pressure levels (SPLs) than some other part of the source. For such situations, Equations 2.10 and 2.11 must be broken down into several parts, where $L_{p_1}$ is the SPL at one element of area $S_1$ on the shell, $L_{p_2}$ is a different SPL at another element of area $S_2$, and so on over the entire range of $L_p$ values over the entire area. Then,

$$L_W = \sum_{i=1}^{n} \left[ L_{p_i} + 10 \log S_i \right] \text{(for S in m}^2\text{)} \qquad (2.12)$$

or

$$L_W = \sum_{i=1}^{n} \left[ L_{p_i} + 10 \log S_i - 10 \right] \text{(for S in ft}^2\text{)}. \qquad (2.13)$$

As an example, Figure 2.8 shows an imaginary shell around a sound source of interest, at a 1-m distance. The source dimensions are 2 m × 3 m × 5 m, as shown in the sketch. The north and south surfaces of the imaginary shell each have an area of 21 $m^2$, the

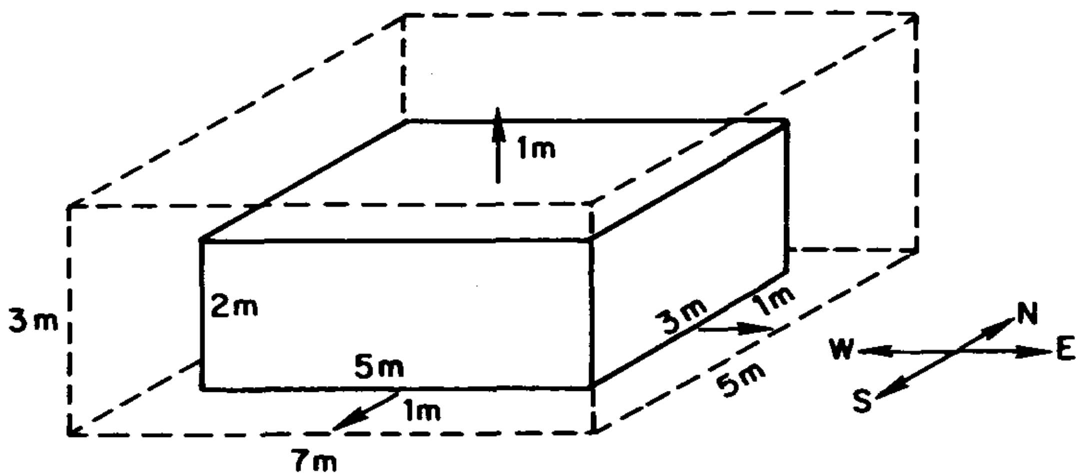


Figure 2.8. Assumed sound source (solid lines) on a factory floor, surrounded by an imaginary shell (dashed lines) at 1-m distance.

east and west ends have an area of 15 m each, and the top of the shell has an area of 35 $m^2$. For this simple example, suppose the SPL all over the north surface of the shell is uniform at 98 dB; for the south surface, it is 93 dB; for the east end, it is 88 dB; for the west end, it is 90 dB; and for the top surface, it is 95 dB. The total sound power radiated from this source would be as follows, using Equation 2.12:

$$\begin{aligned} L_W = {} & (98 + 10 \log 21) \text{ dB (N)} \\ & + (93 + 10 \log 21) \text{ dB (S)} \\ & + (88 + 10 \log 15) \text{ dB (E)} \\ & + (90 + 10 \log 15) \text{ dB (W)} \\ & + (95 + 10 \log 35) \text{ dB (Top)} \end{aligned}$$

These components are to be added by decibel addition. Thus,

$$L_w = 111.2 + 106.2 + 99.8 + 101.8 + 110.4$$

$$= 114.9 \text{ dB or } 115 \text{ dB re } 10^{-12} \text{ W}.$$

Calculations can be carried out to 0.1-dB values, but the final value should be rounded off to the nearest whole number.

Two practical considerations limit the validity of this example. First, in practice it is unlikely that a uniform sound level would exist over an entire large area of the imaginary shell, so it might be necessary to take several SPL values over each large area of interest and to assign a subdivided area value to each SPL value. Second, when SPL measurements are made close to a relatively large-size source, the sound is not radiating as though it were from a point source in a free field. Instead, the SPL value is taken in the *near field* of the sound source, where the sound field is distorted and is not necessarily representative of the true total sound power that would be radiated to a large distance out in the free field. As a result, errors of a few decibels may be encountered at these close-in distances from large sources, and it is essentially impossible to predict the amount of error to be expected. Thus, be prepared to have an unknown error (possibly up to 5 to 8 dB for large sources, but fairly negligible for quite small sources).

In spite of these drawbacks, the concept of sound power level is very helpful in identifying and diagnosing sound sources. To illustrate this assistance, suppose the microphone of a sound level meter can be brought up to within 5 cm of a small sound source in a large machine, and the sound pressure level is found to be 105 dB in the 1000-Hz octave band. Over another, much larger, area of the machine, the close-in sound level is 95 dB in the 1000-Hz octave band. Estimate the sound power levels of these two sources to determine the controlling source at this frequency. Suppose the 105-dB value is found to exist over an area of about 100 cm × 10 cm, or 1000 $cm^2$ (=0.1 $m^2$), whereas the 95-dB value is found to exist over a surface area of about 2.5 × 4 m, or 10 $m^2$. From Equation 2.10, the approximate sound power level of the small-size source is

$$L_w = 105 + 10 \log 0.1$$

$$= 95 \text{ dB re } 10^{-12} \text{ W},$$

while the approximate sound power level of the large-area source is

$$L_W = 95 + 10 \log 10$$

$$= 105 \text{ dB re } 10^{-12} \text{ W.}$$

Even if the power level values are in error by a few decibels, this comparison indicates that the large-area source radiates more total sound power than the small-size source, even though the small source has a higher localized sound pressure level. For noise control on that machine, the noise from the large area source must be reduced by about 10 dB before it is necessary to give serious consideration to the small source.

