Application of Manufacturers' Sound Data

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American Society of Heating, Refrigerating
and Air-Conditioning Engineers, Inc.

CHAPTER 2
SOUND POWER LEVEL AND SOUND PRESSURE LEVEL

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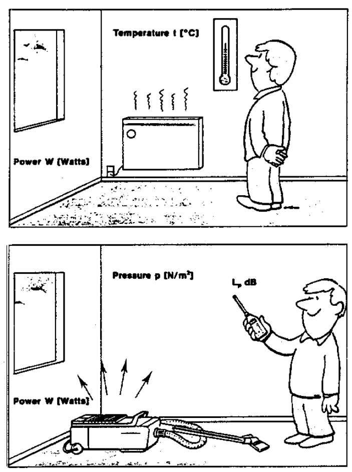

Figure 2-1 Sound analogy with Heat

THE DIFFERENCE BETWEEN SOUND POWER LEVEL (L_W) AND SOUND PRESSURE LEVEL (L_P) AS A BASIS FOR NOISE RATING

SOUND PRESSURE LEVEL (L_P)

What the human ear responds to, and perceives as "sound", is the result of tiny pressure fluctuations above and below the static value of atmospheric pressure. These dynamic pressure fluctuations (force per unit area) cause the eardrum to vibrate, which begins a chain of events in the hearing process that result in signals the brain recognizes as sound. Thus, the quantity of interest, when describing the magnitude of sound perceived by the human ear, is the sound *pressure.*

The range of sound pressures corresponding to hearing sensations extending from very faint to very loud is too huge to measure on a linear scale. A logarithmic or decibel scale is used to compress the range into something manageable. Fortunately, the human ear is *not* a linear device, but responds in a somewhat logarithmic way. Therefore, the use of the decibel scale (dB) is a convenient means for expressing the "level" of a given sound pressure with respect to a reference sound pressure of 2×10^{-5} pascals, corresponding to the threshold of human perception. Thus, the term, Sound Pressure *Level re* 2×10^{-5} pascals, (**L_p**), is used as a descriptor of the sound environment as perceived by the human ear. It is a quantity that depends on the acoustical characteristics of the space, subject to many variables such as room size and geometry, and the amount and placement of acoustical absorption present in the space.

Sound pressure level is the quantity measured with a sound level meter.

SOUND POWER LEVEL (L_W)

A fundamental characteristic of an acoustic source is its ability to radiate sound power, whether weak and small in size (as a cricket) or strong and large (as a compressor). An energy input excites the source, and it radiates this energy in acoustical form. This acoustical power can be expressed in watts. Thus Sound Power is the *rate* at which sound energy is radiated by a sound source. It is a property of the noise source and the magnitude of power radiated by a given source is considered to be essentially independent of the surrounding environment. For this reason, it has become customary in the industry to express the acoustical properties of equipment in terms of the sound power output.[1]

The sound power of a source is determined from either measurements of sound pressure levels or by comparison to a source of known sound power (reference sound source). In some cases sound power of sources are estimated from calculations.

The range of sound power outputs from very quiet to the very loud sources of noise is too huge to handle on a linear scale, so again a logarithmic scale expressed in decibels (dB) is used to compress the range to something manageable. The decibel scale (dB) in this case is used as a means for expressing the "level" of a given sound power with respect to a reference sound power. The reference sound power typically used is 10^{-12} watt. Sound Power *Level*, (**L_w**) re 10^{-12} watt, is the common descriptor used by industry to express the sound power output of a piece of equipment.

THE DIFFERENCES BETWEEN L_W AND L_P ARE FREQUENTLY CONFUSING

The fact that both sound power level, (**L_w**), and sound pressure level, (**L_p**), are expressed in decibels creates much confusion in the engineering design of HVAC systems and the sound specifications of components. What is not understood is that a decibel is simply a means of expressing the magnitude of a specific quantity with respect to a reference quantity when the range of values involved is very large. For example, one could express the number of apples in a bushel as so many dB above a reference value of one apple. Often overlooked is the kind of reference quantity specified. The reference for sound power level is a power expressed in watts; the reference for sound pressure level is a pressure expressed as a force per unit area or pascals.

There is also confusion in applying Lw and Lp. It may help to consider the following analogy between sound and heat. The sound power of a source corresponds to the heat output of a source in BTU/hr. Both depend only upon "source" characteristics. The sound pressure level corresponds to the temperature in the room. Both depend upon space and room properties that vary from space to space. This analogy is often helpful in understanding the difference between L_w and L_p, as illustrated in Figure 2-1.

[1] Strictly speaking, there are special circumstances under which the sound power output of a device, especially at low-frequencies may be modified by the acoustical properties of the test environment. These effects are being investigated at the present time, because of the increasing attention being paid to equipment noise characteristics in the frequency range below 100 Hz.

There is also an analogy between sound and light. For example, the rate of energy consumption of a light bulb is expressed in watts, but the illumination produced in a given room depends upon the room size and the reflectivity of the surfaces. In the case of sound, the rate at which the source delivers energy to the space is expressed in watts, but the sound pressure level that results depends on the room size and the amount of acoustical absorption present.

SOUND PROPAGATION OUTDOORS

Outdoors, there are few reflecting surfaces except for the ground. As a result, the sound pressure, due to the spherical propagation pattern of sound energy, typically decreases inversely as the square of the distance from the source to the receiver[2].

