Table 1 .... Octave Pass Bands

	Frequency, Cycles per Second							
A. Previously Used[a] Bands:								
Frequency Limits Lower	37.5[b]	75	150	300	600	1200	2400	4800
Frequency Limits Upper	75	150	300	600	1200	2400	4800	9600
Approximate Geometric Mid-Frequency	53	106	212	425	850	1700	3400	6900
B. New Preferred[a] Bands:								
Mid-Frequency	63	125	250	500	1000	2000	4000	8000
Approximate Frequency Limits Lower	45	90	180	355	710	1400	2800	5600
Approximate Frequency Limits Upper	90	180	355	710	1400	2800	5600	11200

[a] Since most published data are based on the frequency limits listed under A, these have been used in this chapter. All graphs in this chapter carry mid-frequency scales so that data can be interpolated.
[b] Many octave band analyzers now in use and published data have a lower limit of 20 cps for the first band. This does not significantly affect the design procedure for airborne sound set forth in this chapter, or any of the design data.
[c] See Reference 16.

where W = the sound power in watts.

In this chapter, the reference value for sound pressure is always 0.0002 microbar, and for sound power is always 10^{-13} watt. In the text, these reference values usually are not repeated, in order to simplify the presentation. When data in this chapter is compared with other publications, reference values must be considered, because other values (notably 10^{-12} watt for sound power) have often been used. Earlier editions of the GUIDE AND DATA BOOK used 10^{-12} watt for sound power.

Sound Frequency

The quality of a sound is determined primarily by its frequency spectrum. Unlike a tone, which has a *single* frequency, the *sounds generated by air-conditioning system components* have a *multitude* of frequencies. A plot of sound pressure level or sound power level against frequency is called a frequency spectrum. For engineering calculations, it would be too complicated to deal with the levels at all individual frequencies. Instead, the audible frequency range of approximately 20–10,000 cycles per second is divided into bands, usually one octave wide. The term octave means that the frequency of the upper limit of the pass band is twice that of the lower limit. An octave band sound pressure spectrum is obtained with an octave band analyzer containing electronic filters which pass only those components of the measured sound which have frequencies within the limits of the filter.

The second set of octave band limits shown in Table 1 is presently specified by an ASA standard.[15] New instruments and most new published data can be expected to comply with this set of limits. However, the first set given has been in general use for some time, and the bulk of literature now available will be found with this division by frequency. Data in this chapter are presented so that values are apparent for either set of band limits. Experimental data obtained in one system can usually be interpolated to the other system with sufficient accuracy for all practical design problems.

Attenuation

In the design of sound control for air handling systems it is usually necessary to calculate the extent to which sound is attenuated (reduced) as it travels through a duct system. A certain percentage of the entering sound is lost in each duct element (elbow, bare or lined duct section, sound trap, etc.).

The ratio of the sound power entering the element to the sound power continuing beyond the element, expressed in decibels, is called the *attenuation* of that duct element:

$$\text{Attenuation, in db} = 10 \log_{10} (W_e/W_c) \qquad (4)$$

where

W_e = sound power entering a duct element, watts.
W_c = sound power continuing beyond a duct element, watts.

Fig. 1 may be used to convert the ratio of sound powers in Equation 4 to decibels and vice versa. If the sound power continuing beyond a bare duct section is 25 percent of the entering sound power, the ratio is $100/25 = 4$, and the attenuation is 6 db. If a sound trap absorbs 99.9 percent of the entering sound, the ratio is $100/0.1 = 1000$, and the attenuation is 30 db. Since the attenuation ratio is dimensionless, no reference value is needed.

When the sound power entering a duct element and the power continuing beyond the element are expressed as power levels in decibels, Equation 4 reduces to a simple arithmetic expression:

$$\text{Attenuation, in db} = Lw_e - Lw_c \qquad (5)$$

where

Lw_e = sound power level entering a duct element, decibels.
Lw_c = sound power level continuing, decibels.

The attenuation of a group of elements through which the sound passes *in succession is the arithmetic sum* of all individual attenuation values, in decibels. The maximum net attenuation that can be obtained is limited by noise regeneration due to air flow.

Since attenuation cannot be measured directly, duct elements are generally rated in terms of differences between measured sound pressure levels. Depending on the way in which these measurements are made (see Reference 3), the ratings will differ in magnitude, usefulness and name, although the unit is always the decibel.

Insertion loss (IL) ratings of duct silencers most accurately describe the reduction of fan noise level as measured in an air-conditioned room. Insertion loss data, however, are relatively costly to establish because full sets of readings have to be taken both before and after the insertion of the silencer into a typical duct system. Insertion loss measurements can be made with air flowing through the duct and silencer.

Noise reduction (NR) ratings of duct silencers are merely the difference in the sound pressure levels measured inside the duct at points just ahead of and just after the silencer. Due to *peculiarities of wave propagation*, the effect which will be achieved by installing the silencer in the actual system is usually less than the measured noise reduction. Since induct measurements generally have to be made without air flow, the usefulness of NR measurements is limited.

Transmission loss (TL) ratings are a measure of sound transmission from one room to another, and are generally used to describe the sound isolating value of walls. When this technique is used to rate duct silencers, it is necessary to subtract the TL of the test set-up.

Table 2 .... Combining Two Sound Levels

Difference between Two Levels To Be Combined, db	0	1	2	4	6	9
Number of Decibels To Be Added to Higher Level to Obtain Combined Level	3	2½	2	1½	1	½

Decibel Calculations

Equation 5 shows that calculations involving attenuation are simplified by the decibel notation system, because the various db values are simply added or subtracted. The same holds true for the conversion of sound power levels to sound pressure levels, and vice versa. To further simplify sound control calculations, all data in this chapter are presented directly in decibels.