For another illustration of how sound power level data are used in source diagnosis, look at Figure 2.9. It shows the noise spectrum found at the property line of a plant and a sound spectrum indicative of a target goal for the situation. Note that the sound pressure levels are excessive in the 125-Hz to 8000-Hz octave bands.

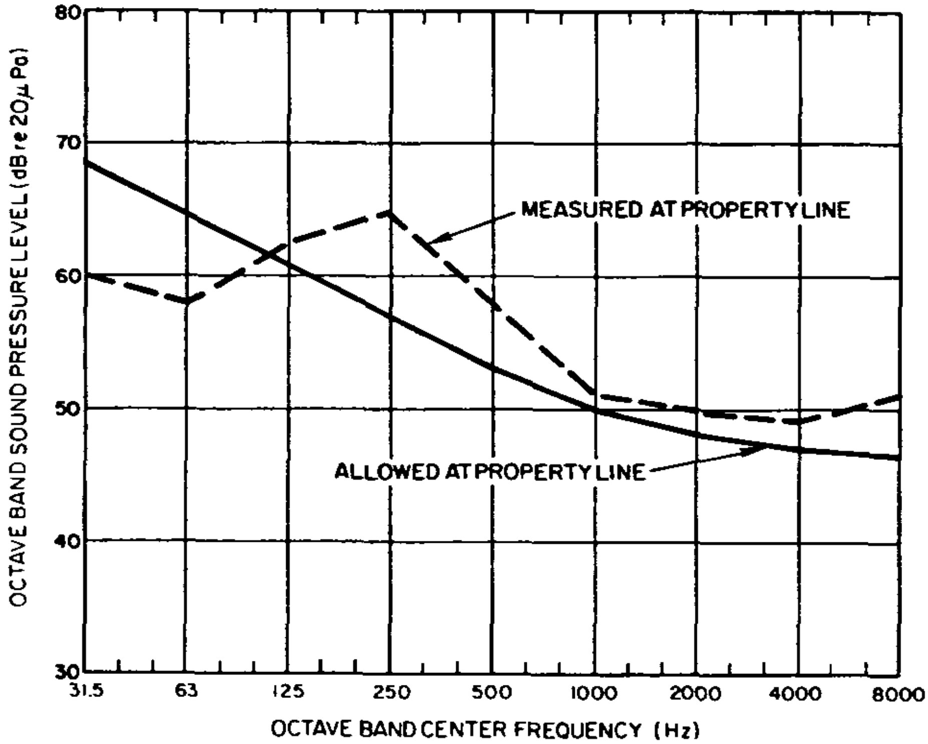


Figure 2.9. Hypothetical problem situation.

Close-in data, obtained 1 m from each of the three possible sources (Figure 2.10) of the property line noise, were then examined to determine which noise sources should be treated. From Equation 2.10, the power level of each source is obtained, and from Equation 2.8, the expected contribution of each source to the property line measurement is estimated (in this example,

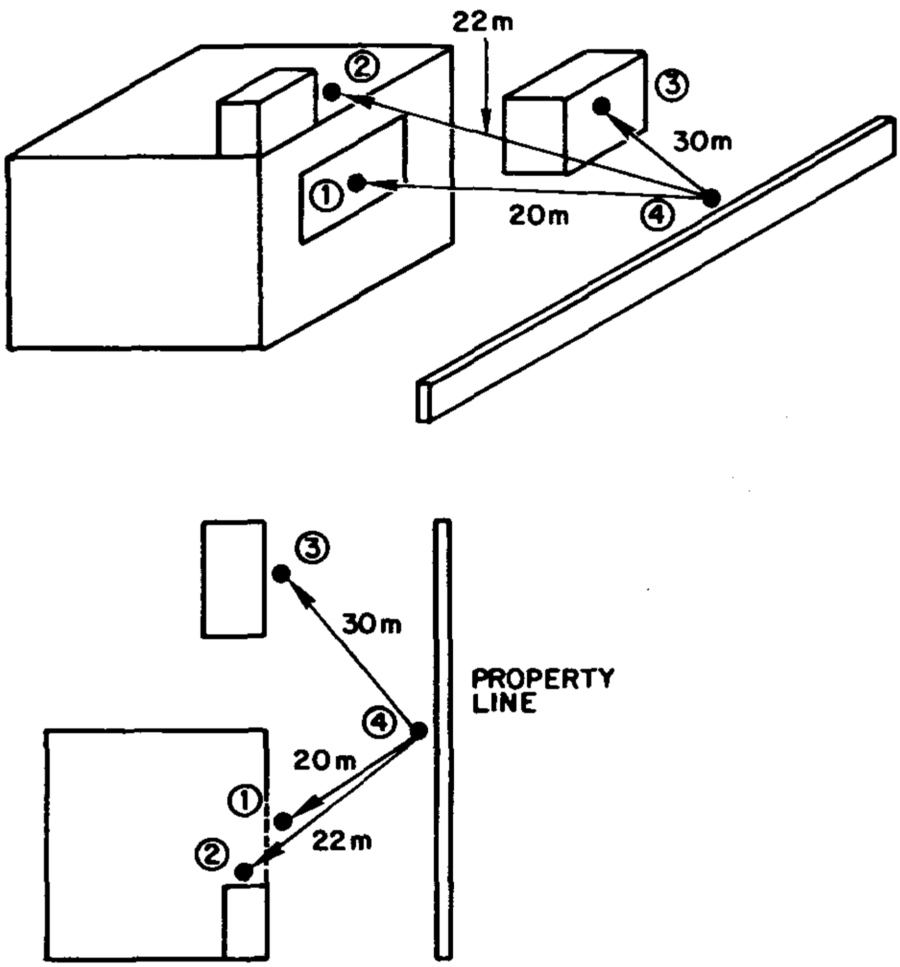


Figure 2.10. Location of noise sources (1-3) relative to property level position (4) for use in example on these pages.

each noise source is assumed to radiate hemispherically). Figure 2.11 illustrates the results of the computations shown in Table 2.5. The calculations indicate the vent noise is responsible for the 31.5-Hz and 63-Hz octave-band sound pressure levels, the compressor noise is responsible for the 125-Hz to 500-Hz octave-band sound pressure levels and partly responsible for the 1000-Hz and 2000-Hz octave-band sound pressure levels, and that sound coming through the window contributed to or is responsible for sound pressure levels in the 1000-Hz to 8000-Hz bands.