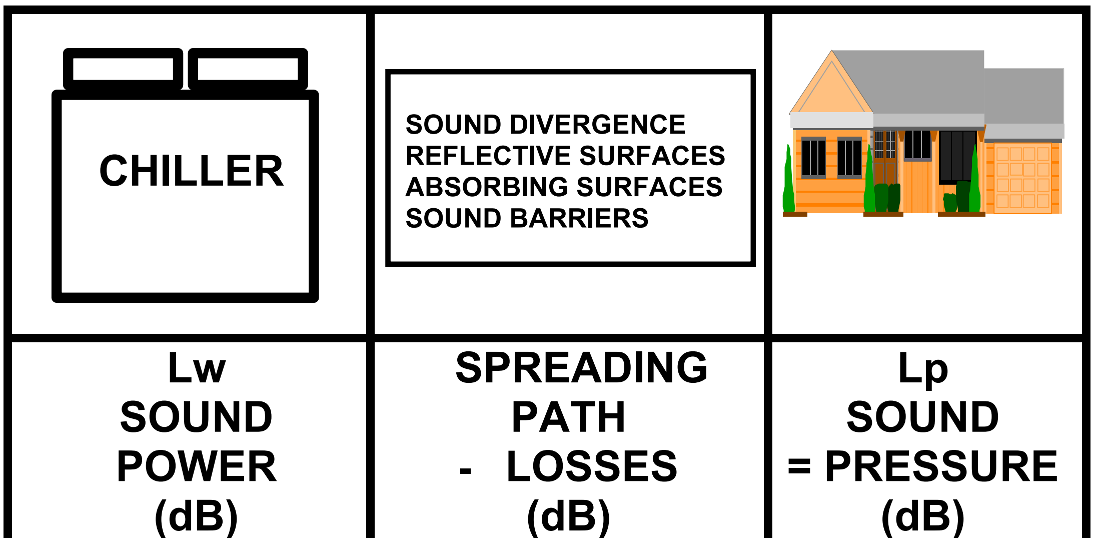

Figure 2-2 Relationship of sound power to sound pressure outdoors

As a result, sound level decreases with increasing distance between the noise source and sound receiver[3].

Figure 2-2 illustrates the relationship between the source sound power level, **Lw**, outdoors and the sound pressure level, **Lp**, at a given distance from the source. The sound pressure level, **Lp**, can be estimated at various locations by subtracting the path losses, such as might be introduced by barrier objects, and the loss due to spherical spreading. Appropriate factors to use are described in the ASHRAE Handbook[4,5,6] and in various industry application standards.

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[2] Beranek, "Noise and Vibration Control," McGraw Hill
[3] Beranek, "Noise and Vibration Control," McGraw Hill
[4] "1995 ASHRAE HVAC Applications Handbook", Chapter 43
[5] "Application Of Sound Rated Outdoor Unitary Equipment", ARI Standard 275-84
[6] "Algorithms for HVAC Acoustics", ASHRAE Publication

FREE AND REVERBERANT FIELDS

Free Field

A free-field is one with no sound reflections. A source of sound located in a free-field environment produces a sound pressure level pattern changing only as a function of distance, due to only direct radiation from the source with no contributions due to reflected energy.

Essentially Free Field

Many times manufacturer's measured Sound Pressure Level data, such as for cooling towers and water cooled chillers and centrifugal pumps, are often obtained under "essentially free-field conditions". A flat open space with minimal obstructions except for the ground, or a very large volume room free of significant reflections is often used to obtain "essentially free field" sound data so that sound pressure levels close to the sound source are not affected by reflections from distant surfaces.

Reverberant Field

A reverberant field environment is one in which the magnitude of the sound pressure level in the space of interest is dominated by reflected sound energy***.*** For example a small mechanical equipment room with hard reflective surfaces is highly reverberant.

SOUND PROPAGATION INDOORS

Figure 2-3 illustrates the indoor relationship between sound power level, **Lw**, and sound pressure level, **Lp**. The estimation of sound pressure levels indoors is usually more complicated than in an outdoor situation, because both the direct and reflected components of the sound energy have to be taken into account. In outdoor situations, usually only the direct sound component has to be estimated at the location of interest.

The sound pressure level indoors (at a specific location) can be estimated by subtracting from the known sound power level, **Lw,** the energy losses that occur in the various sound paths between the sources and receiver. These losses in decibels are described in detail in the ASHRAE Handbook[7] and in various industry application standards[8].

MOST SOURCES ARE RATED IN TERMS OF SOUND POWER LEVEL EXCEPT FOR COOLING TOWERS AND WATER COOLED CHILLERS

While most HVAC sources are rated in terms of sound *power* level, cooling towers and water cooled chillers are exceptions. These are rated in terms of sound *pressure* level measured at a specified distance from the face of the unit in "essentially free field conditions" as previously described in this chapter.

[7] 1995 ASHRAE HVAC Applications Handbook, Chapter 43

[8] "Procedure For Estimating Occupied Space Sound Levels In The Application Of Air Terminals and Air Outlets", ARI Standard 885-90

Both the Air Conditioning and Refrigeration Institute and the Cooling Tower Institute recognize that very large equipment[9] is nearly impossible to rate unless the testing is accomplished in-situ when the equipment is delivering rated tonnage. It is unrealistic to expect that these machines can be moved to a reverberant or anechoic space for testing. Consequently, these units are normally tested outdoors, or in an "essentially free-field" environment indoors.

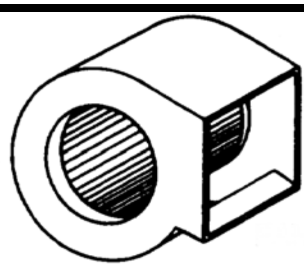
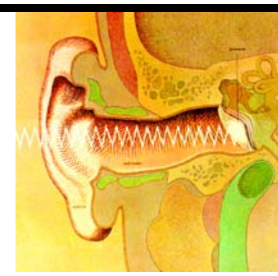
	**DUCT ATTENUATION SOUND ATTENUATORS DUCT BRANCHING DUCT BREAKOUT DUCT END REFLECTION PLENUM CEILING ATTN SPACE EFFECT**	
Lw SOUND POWER (dB)	**PATH - LOSSES (dB)**	**Lp SOUND = PRESSURE (dB)**

Figure 2-3 Relationship of sound power to sound pressure indoors

HOW SOUND PRESSURE LEVELS ARE ESTIMATED FROM SOUND POWER LEVELS (FAN POWERED TERMINAL EXAMPLE)

Figure 2-4 (after ARI Standard 885-90) illustrates the acoustical model of a fan powered air terminal device. This model shows six paths that the sound can take to reach the receiver's ears. Most often, only one or two of the paths are significant contributors, which can simplify the calculations. In most cases, paths 1 and 5 are usually the most significant contributor to the receiver's sound level.