However, when the decibel system is used, it should be noted that the levels of two or more sounds *cannot* be added arithmetically. Table 2 may be used to combine two sound levels. If more than two levels are to be combined, the two highest levels should be combined, and the resulting value combined with the next highest level. Levels which are more than 10 db below the others in a combination can be neglected.

Example 1: Find the combined level of two sounds having levels of 40 and 34 db respectively.
Solution: The difference between the two levels is 40 − 34 = 6 db. From Table 2, for 6 db difference in levels, it is found that 1 db must be added to the higher level to obtain the combined level of 40 + 1 = 41 db.

ANALYSIS OF THE SOUND CONTROL PROBLEM

Every major piece of air-conditioning equipment is, to some extent, a *source* of sound. The quantity and quality of sound that a source may generate without being considered too noisy depends on two factors:

1. The extent to which the sound is dispersed and attenuated along the *path* from the sound source to the *receiver* (the person hearing the sound).
2. The maximum sound level which the receiver will consider acceptable. For noise radiated to the inside of a building, this is the *design goal*, which is established to suit the requirements of the occupants (see Design Step 1 in later text). For noise radiated to the outside, local ordinances may dictate the design goal.[16,17,18]

In order to meet the established design goal, the system designer may either reduce the source strength, or increase the path attenuation, whichever is more practicable. Due to the logarithmic nature of the decibel, this can again be expressed as a simple arithmetic relationship:

$$\begin{bmatrix}\text{Source}\\ \text{Strength, db}\end{bmatrix} - \begin{bmatrix}\text{Sound Dispersion}\\ \text{and Attenuation}\\ \text{Along Path, db}\end{bmatrix} \leq \begin{bmatrix}\text{Design}\\ \text{Goal, db}\end{bmatrix} \qquad (6)$$

Table 3 lists the major sound sources in an air-conditioning system, together with the sound path from each source to the receiver, and gives suggested methods for sound reduction. It should be noted that in most cases there are several parallel paths, all of which have to be considered for effective sound control.

Sound paths are classified in two categories, (1) airborne and (2) solid borne. Each category must be approached differently because different mechanisms of sound transmission are involved.

1. Along the paths 1, 3, 7, 8, and 9 (Table 3), sound is carried entirely by air from the source to the receiver. Airborne sound transmission, whether through ducts or through open spaces, can be calculated readily. This chapter contains the necessary data and procedures for solving Equation 6 for these paths.
2. Paths 2, 4, 5, 6, and 10, *in Table 3, contain solid partitions* or structural elements through which the sound must pass. In many cases, it may be possible to calculate solid-borne sound transmission, but the procedure[6,7,13] is quite complicated. Design data for these paths in this chapter are, therefore, limited to the isolation of machine vibrations.

To make sure that sound reaching the listener through the structure will not exceed the design goal, the engineer has to use empirical methods, relying mostly on his judgment and past experience. It may be helpful to review the check list of commonly used noise reduction methods in Table 3, and some of the case histories in Reference 19. It cannot be overemphasized that every effort should be made to install the major noise sources as far away from critical areas as possible. This phase of noise control *must* be jointly considered in the advanced planning stage[20] by the architect, the mechanical engineer and, if he is available, the acoustical consultant.

DESIGN PROCEDURE

In order to deal with each of the various noise sources and paths in the simplest manner, the following seven steps should be taken:

1. Determine the design goal for the air-conditioning system sound for each room, according to its use and construction.
2. Select equipment radiating directly into the room that will be quiet enough to meet the design goal.
3. Calculate the amount of duct attenuation required to reduce central system sound to design goals.
4. Design or select sound attenuating duct elements to meet the requirements established in Step 3.
5. Make sure noise radiated by outdoor equipment will not disturb neighbors.
6. Prevent transmission of vibration through the structure.
7. Examine the completed installation and make necessary adjustments.

Steps 2 to 4 are illustrated in the text of this chapter by a typical problem, covered in segments as the material is discussed (Examples 2 to 6). The same problem solution is shown on a sound control work sheet at the end of this chapter. *These work sheets are available from ASHRAE.*

STEP 1—DETERMINATION OF DESIGN GOAL

The degree of quietness required in each type of room in the building should be specified by the architect. On large projects and special buildings, such as music halls, it is usually necessary to obtain the advice of an acoustical consultant.

How to Express Design Goals

In order to be an adequate criterion of the degree of quietness required, the design goal must be expressed in such a manner as to define sound quality as well as magnitude. Of the many methods available for doing this, the following are most widely used: (1) various criterion curves or tables defining maximum permissible sound pressure levels for each frequency band (usually octave or ½ octave band), (2) loudness, in sones, or loudness level, in phons, and (3) the A-scale sound level, in decibels.

Each of these methods will be discussed in some detail.

Design Specifications Based on Frequency Band Levels. Single number systems of evaluation are useful in describing the relative acceptability of sound quality and magnitude in an occupied space. Accordingly, they may be useful for purposes of sound control specification and equipment rating in the case of equipment radiating directly into the occupied space (such as found originating at grilles and diffusers).

Single number systems are not adequate, however, to express the nature of remotely located sound sources. Frequency has such an important effect on all three terms of Equation 6 that this equation must be solved separately for a number of frequency bands not exceeding one octave in width (see Table 1). This procedure is not as time consuming as it may seem because, with some experience, it is usually possible to pick two or three octave bands which could be the critical ones in a given situation.

While a breakdown into octave band levels is sufficient for design specifications and calculations, it is necessary to use narrower band widths when establishing ratings for sound sources producing pure tones.[1] Pure tones are often found in