Because the 31.5-Hz and 63-Hz octave-band levels are not consequential to the problem, the vent need not be treated. However, both the window and compressor do require treatment, and the

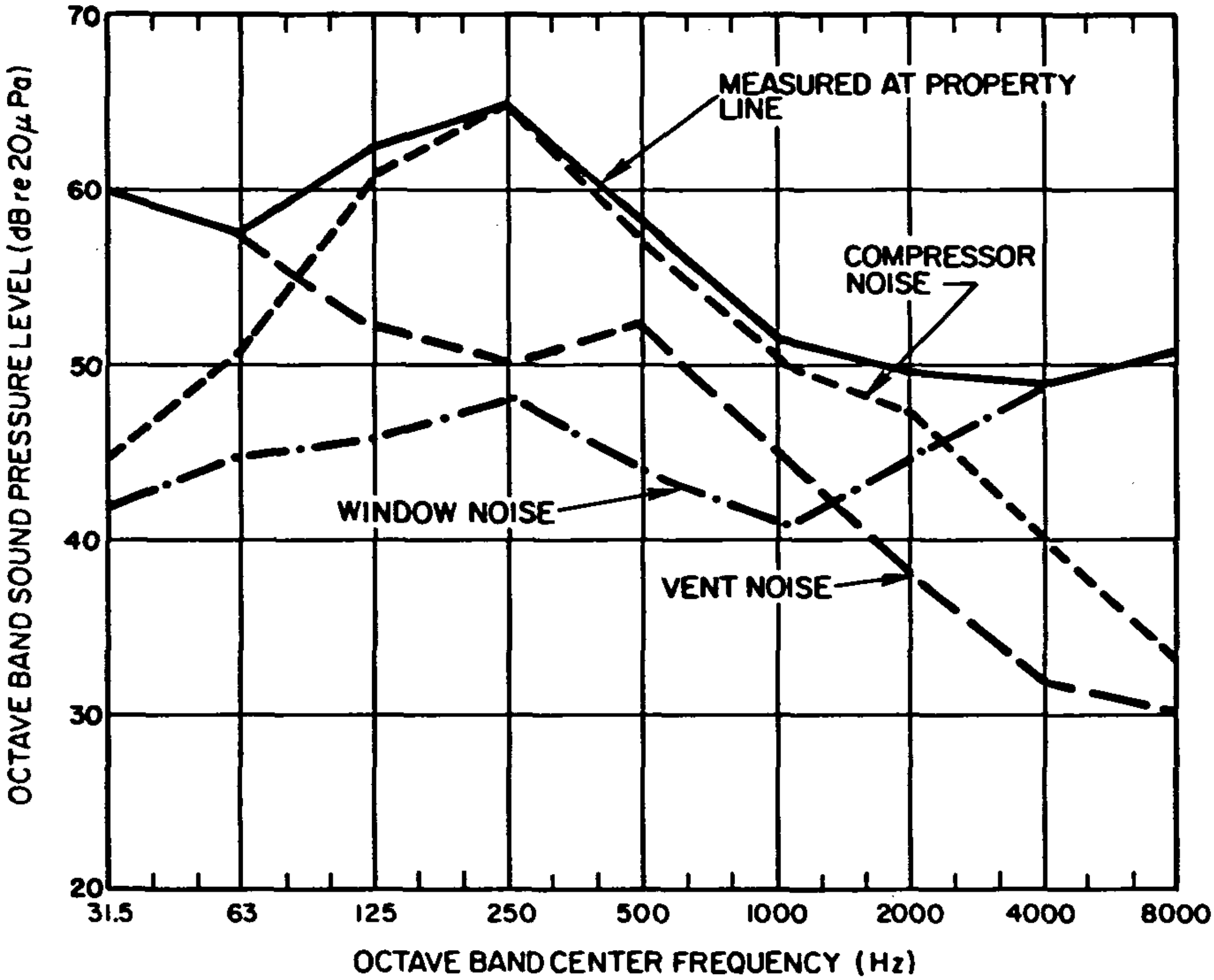


Figure 2.11. Results of power level extrapolations for problem shown in Figure 2.9.

Table 2.5. Calculations for example problem discussed on previous pages.