- For Path 1 the **Sound Pressure Level** is calculated by subtracting from the **Casing Radiated Sound Power Level (S),** the **Ceiling-Plenum Attenuation (P)** , and the **Space Effect (S).**
- For Path 5 the **Sound Pressure Level** is calculated by subtracting from **the Discharge Sound Power Level (D),** the **Duct Insertion Losses, Elbow and Tee Losses , Branch Power Division, End Reflection Factor and the Space Effect (S)**.
- These two sound pressure levels are then added logarithmically to estimate the sound pressure level for the octave band of interest.
- For critical applications, the sound pressure level of all paths should be added logarithmically.

[9] Cooling towers are usually tested at 5 ft and at 50 ft. from the source. Water-cooled chillers are tested at 1 meter from the chiller surfaces at height of 1.5 meters above the floor. Cooling towers and water-cooled chillers are rated in terms of sound *pressure* level at specified distances from the source.

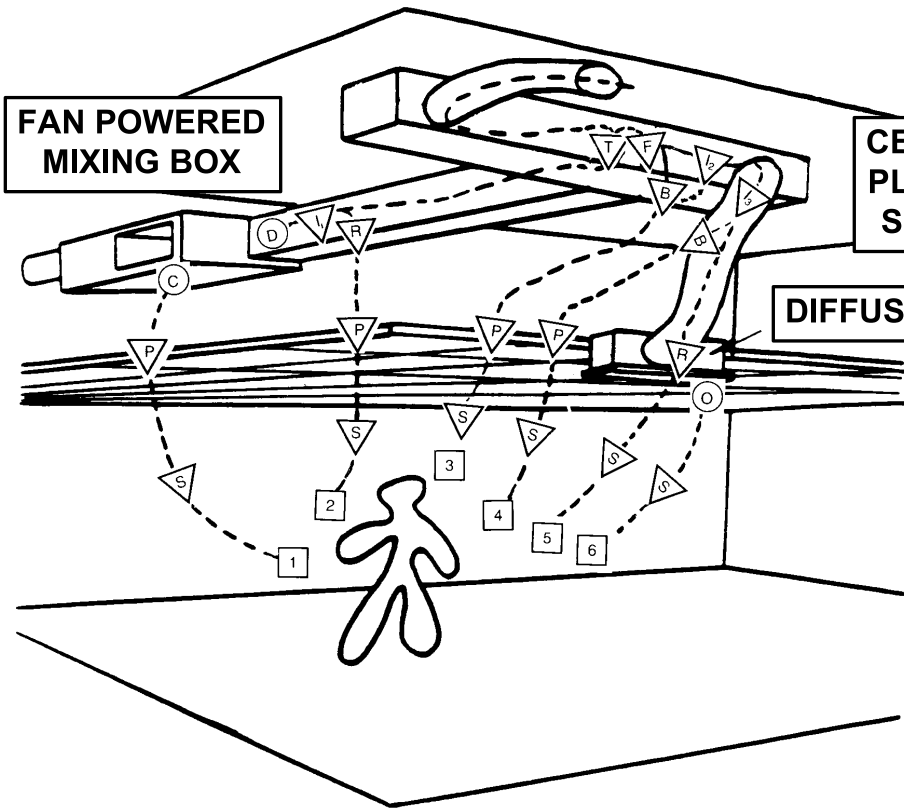


Figure 2-4 Sound paths from noise sources to receiver in occupied space from ARI 880

Path 1- sound breakout from the casing of the unit

Path 2- sound breakout from the discharge duct

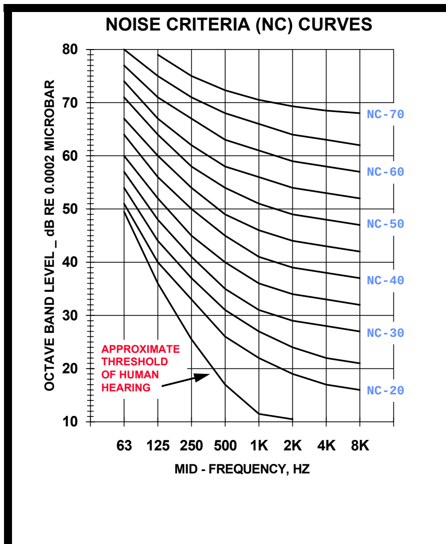
Path 3- sound breakout from follow-on ducting

Path 4- sound breakout from flexible ducting

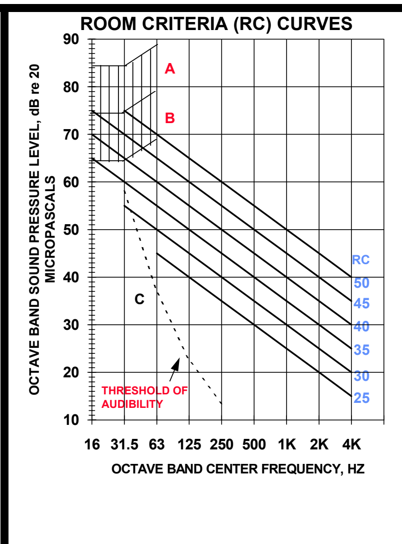
Path 5- discharge sound radiating to the room through the entire duct system, openings, etc.

Path 6- airflow generated noise from the outlet

CHAPTER 3
NOISE RATINGS AND ACCEPTABILITY CRITERIA
Warren Blazier



NC CRITERIA CURVES



RC CRITERIA CURVES

OVERVIEW

Several descriptors are in present use for evaluation of noise from HVAC system operation. The ones most commonly used are:

- A-weighted Sound Level, expressed in dB(A), such as 55 dB(A). This rating is most frequently encountered in *outdoor* applications where compliance with community noise ordinances is a consideration. However, dB(A) is still occasionally used in specifying limits for indoor applications.

- NC Criteria, expressed as a rating number, such as NC-35. NC Criteria are most frequently used for rating HVAC noise in *indoor* applications, where permissible noise limits have been specified on this basis.