| DESCRIPTION | Octave Band Center Frequency in Hz 31.5 | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|---|
| Source 1 (window on side of building) 2 m × 4 m; shell surrounding window at 1 m distance has area of 4 m × 6 m = 24 $m^2$ | | | | | | | | | |
| $L_w$ of window = $L_p$ + 10 log 24 = $L_p$ + 13.8, say $L_p$ + 14 | | | | | | | | | |
| $L_p$ at 1 m | 82 | 65 | 66 | 68 | 64 | 61 | 65 | 69 | 71 |
| plus | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 14 |
| $L_w$ of window | 76 | 79 | 80 | 82 | 78 | 75 | 79 | 83 | 85 |
| Window noise at property line (from Eq. 2.8) $L_p=L_w-10\log 2\pi r^2$; r=20m $L_p = L_w - 10\log 2\pi 20^2 = L_w - 34.0$ | | | | | | | | | |
| minus | 34 | 34 | 34 | 34 | 34 | 34 | 34 | 34 | 34 |
| Window noise | 42 | 45 | 46 | 48 | 44 | 41 | 45 | 49 | 51 |
| Source 2 (small vent on roof) surface area of sphere centered at vent radius of 1 m = $4\pi r^2$ = 12.56 $m^2$ | | | | | | | | | |
| $L_w$ of vent = $L_p$ + 10 log 12.56 = $L_p$ + 10.99, say $L_p$ + 11 | | | | | | | | | |
| $L_p$ at 1 m | 84 | 81 | 76 | 74 | 76 | 69 | 62 | 56 | 54 |
| plus | 11 | 11 | 11 | 11 | 11 | 11 | 11 | 11 | 11 |
| $L_w$ vent | 95 | 92 | 87 | 85 | 87 | 80 | 73 | 67 | 65 |
| Vent noise at property line (from Eq. 2.8) $L_p=L_w-10\log 2\pi r^2$; r=22m $L_p = L_w - 10\log 2\pi 22^2 = L_w - 34.8$, say $L_w$ = 35 | | | | | | | | | |
| minus | 35 | 35 | 35 | 35 | 35 | 35 | 35 | 35 | 35 |
| Vent noise | 60 | 57 | 52 | 50 | 52 | 45 | 38 | 32 | 30 |
| Source 3 (compressor) 2 × 3 m; shell surrounding compressor with 1 m distance = 3 m × 5 m = 15 $m^2$; $L_w$ compressor = $L_p$+10 log 15 = $L_p$ + 11.8 dB, say $L_p$ + 12 dB | | | | | | | | | |
| $L_p$ at 1 m | 71 | 77 | 87 | 91 | 84 | 76 | 73 | 66 | 59 |
| plus | 12 | 12 | 12 | 12 | 12 | 12 | 12 | 12 | 12 |
| $L_w$ compressor | 83 | 89 | 99 | 103 | 96 | 88 | 85 | 78 | 71 |
| Compressor noise at property line (from Eq. 2.8) $L_p=L_w-10\log 2\pi r^2$ (r=30 m) $L_p=L_w-10\log 2\pi 30^2 = L_w-37.5$, say $L_w$=38 | | | | | | | | | |
| minus | 38 | 38 | 38 | 38 | 38 | 38 | 38 | 38 | 38 |
| Compressor noise | 45 | 51 | 61 | 65 | 58 | 50 | 47 | 40 | 33 |

amount of treatment required is indicated by the difference between the estimated levels from the window or compressor and the target goal (in those octaves dominated by the individual source).

These examples illustrate the importance of obtaining close-in SPL values near each operating mechanism or component of a source and of estimating the area of that component or the area through which its SPL is radiating. Sound control work should be directed to those components that yield large values of sound power level. It is also necessary to investigate the frequency variation of the component sources as measurements are being carried out. Some components may shift from small-valued sound sources in some frequency regions to high-valued sources in other frequency regions.

## Influence of Room Acoustics

In the previous section, the sound source was presumed to be located in a large open area, so that nearby reflecting surfaces (other than the floor or ground) would not alter the free-field radiation of the sound. In most indoor plant situations, the confining walls and ceiling of the work space keep much of the sound from escaping to the outdoors. Instead, each *ray* of sound from the source strikes a solid surface and is reflected to some other direction inside the room. That same ray may travel 300 m and be reflected a dozen times before its energy is sufficiently dissipated for it to be ignored. In the meantime, other rays of sound are also radiated and reflected all around the room until they are dissipated. In a small room, the sound pressure levels caused by the confinement of sound can be built up to values as much as 15 to 30 dB above the values that would exist at comparable distances outdoors. This build-up of sound can influence the sound level at the operator position of a machine. In fact, a machine that might have an 85-dBA sound level at a 2-m distance when tested outdoors in a large, open parking lot could produce a sound level of 95 to 100 dBA at the same distance when it is moved indoors into a small, highly reverberant room. Note that the sound power level of the source didn't change, but that the acoustic environment made a major difference in the sound levels. To analyze this type of situation, it is necessary to know the influence of the room conditions on the sound field around the machine. This general subject, referred to as "room acoustics," can be almost as important as the sound power of the source in determining sound levels to the machine operator or to other people working in a room where machines are in operation.

## Room Constant or Room Absorption

To work quantitatively in the subject of room acoustics, you should know how to calculate and to use the term *room constant*, designated by R, or a similar term, *room absorption*, designated by A. In this *Manual*, room constant R is used.

The room constant for a room is calculated from the equation:

$$R = S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \ldots + A_1 + A_2 + \ldots \qquad (2.14)$$

where $S_1$ is the area of some surface of a room that has a sound absorption coefficient $\alpha_1$, $S_2$ is the area of another surface of the room having a sound absorption coefficient $\alpha_2$, and so on, until all surface areas of the room are added, including all walls, doors, windows, the floor, the ceiling, and any other surfaces that make up the room boundary. The S values may be expressed either in square feet or square meters, and the calculated R value will be in the same unit. The $\alpha$ values are called Sabin sound absorption coefficients and are given in various textbooks for most room finish materials and in the catalogues of manufacturers and suppliers for their sound absorption products, such as glass fiber, mineral wool tiles or panels, or sound-absorbing cellular foam products. A sound absorption coefficient of 0.6 is intended to mean that 60% of the sound energy in a wave will be absorbed (and 40% reflected) each time the sound wave strikes a surface of that material. ASTM C423-66* specifies the method of measurement of the Sabin absorption coefficients. The $A_1$, $A_2$, etc. values of Equation 2.14 are lumped constants of absorption, provided by suppliers for some acoustical products (such as ceiling-hung absorbent baffles) and whose units may be either square feet-Sabins or square meter-Sabins (1 $ft^2$-Sabin = 1 $ft^2$ of perfect absorption; 1 $m^2$-Sabin = 1 $m^2$ of perfect absorption). The resulting value of R in Equation 2.14 is in units of $ft^2$-Sabin or $m^2$-Sabin, consistent with the other area units used in the equation.