- RC Criteria, expressed as a rating number *plus* a subjective quality descriptor, such as RC 35(N). RC Criteria are intended for use in *indoor* applications where the subjective quality of the sound, and not just the level at which it occurs, is to be taken into account.

Although tables of numerical criteria are available that list "acceptable" ratings in terms of one or more of the noise rating methods in common use[1,2] , such tables rarely explain the

[1] 1995 ASHRAE HVAC Applications Handbook, Chapter 43, Table 2.
[2] Cyril M. Harris, "Handbook of Acoustical Measurements and Noise Control" (Third Edition), Table 43.1.

advantages or disadvantages of one descriptor over another for a given type of application. For example, if a recommended noise criterion has been met in a given application it does not necessarily ensure that the occupant will find the environment satisfactory, if the subjective character or quality of the noise is not addressed by the particular rating procedure used. Satisfaction can be a gamble if the specified noise criterion is only *level-sensitive* and does not take into account the subjective quality of the sound as perceived by the ear of the room occupant.

The A-weighted sound level and NC rating procedures, which have been in common use for over 30 years, are only *level-sensitive*. Such ratings do not differentiate between acceptable or unacceptable background sounds as they might be perceived by the human ear. For example, it is not uncommon for several noises having the same dB(A) or NC rating to rank significantly different, when subjectively judged by a listener on a scale of relative *acceptability*. An example of the inability of either the dB(A) or NC rating methods to differentiate between acceptable and unacceptable sounds, on the basis of subjective quality, is illustrated in Fig. 3-1.

In this example, three different noise spectra are shown that have *identical* A-weighted sound levels of 42 dB(A), and also *identical* NC ratings of NC 35. However, the subjective character, or "quality", of each noise spectrum illustrated is significantly different when perceived by the human ear. The spectrum labeled (H) sounds hissy, the spectrum labeled (R) sounds rumbly, while the spectrum labeled (N) is neutral in character -- i.e. no one part of the spectrum dominates the character of the noise. These three spectra each receive a noise rating of NC-35, because the highest *tangent* NC curve in each instance is the NC 35; the (H) spectrum is tangent at 2000 Hz, the (N) spectrum is tangent at 500 Hz, and the (R) spectrum is tangent at 250 Hz. (The NC rating, by convention, is determined by the *highest* NC curve *tangent* to the spectrum, independently of the frequency-band in which it occurs).

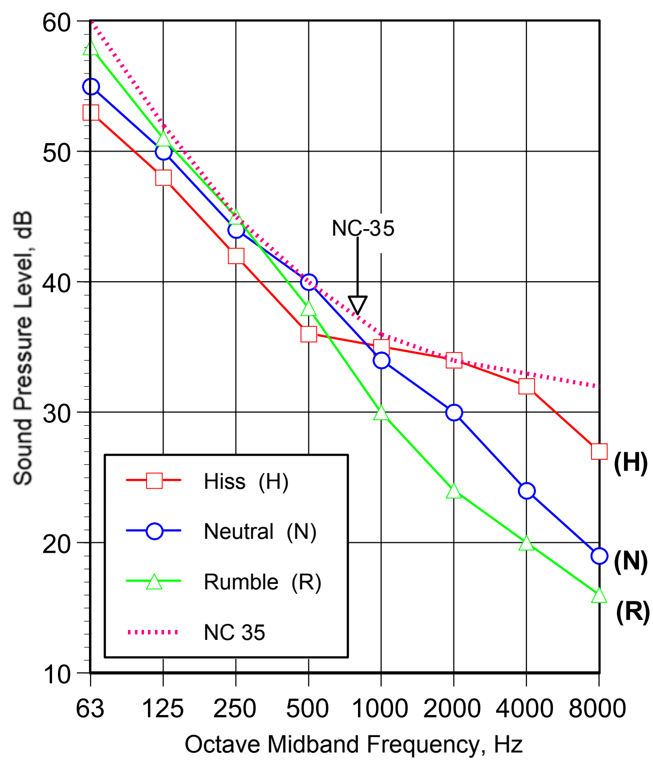


Fig. 3-1: Three noise spectra having identical A-weighted sound levels (42 dBA) and identical NC ratings (NC 35).

On the other hand, the RC noise rating procedure has the advantage of being not only level-sensitive, but also *quality-sensitive*. The RC rating procedure takes into account the subjective character of a noise such as "neutral", "rumble", or "hiss", by identifying these with the letter-descriptors (N), (R) or (H), respectively, appended to the rating number. If the RC procedure had been used to rate the three spectra illustrated in Fig. 3-1 above, the resulting ratings would have been RC-35(H) for the hissy-sounding spectrum, RC-35(N) for the neutral-sounding spectrum, and RC-31(R) for the rumblysounding spectrum, thus identifying the differences in perceived sound quality among the three spectra.

The present industry trend is away from the use of either A-weighted sound levels or NC ratings. ASHRAE presently recommends only the use of the RC procedure for rating HVAC system noise in indoor spaces, especially if the application is sensitive to the *quality* of the sound.

Table 3-1 lists numerical RC(N) criteria, corresponding to various types of space occupancies, consistent with recommendations published in the 1995 ASHRAE Applications Handbook. The “(N)” designates a spectrum having a neutral, inoffensive sound quality. Details of the RC rating procedure, and the guidelines that establish an “N” sound quality attribute, are discussed in a subsequent section of this Chapter.

Room Types	RC(N) Criterion[a,b]
Residences, Apartments, Condominiums:	25-35
Hotels/Motels	
Individual rooms or suites:	25-35
Meeting/banquet rooms:	25-35
Corridors, lobbies:	35-45
Service/support areas:	35-45
Office Buildings	
Executive and private offices:	25-35
Conference rooms:	25-35
Tele-conference rooms:	25 (max)
Open-plan offices:	30-40
Corridors and lobbies:	40-45
Hospitals and Clinics	
Private rooms:	25-35
Wards:	30-40
Operating rooms:	25-35
Corridors & public areas:	30-40
Performing Arts Spaces	
Drama theaters:	25 (max)
Concert and recital halls:[c]	
Music teaching studios:	25 (max)
Music practice rooms:	35 (max)
Laboratories (w/fume-hoods)	
Testing/research, minimal speech communication:	45-55
Research, extensive telephone use, speech communication:	40-50
Group teaching:	35-45
Church, Mosque, Synagogue	
General assembly:	25-35
With critical music programs:[c]	
Schools	
Classrooms up to 750 ft^2 :	40 (max)
Classrooms over 750 ft^2 :	35 (max)
Large lecture rooms, without speech amplification:	35 (max)
Libraries:	30-40
Courtrooms	
Unamplified speech:	25-35
Amplified speech:	30-40
Indoor Stadiums, Gymnasiums	
Gymnasiums and natatoriums:[d]	40-50
Large seating-capacity spaces with speech amplification:[d]	45-55

Table 3-1: Design Guidelines for HVAC System Background Noise in Rooms.

NOTES:

a: The values and ranges are based on judgment and experience, not on quantitative evaluations of human reactions. They represent general limits of acceptability for typical building occupancies. Higher or lower values may be appropriate and should be based on a careful analysis of economics, space usage, and user needs.

b: When the quality of sound in the space is important, specify criteria in terms of RC(N). If the quality of the sound in the space is of secondary concern, the criteria may be specified in terms of NC levels of similar magnitude.

c: An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces .

d: RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.

METHODS OF RATING HVAC SYSTEM NOISE

It is important to make a distinction between ***source*** noise ratings of individual components, and ***system*** noise ratings of the combination of components when measured in the environment of the room occupant. In general, noise criteria are ***space-related*** and predicated upon achieving an acceptable room environment for a given type of occupancy. It is the ***system*** noise rating that must meet the noise criterion specified for a given type of space use.

On the other hand, the noise characteristics of system *components* are commonly expressed in terms of sound *power* level which is *not* a space-related quantity. Therefore, **component** noise ratings can not be directly compared to space-related criteria, without first taking into account the noise reduction which occurs in the *path* between the source location in the system and the environment of the receiver, as discussed in Chapter 2.

A-WEIGHTED SOUND LEVEL: INDOOR APPLICATIONS

The A-weighted sound level is a frequency-weighted sum of the energy contained in the sound pressure level spectrum, expressed in decibel format. This weighting factor approximates the frequency response of the human ear at *moderate* sound levels, and thus tends to rank-order noises on the basis of comparative loudness as perceived by an average individual with normal hearing.

It is important to understand that "loudness" describes only the *magnitude* of a sound, and that different noises may have the *same* A-weighted sound level or relative loudness, but rank differently in the subjective opinion of a listener. This occurs because, in addition to loudness, the ear perceives differences in the frequency structure or "character" of sounds, which subjectively influences the comparative rank-ordering.

Returning to the example of three different spectra having the same A-weighted sound level illustrated previously in Fig. 3-1, a typical listener would probably express a preference for the spectrum labeled (N), because its shape is usually inoffensive to the ear, if not too loud. However, either of the other spectrum shapes labeled (H) and (R) would likely be considered annoying, due to the hissy character in one case, and an excessive rumble in the other. Thus, a loudness-sensitive metric, such as the A-weighted sound level, can not reliably rank-order noises on a subjective scale, *unless* the frequency structure is *nominally similar* and the differences are mainly those of relative loudness. An example of this latter case is vehicular traffic noise, which is commonly rated in terms of A-weighted sound level.

Because HVAC system noises are widely variable with respect to frequency content and structure, the use of A-weighted sound level to specify criteria for *indoor* applications is not recommended, except where dictated by present occupational health and safety noise-exposure limits.

A-WEIGHTED SOUND LEVEL: OUTDOOR APPLICATIONS

Because of the present widespread use of A-weighted sound levels, dB(A), to specify *outdoor* environmental noise standards and regulations, the sound ratings of outdoor HVAC equipment such as cooling towers, air-cooled condensing units, air-cooled chillers, etc. are frequently expressed in terms of A-weighted sound *power* level. However, such ratings must first be converted to A-weighted sound *pressure* level at some specified distance from the equipment, before a comparison can be made with respect to specified environmental regulations. It is important to understand that simply meeting the property line noise limit prescribed by a regulation does not necessarily ensure that complaints in the neighborhood will be avoided, particularly if the *quality* of the noise is tonal in character, or fluctuating in amplitude. These factors are not taken into account in the A-weighted rating.

NC, NOISE CRITERIA CURVES, NC-nn

The NC method of noise rating has been used extensively for over 30 years. The method is based on the use of a family of curves expressed in terms of octave-band sound pressure levels over the frequency range from 63 Hz to 8,000 Hz. Each curve in the family has a numerical designation, for example NC-40, which is keyed to a set of acceptability criteria expressed in terms of an identified space use. In conventional application by the industry, the NC rating is determined by plotting the octave-band spectrum of the noise in question on a family of NC curves, and then assigning a numerical rating according to the highest NC curve that is just *tangent* to the spectrum. Although NC curves were originally defined in 5-NC increments, it has mistakenly become common practice to interpolate between the curves in 1-NC increments. *It is not a requirement of the rating procedure that the noise spectrum follow or approximate the actual shape of an NC curve. This is the principal reason that several noises which have the same NC rating, may rank differently on the basis of subjective sound quality, as illustrated previously in Figure 3-1.*

In application to *constant-volume* HVAC systems, the past use of the NC rating procedure has proven to be relatively satisfactory for predicting or assessing the degree of potential occupant satisfaction. The primary reason is that a majority of the noise problems which occur in *constant-volume* systems are usually diffuser-related and can be identified with excessive noise in the mid- to high-frequency range, thus controlling the magnitude of the NC rating. Problems with excessive noise in this frequency region can often be corrected, and the NC rating reduced to an acceptable value, by refining the system air-balance, or by adding or re-sizing air diffusers.

With the introduction of the *variable-volume* (VAV) systems in the early 1980s, a new class of noise problems emerged which the NC rating procedure fails to detect. Most of these new problems typically result from excessive noise being produced in the frequency spectrum below 63 Hz, caused by a common practice of modulating air delivery by throttling at the fan and at air-control valves in the branch ductwork. The NC curves are not defined below the 63 Hz octave-band, and for this reason they should not be used to rate noises that have a significant low-frequency content in the 16 Hz and 31.5 Hz octave-bands.