Table 2.5 gives sound absorption coefficients for several building materials that are not normally regarded as absorptive. Note that the coefficients are quoted for the 6 octave-band center frequencies of 125 Hz to 4000 Hz, and that the coefficients vary with frequency. Thus, the room constant R varies with frequency, and Equation 2.14 must be calculated for each frequency of interest. Sound absorption coefficients are not measured or quoted for 31.5, 63, and 8000 Hz. Relatively few noise sources cause problems at these low and high frequencies.

An example of a room constant calculation illustrates the use of Equation 2.14. A room is 40 m long, 10 m wide, and 5 m high. The floor is a thick concrete slab, the two 40-m-long walls are of painted concrete block, the two 10-m-wide walls are made up of gypsum board on 2-in. × 4-in. studs, and the ceiling is the exposed underside of an overhead concrete floor slab. To simplify, ignore two doors in the room. The absorption coefficients of these materials are given in Table 2.6. The room constant calculation at 1000 Hz is:

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*Or latest version

Table 2.6. **Coefficients of general building materials and furnishings.**

| Materials | Coefficients 125 Hz | 250 Hz | 500 Hz | 1000 Hz | 2000 Hz | 4000 Hz |
|---|---|---|---|---|---|---|
| Brick, unglazed | .03 | .03 | .03 | .04 | .05 | .07 |
| Brick, unglazed, painted | .01 | .01 | .02 | .02 | .02 | .03 |
| Carpet, heavy, on concrete | .02 | .06 | .14 | .37 | .60 | .65 |
| Same, on 40 oz hairfelt or foam rubber | .08 | .24 | .57 | .69 | .71 | .73 |
| Same, with impermeable latex backing on 40 oz hairfelt or foam rubber | .08 | .27 | .39 | .34 | .48 | .63 |
| Concrete Block, coarse | .36 | .44 | .31 | .29 | .39 | .25 |
| Concrete Block, painted | .10 | .05 | .06 | .07 | .09 | .08 |
| Fabrics | | | | | | |
| Light velour, 10 oz per sq yd, hung straight, in contact with wall | .03 | .04 | .11 | .17 | .24 | .35 |
| Medium velour, 14 oz per sq yd, draped to half area | .07 | .31 | .49 | .75 | .70 | .60 |
| Heavy velour, 18 oz per sq yd, draped to half area | .14 | .35 | .55 | .72 | .70 | .65 |
| Floors | | | | | | |
| Concrete or terrazzo | .01 | .01 | .015 | .02 | .02 | .02 |
| Linoleum, asphalt, rubber or cork tile on concrete | .02 | .03 | .03 | .03 | .03 | .02 |
| Wood | .15 | .11 | .10 | .07 | .06 | .07 |
| Wood parquet in asphalt on concrete | .04 | .04 | .07 | .06 | .06 | .07 |
| Glass | | | | | | |
| Large panes of heavy plate glass | .18 | .06 | .04 | .03 | .02 | .02 |
| Ordinary window glass | .35 | .25 | .18 | .12 | .07 | .04 |
| Gypsum Board, 1/2 in. nailed to 2×4's 16 in. o.c. | .29 | .10 | .05 | .04 | .07 | .09 |
| Marble or Glazed Tile | .01 | .01 | .01 | .01 | .02 | .02 |
| Openings | | | | | | |
| Stage, depending on furnishings | | | .25 - .75 | | | |
| Deep balcony, upholstered seats | | | .50 - 1.00 | | | |
| Grills, ventilating | | | .15 - .50 | | | |
| Plaster, gypsum or lime, smooth finish on tile or brick | .013 | .015 | .02 | .03 | .04 | .05 |
| Plaster, gypsum or lime, rough finish on lath | .14 | .10 | .06 | .05 | .04 | .03 |
| Same, with smooth finish | .14 | .10 | .06 | .04 | .04 | .03 |
| Plywood Paneling, 3/8-in. thick | .28 | .22 | .17 | .09 | .10 | .11 |
| Water Surface, as in a swimming pool | .008 | .008 | .013 | .015 | .020 | .025 |
| Air, Sabins per 1000 cu ft at 50% RH | | | | .9 | 2.3 | 7.2 |

ABSORPTION OF SEATS AND AUDIENCE

Values given are in Sabins per square foot of seating area or per unit

| | 125 Hz | 250 Hz | 500 Hz | 1000 Hz | 2000 Hz | 4000 Hz |
|---|---|---|---|---|---|---|
| Audience, seated in upholstered seats, per sq ft of floor area | .60 | .74 | .88 | .96 | .93 | .85 |
| Unoccupied cloth-covered upholstered seats, per sq ft of floor area | .49 | .66 | .80 | .88 | .82 | .70 |
| Unoccupied leather-covered upholstered seats, per sq ft of floor area | .44 | .54 | .60 | .62 | .58 | .50 |
| Wooden Pews, occupied, per sq ft of floor area | .57 | .61 | .75 | .86 | .91 | .86 |
| Chairs, metal or wood seats, each, unoccupied | .15 | .19 | .22 | .39 | .38 | .30 |

$$
\begin{aligned}
R_{1000} &= 2 \times 40 \times 10 \times 0.02 \text{ (floor, ceiling)} \\
&\quad + 2 \times 40 \times 5 \times 0.07 \text{ (40-m walls)} \\
&\quad + 2 \times 10 \times 5 \times 0.04 \text{ (10-m walls)} \\
&= 16 + 28 + 4 \\
&= 48 \text{ m}^2\text{-Sabin} \; . \qquad (2.15)
\end{aligned}
$$

Now, suppose a suspended acoustic tile ceiling is installed under the overhead slab. The ceiling height is reduced to 4.5 m. The sound absorption coefficients of the ceiling are as follows:

| frequency, Hz | 125 | 250 | 500 | 1000 | 2000 | 4000 |
|---|---|---|---|---|---|---|
| coefficient | 0.4 | 0.5 | 0.72 | 0.90 | 0.94 | 0.82 |

The room constant calculation at 1000 Hz is:

$$
\begin{aligned}
R_{1000} &= 40 \times 10 \times 0.02 \text{ (floor)} \\
&\quad + 40 \times 10 \times 0.90 \text{ (ceiling)} \\
&\quad + 2 \times 40 \times 4.5 \times 0.07 \text{ (40-m walls)} \\
&\quad + 2 \times 10 \times 4.5 \times 0.04 \text{ (10-m walls)} \\
&= 8 + 360 + 25.2 + 3.6 \\
&= 396.8 \text{ m}^2\text{-Sabin} \qquad (2.16)
\end{aligned}
$$

You may wish to calculate the room constants at other frequencies.

Two generalizations may be drawn from the room constant discussion and calculations: (1) The room constant value increases as the room volume increases, because the surface areas must increase to accommodate the larger volume; and (2) the relatively high values of the Sabin absorption coefficients (at least in the 500- to 4000-Hz frequency region, which is important in terms of A-weighted sound levels) wield strong influences on the room constant when acoustic absorption material is used.

Noise Reduction Coefficients (NRC)--
This is a term that is used widely as a single-number figure-of-merit of sound-absorbing materials. NRC is the arithmetic average of the sound absorption coefficients of 250, 500, 1000, and 2000 Hz, rounded off to the nearest 0.05. Some sound absorption materials of 1-in. to 3-in. thickness have Sabin absorption coefficients as high as 0.90 to 0.99 in the 1000- to 2000-Hz region, and NRC values of these products range from about 0.65 to about 0.90. However, these products may have Sabin coefficients of only about 0.15 to 0.40 in the 125-Hz to 250-Hz region. Larger

thicknesses will cause increases in the low-frequency absorption coefficients.

Sound Distribution in a Room--
Figure 2.12 shows the influence of sound level distribution in a room as a function of the distance from a sound source and the value of room constant. Suppose a worker is 1 m from a sound source in a room whose room constant is 50 $m^2$-Sabin at 1000 Hz. (In a complete analysis, room constants would be calculated for all octave bands, and the A-weighted sound level would be calculated from the octave-band sound pressure levels.) At that position, the worker experiences a sound pressure level of 93 dB in the 1000-Hz band. Find the point on Figure 2.12 that corresponds to a distance of 1 m and a room constant of 50 $m^2$-Sabin. The *relative sound pressure level* value for this point is about -7.5 dB, as read from the vertical scale on the right of the figure. Suppose the worker backs away from that machine to a distance of 4 m. The relative SPL drops to about -11 dB, indicating a sound pressure level reduction of about 3.5 dB. This room is so small and reverberant that the sound level remains almost constant throughout the room, except at positions quite close to the source. Next, suppose that with the use of acoustic absorption material, the room constant is increased to 200 $m^2$-Sabin. At the 1-m distance, the relative SPL is about -10 dB, and at a 4-m distance, the relative SPL is about -17 dB. This finding indicates that sound pressure levels in the room at a distance of 6 or more m from

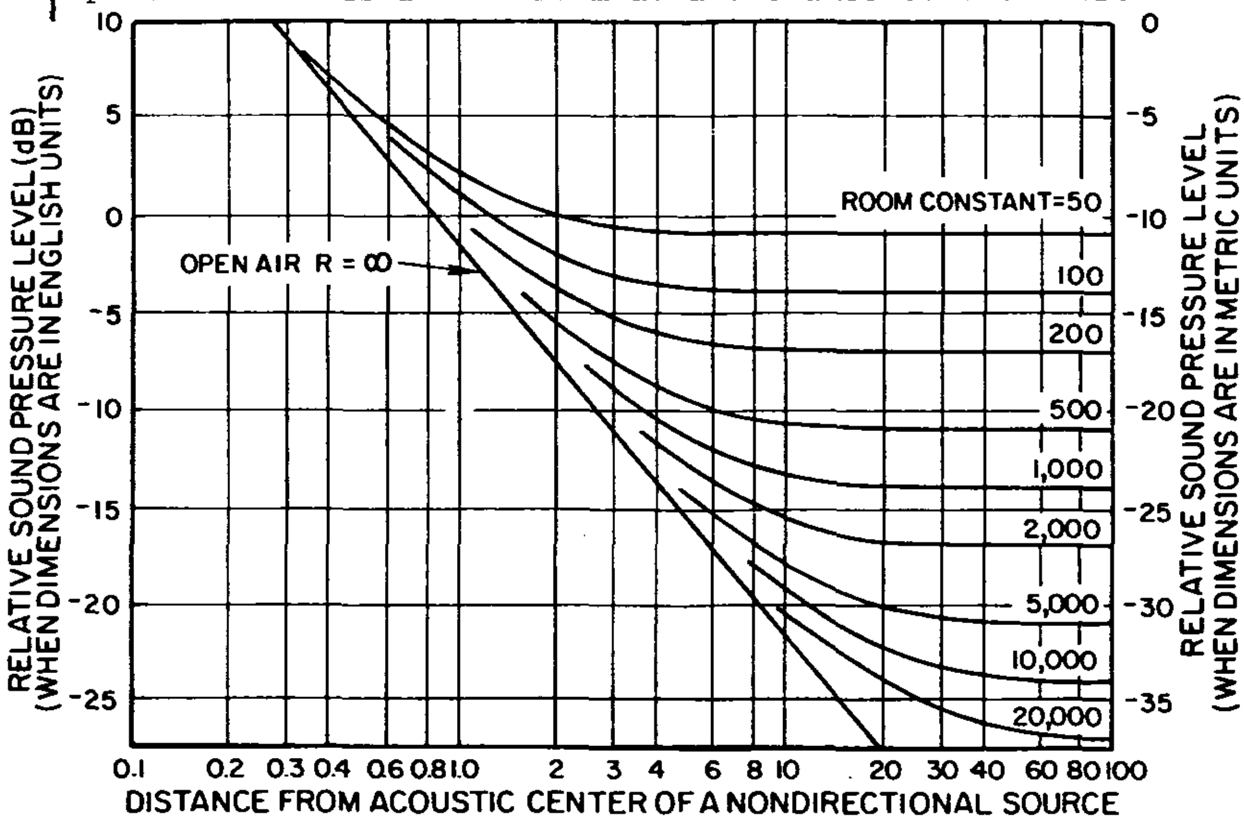


Figure 2.12. Sound level distribution in spaces with various room constants.