Furthermore, an air diffuser selection in *variable-volume* systems tends to be made on the basis of the system operating at *full* air capacity. Thus the diffuser noise levels at the reduced airflow requirements that occur during periods of *normal occupancy* (typically 50-80% of total capacity) may be as much as 15 dB lower than at full-volume design capacity. This *reduction* in mid-to high-frequency diffuser noise is often accompanied by a significant *increase* in low-frequency noise, due to throttling at the fan and air-control terminals to match the changing demand. The overall result in the system as a whole is a low-frequency spectrum imbalance that in many instances the ear perceives as an offensive rumble.

Figure 3-2 illustrates an example of the typical result in a VAV system, when the air volume demand is reduced to 65% of full capacity. The example assumes that an NC-35 noise criterion has been specified at *full* air capacity, and that the system design meets this requirement.

The noise spectrum coded as the heavy *solid* line corresponds to the full-volume flow condition. This is a relatively well-balanced neutral-sounding spectrum that meets the NC-35 design goal, because of its tangency to the NC-35 criterion curve (dotted line) in the 250 Hz octave-band.

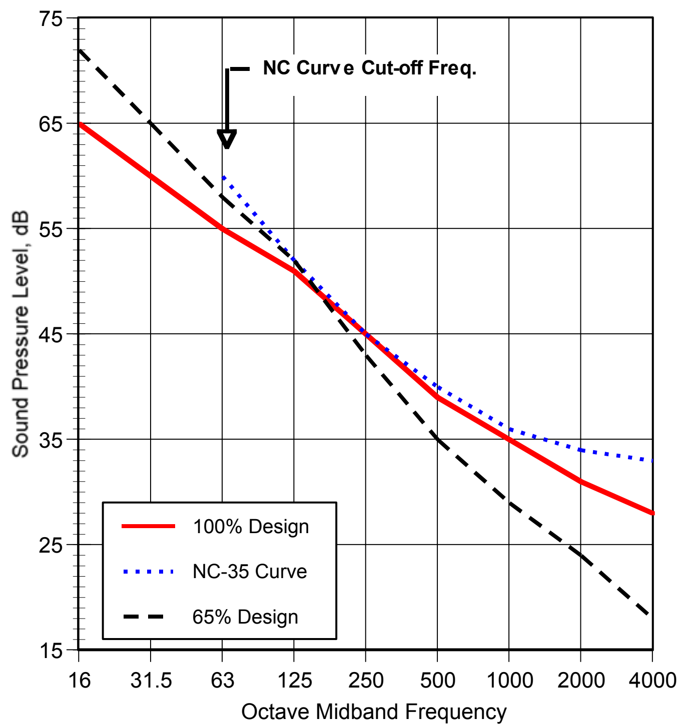


Fig. 3-2: Effect of flow modulation on the spectrum shape and sound quality of HVAC noise in a variable-volume system.

The spectrum shown as a heavy *dashed* line illustrates the changes in shape of the noise spectrum that occurs when the air flow is reduced to 65% of full design capacity in a VAV system. Note that the level of low-frequency noise is *increased*, due to throttling at the fan and air terminal devices, while at the same time the level of mid- to high-frequency noise is *decreased*, because of the reduction in air flow through the diffusers. The combined result is a significantly unbalanced spectrum which is likely to be perceived as an annoying “rumble”, because of the dominant low-frequency content. It is paradoxical that this spectrum *also* would be rated at NC-35, because of its tangency to the NC-35 curve at 125 Hz, and the fact that the rating procedure *ignores* the level of the spectrum below 63 Hz. Thus, a room occupant frustrated by the rumbly quality of the sound environment at this “normal” operating condition might be difficult to convince that the system performance met the specified design goal.

Because the NC rating does not take spectrum shape into account, nor address the frequency range below 63 Hz, it is *not* recommended for the assessment of *variable-volume* HVAC systems, if sound-quality is an important consideration in the system design.

RC, ROOM CRITERIA CURVES, RC-nn( )

The RC procedure for rating HVAC noise corrects several of the shortcomings of the A-weighted sound level and NC rating methods, because the *shape* of the noise spectrum is taken into account in the assessment of sound quality. In addition, the frequency range of evaluation extends down to the 16 Hz octave-band, thus addressing problems associated with excessive low-frequency noise.

The procedure for determining the RC rating of an octave-band noise spectrum provides valuable information for use in estimating the likely acceptability of a given system design. Four steps are required in the procedure:

1) The first step is to plot the spectrum to be rated, and then calculate the arithmetic average of the octave band levels in the 500, 1000, and 2000 Hz octave-bands. This average value becomes the numerical part of the RC rating, which is important in addressing the speech communication or acoustical privacy requirements of the application, which are affected by the sound pressure levels in this frequency region.

2) The second step is to plot a reference curve that has a slope of -5 dB/octave from 16 Hz to 4000 Hz, which passes through the 1000 Hz octave band at the average value determined in the first step. This reference curve represents the *optimum* shape of a "neutral-sounding" spectrum having the same degree of speech communication or acoustical privacy as the spectrum being rated.

3) The third step is to plot the limits above the *reference curve* which cannot be exceeded by the noise spectrum being rated, in order to be classified as a neutral-sounding, subjectively inoffensive sound. The limits are +5 dB, for the 16 Hz through 500 Hz octave-bands, and +3 dB, for the 1000 Hz through 4000 Hz octave-bands.