the sound source could be reduced by about 7 dB at 1000 Hz with this application of sound-absorbing material. Note, however, at very close distances to the sound source, there is less effect from the addition of absorption material. At a 3-m distance, the change is only about 4 dB, and at a 1-m distance, only about 2 dB. This illustration summarizes briefly the value of sound absorption in a room: It can be quite beneficial in reducing sound levels for people located at large distances from a sound source, but it is not very beneficial to an operator who must remain at a position very close to the source. What this example emphasizes, however, is the importance of devising methods for keeping the operator at greater distances from his machine, so that sound absorption in the room can be beneficial.

As an exercise in using Figure 2.12, study the sound level changes for workers 1 m and 10 m from a sound source in the room whose room constant was calculated above, with and without an acoustic tile ceiling (see Equations 2.15 and 2.16 for the calculated room constants at 1000 Hz). At a 1-m distance, Figure 2.12 shows a reduction of about 2.5 dB in going from an R = 50 $m^2$-Sabin room to an R = 400 $m^2$-Sabin room. At a 10-m distance, a reduction of about 10 dB is achieved when the sound absorptive ceiling is added.

In a typical plant situation, a machine operator may spend most of the time about 1 to 2 m from the nearest machine, but remain within about 5 to 20 m from a number of other machines in the same room. By methodically working out the decibel sum of all the machine sound levels to that operator for a *bare* room (with no acoustic absorption) and for a *treated* room (with sound absorption material added), it is possible to calculate the approximate sound level reduction that would be achieved. For various geometries of room size, machine distances, and number of machines, the benefit can range from 0 dB (no benefit) to as much as 10 to 12 dB. The calculation is inexpensive, and, if the calculations should reveal that a 10- to 12-dB reduction is possible, adding sound absorption material may also be a relatively inexpensive solution to a plant noise problem.

Although noise control treatments are discussed in detail elsewhere in this *Manual*, the noise reduction aspects of the room constant calculations are offered here as a part of the noise problem evaluation.

Sound Power Level Application

In previous examples, Figure 2.12 was used to show that a sound pressure level could vary as a function of distance from the source and room constant of the space. This figure can also be used to estimate sound pressure levels when the sound power level of a source is known. The equation is:

$$L_{p_{D,R}} = L_w + \text{REL SPL}_{D,R} , \tag{2.17}$$

where $L_w$ is the sound power level of the source, in dB re $10^{-12}$ W, REL $SPL_{D,R}$ is the relative sound pressure level taken from Figure 2.12 for the distance D and room constant R, and $L_{p_{D,R}}$ is the estimated SPL at the distance D in that room. Some manufacturers provide sound power level data for their products.

In Equation 2.17, the correct positive or negative sign for relative SPL should be used. For all distances of practical concern, the sign is negative, so that a subtraction of numbers occurs. For example, suppose a source has a sound power level of 110 dB re $10^{-12}$ W at the 250-Hz octave band, and you want to determine the sound pressure level for an operator distance of 2 m in a room whose R value is 50 $m^2$-Sabin. Figure 2.12 shows REL SPL = -10 dB. Thus, Equation 2.17 would give

$$L_{p_{2,50}} = 110 - 10 = 100 \text{ dB}.$$

Critical Distance--
The derivation of the curves shown in Figure 2.12 is based on material presented in room acoustics sections of most textbooks in acoustics and will not be repeated here. However, there is a useful term that may be obtained from that derivation: *critical distance*, or $D_c$. The critical distance is defined as the distance from a sound source at which the direct sound pressure level from the source approximately equals the reverberant sound pressure level contributed by the room. In its simplest interpretation, if a machine operator must work closer to the machine than this critical distance, sound absorption in the room will not be very helpful, but for distances larger than the critical distance, sound absorption material can be helpful. The equation for $D_c$ is:

$$D_c \doteq 0.14 \sqrt{R}, \tag{2.18}$$

where R is the calculated room constant for the particular frequency band of interest and where both $D_c$ and R are in consistent units. If a room should contain N identical machines, more or less uniformly distributed throughout the room,

$$D_c \doteq 0.14 \sqrt{R/N} . \tag{2.19}$$

The most interesting and unexpected revelation of these two equations is that the critical distance is related almost entirely to the room constant and is not clearly related to the size of the machine. In practice, because some sources have dimensions that are comparable to this critical distance, there may still be some influence of machine size on the actual $D_c$ value.

For the room constant calculated in Equations 2.15 and 2.16, $D_c$ would be about 1.0 m and 2.8 m, respectively, for the bare room and the treated room containing one machine, or about 0.4 m and 1.1 m, respectively, for the bare and treated rooms containing 6 identical machines.

Source Directivity--
Many sources radiate more sound in some directions than in other directions. This radiation can be a point of consideration in studying the position occupied by a machine operator. Where possible and practical, the nearby operator should try to remain in the quieter region of the sound field most of the time. In the reverberant sound field of the source, the region of possibly lower sound levels will be filled in by the higher sound levels, and the source essentially loses its directivity characteristics. The greater the room constant (the more absorptive the space), the greater the distance from the machine before the quieter regions are filled in by the reverberant stronger levels.