4) The final step is to note any deviations in the noise spectrum that exceed the level of the reference curve. If the deviations do not exceed 5 dB in the octave-bands from 16 Hz to 500 Hz, nor 3 dB in the octave-bands from 1000 Hz to 4000 Hz, the spectrum is classified as "neutral", and the letter descriptor, (N), is appended to the numerical RC rating obtained in step one. However, if the deviations exceed 5 dB in the lower frequency range, the spectrum is classified as "rumbly" and assigned the letter descriptor "R". Conversely, if the deviations are in excess of 3 dB in the upper frequency range, the spectrum is classified as "hissy" and assigned the letter descriptor "H".[3]

An example using the RC rating procedure is illustrated in Figure 3-3. The spectrum to be rated is shown as the coded heavy solid line. The average of the sound pressure levels in the 500, 1000 and 2000 Hz octave-bands is 35 dB, and this establishes the level of the -5 dB/octave reference curve in the 1000 Hz octave-band (heavy dashed curve). The permissible low-frequency limit above the reference curve of +5 dB (from 16 through 500 Hz) is plotted as the lighter dashed line; the permissible high-frequency limit above the reference curve (1000 through 4000 Hz) of +3 dB is plotted as the dotted line. This spectrum has a rating of RC 35(R), because the levels at 16, 31.5 and 63 Hz exceed the low-frequency limit curve.

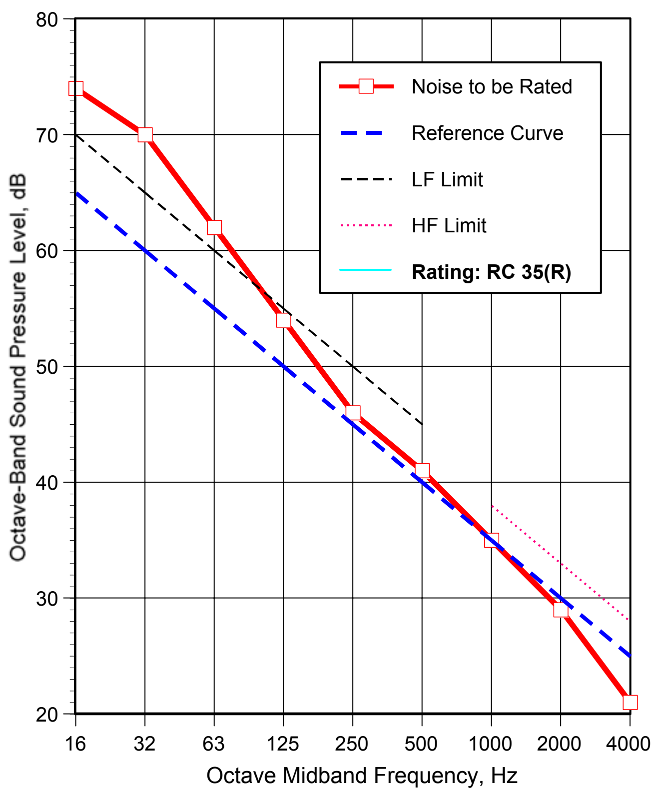


Fig. 3-3: Example of steps to assign an RC rating to a noise spectrum. The spectrum shown above has a rating of RC 35(R). It has a rumbly character, because the low-frequency limit curve is exceeded in the 16, 31.5 & 63 Hz octave bands.

[3] For a more complete description of the RC methodology, including graphic examples, see Chapter 43 of the 1995 ASHRAE Applications Handbook.

With regard to achieving occupant satisfaction, it is obviously desirable to obtain an “N” rating in the assessment of sound quality. Should the spectrum receive an “R” or “H” rating, a potential for occupancy complaints exists. As a general rule, rumble and hiss complaints are likely if the levels of the spectrum exceed the reference curve by more than 5 dB or 3 dB, respectively.

CRITERIA FOR AVOIDING ACOUSTICALLY INDUCED LOW-FREQUENCY VIBRATION

The RC noise evaluation procedure also identifies sound pressure levels in the low-frequency region that are capable of producing acoustically-induced vibration in the walls and ceiling of rooms constructed of lightweight materials, such as wood or steel studs faced with gypsum-board. The octave midband frequencies and corresponding sound pressure levels, where a problem due to acoustically-induced vibration is likely, are listed in Table 3-2. If the noise spectrum being rated has low-frequency sound pressure levels lying within the range shown below, the quality descriptor appended to the numerical RC rating is usually (RV). This indicates that the spectrum may not only be perceived as rumbly, but that acoustically-induced vibration of lightweight room surfaces and furnishings may be a potential problem.

CRITERION	16 Hz	31.5 Hz	63 Hz
MODERATELY NOTICEABLE VIBRATIONS	65 - 75 dB	65 - 75 dB	70 - 80 dB
CLEARLY NOTICEABLE VIBRATIONS	Above 75 dB	Above 75 dB	Above 80 dB

Table 3-2: Criteria for Perceptible Acoustically-Induced Vibration

EFFECT OF SMALL INCREMENTS IN NUMERICAL NOISE RATINGS

Although there may be a contractual requirement that a particular noise criterion or limit not be exceeded in a specified application, it should be recognized that people are not as precise as instruments in detecting differences in noise level. For example, the failure to achieve a specified criterion by 2 or 3 dB is normally inconsequential to a human observer, provided that the specified criterion or limit was appropriate to the application in the first place. However, exceeding specified criteria by 5 dB or more should be viewed as *not* conforming to the spirit of the noise specification.

As a general rule, when subjectively rank-ordering *similar-sounding* noises, an increase in level of about 5 dB is required to obtain reliable evidence that a significant subjective change has occurred. In other words, incremental differences of less than 5 dB in the level of a given noise are not generally sufficient for an average person to detect the change, particularly if the *interval* between comparisons is longer than a few seconds. However, if a prescribed noise criterion is missed by 5 dB or more, it is reasonable to assume that the “spirit” of the specification has not been met and occupant complaints are probable.

The relative significance of a 5 dB incremental change is reflected in nearly all of the criteria for noise assessment which have been developed over the years. For example, both the NC curves and the RC curves, as originally defined, were graduated in 5 dB increments. The same is true of most tables of acceptability criteria and application guidelines which have been published. Unfortunately, it has become common practice by the industry to interpolate in 1 dB increments between these criterion curves, when rating room noise with respect to specified contractual requirements, or in making comparisons among manufacturers published acoustical data. Although this practice is at odds with the way people rank-order differences in noise level, it is nevertheless a fact of life that is not likely to change unless tolerances of at least 2 - 3 dB are incorporated in the specification of acoustical goals.