In outdoor situations (and in anechoic test chambers), sound sources retain their directivity characteristics, and this retention should be taken into account when orienting directional outdoor sound sources (such as some types of mechanical-draft cooling towers) relative to critical neighbor positions or areas.

Using acceleration measurements--
Accelerometers are sometimes used to assist noise control engineers in identifying noise sources, especially in difficult situations where the sound field under investigation is relatively uniform and where there are many possible noise sources operating simultaneously.

Accelerometers may be used in place of microphones on some of the more sophisticated sound level meters. The meters then serve to amplify and/or filter the accelerometer signal rather than the microphone signal.

When properly secured to a vibrating surface (refer to instruction manuals), accelerometers will produce a signal proportional to the accelerations that surface undergoes as it vibrates back and forth. The acceleration levels (in decibels, as read from the meter) are related to the sound pressure levels on the surface, radiating into air approximately by:

$$SPL_s \cong AL + 150 - 20 \log f \ , \qquad (2.20)$$

where $AL = AL_m - AL_{1g}$, $AL_m$ is the acceleration level as read from the meter, $AL_{1g}$ is the acceleration level as read from the meter when the measuring system is subjected to an acceleration of 1 g, and f is the octave-band or third-octave-band center frequency of the vibration.

Vibration calibrators are available to ascertain the meter reading when the measuring system is subjected to an acceleration of 1 g. The calibration need only be made at a single frequency.

A typical set of octave-band acceleration data and relevant calculations would be as follows (for a system calibrated to read 1 g = 82 dB):

| Frequency (Hz) | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|
| $AL_m$ (dB) | 67 | 84 | 77 | 75 | 62 | 62 | 72 |
| $AL_{1g}$ | 82 | 82 | 82 | 82 | 82 | 82 | 82 |
| AL | -15 | +2 | -5 | -7 | -20 | -20 | -16 |
| 150-20 log f | 108 | 102 | 96 | 90 | 84 | 78 | 72 |
| $SPL_s$ | 93 | 104 | 91 | 83 | 64 | 58 | 56 |

The final line above indicates the octave-band sound pressure levels at the surface of the vibrating structure.

An approximate relationship between the sound power level of the vibrating surface and the calculated sound pressure levels at the surface of the vibrating structure is:

$$PWL = SPL_s + 10 \log A_m , \tag{2.21}$$

where $A_m$ is the area of the vibrating surface in square meters or

$$PWL = SPL_s + 10 \log A_{ft} - 10 , \tag{2.22}$$

where $A_{ft}$ is the area of the vibrating surface in square feet. Thus, in the above example, if the vibrating surface had a surface area of 1 $m^2$, the octave-band PWL of the surface would be equal to the calculated octave-band surface sound pressure levels.

Equation 2.20 assumes that the vibrating surface is an efficient radiator of sound. This assumption is not always true. In fact, small surfaces (small compared to the wavelength of the frequency of sound considered) are very inefficient sound radiators. Also, thin materials do not radiate sound efficiently. These aspects are discussed more fully in the technical references given in the bibliography. The reader should be aware, however, that determinations of the octave-band power level of a vibrating surface by the above procedure may be as much as 25 to 30 dB too high for some thin or small vibrating surfaces.

Notwithstanding the shortcomings of the calculations involved, acceleration data can serve to eliminate from consideration surfaces which might otherwise be suspected of being significant noise sources and can also serve to help pinpoint surfaces which deserve further study.

## SUMMARY OF DIAGNOSTIC APPROACHES

This chapter has introduced many of the fundamentals of sound that are not only essential background information for noise control practitioners, but also serve as steps in the identification and diagnosis of noise sources and components. To recapitulate:

- Turn machines on and off during sound measurements to determine major and minor sources.
- Use decibel addition to supplement the sound measurements in determining quantitatively the relative strength of the various contributors to the total noise.
- Understand and use the A-weighted filter response to emphasize the importance of the sounds that most influence the A-weighted sound levels.
- Make extensive sound measurements at many close-in positions and at all frequencies of concern to permit suitable study of the internal details of the many potential sources. This is necessary because on the basis of wavelength considerations alone, small-size sources (small compared to the wavelength of sound in air for the frequency of interest) cannot be strong low-frequency sound sources, but they can be important high-frequency sources.
- Calculate the approximate sound power levels of various source components to rank-order or diagnose the components in terms of their noise output. This is necessary because frequency analysis (in octave bands or even narrower filters) is essential to a proper study of a multitude of sound sources.
- Take room conditions into account when estimating sound levels for equipment in various spaces.
- Attempt to identify and quantify airborne and structural sources and paths of noise. Different noise control approaches must be used on these two broadly different types of sources.
- Do not ignore your ears as sensitive and useful instruments. Sometimes, certain sound signals may not be differentiated with sound measurement instruments, whereas your ears can pick up and distinguish unusual signal characteristics that can be attributed uniquely to certain sources.

- Where possible and practical, obtain and use a separate small microphone and preamplifier with cable connection to the sound level meter. As the microphone is probed carefully around the working parts of the machine, watch the sound level meter (at A-setting or any specific octave band of interest) and look for peaks indicating that the microphone is close to a sound source. Sometimes, microphone movements of only a few centimeters, when held perhaps 1 cm from a complex mechanism, can reveal important close-in sources that deserve special attention.

- Repeat crucial measurements to guard against errors in readings and to ascertain that the machine is performing consistently.

- Make detailed notes and sketches to augment the noise data. Be as accurate as time will allow.

- Take time to think. Do not leave the job without having some specific thoughts on dominant noise sources and possible treatments. Also, consider possible alternatives to those first thoughts. Later data analysis may reveal errors in the initial ideas.

- Above all, apply thought and ingenuity in planning the measurements, obtaining the data, and analyzing the results. Do not allow yourself to be rushed through an important problem without adequate preparation, study, and analysis.