NOISE RATING OF COMPONENT NOISE SOURCES

As earlier discussed, there are presently no space-acceptability criteria for component noise ratings, since the relative impact on the environment of the receiver depends entirely upon the source noise levels, their location in a system, and the noise attenuation which occurs along the path between source and receiver. Unfortunately, this places the manufacturers of HVAC components, such as air terminal and control devices, in a dilemma when asked to furnish ratings on the acoustical performance of their products. Most manufacturers can supply laboratory test data expressed in terms of **sound *power*** level (a measure of the acoustical energy emitted by the source and analogous to BTUH in thermodynamics). But this is not usually what the typical user is requesting. The question really being asking is, “How much noise is your product going to contribute to the noise level in the environment served by my HVAC system”. Unfortunately, the answer depends upon how the system is laid out, what noise control measures are incorporated in the source to receiver path, and the acoustical characteristics of the receiving space. These are all factors over which a component manufacturer has no control.[4]

FIELD MEASUREMENT OF ROOM SOUND PRESSURE LEVEL

In the commissioning of HVAC systems in buildings, it is frequently required that the achievement of a specified room noise criterion be demonstrated. Measurement procedures are often not specified for obtaining the data necessary to demonstrate this qualification. This leads to considerable confusion when the various interested parties each make measurements using different methodologies. The results often do not agree. The problem is that most *real* rooms encountered in typical field situations exhibit some point-to point variation in sound pressure level. Thus, measurements made at different locations in a room will not usually be the same.

Except where there are audible tonal components in the noise, the differences in measured sound pressure level at several locations in a room are not usually large enough to be significant to the casual observer (± 2 dB). However, when audible tonal components are

[4] Industry associations such as Air Conditioning and Refrigeration Institute (ARI) are presently trying to develop methods of converting component noise ratings expressed in terms of sound power level, to space-related noise criteria, by defining default values for noise reduction in system applications for various configurations of equipment. For specific details refer to the discussion of individual system components in other Chapters of this book.

present, the variations due to standing waves may exceed 5 dB, and these are generally noticeable to the average listener, depending upon their location in the room.

Unfortunately in the commissioning process, where precise limits usually are established as the basis for demonstrating compliance, the outcome can be controversial, unless the measurement procedure to be followed has been specified in detail. At the present time, there is no general agreement within the industry regarding an acoustical measurement procedure for commissioning HVAC systems. However, Air-Conditioning & Refrigeration Institute (ARI) has taken an initial step in this direction by incorporating a "suggested procedure for field verification of NC/RC levels" in ARI Standard 885-90, "Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets".

ARI Standard 885-90 recommends defining the octave-band *room* sound pressure level of the spectrum as the *energy-average* of each octave-band level, measured at four (4) specified points in the space of concern. The recommended locations of these four points is illustrated in Figure 3-4.

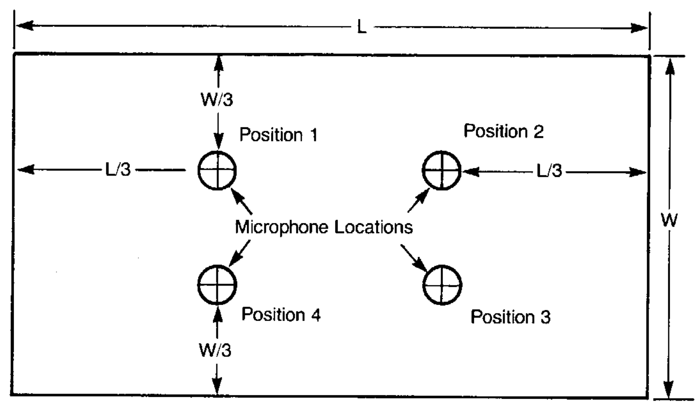


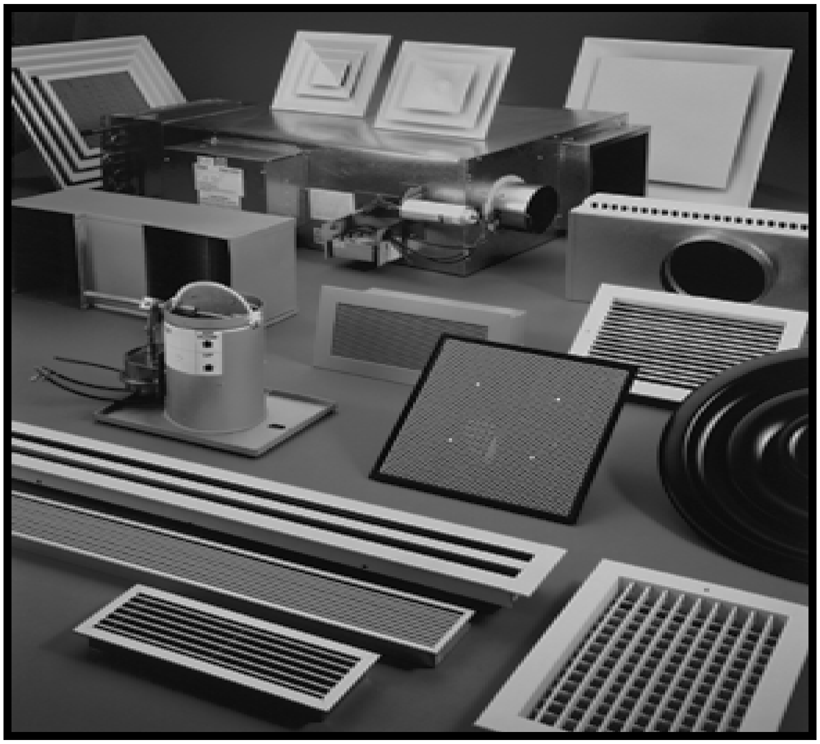
Figure 3-4: Recommended microphone locations for measuring the average room sound pressure level. Each microphone location is 5 ft above the floor.

ARI Standard 885-90 also provides a procedure for processing the data obtained at each measurement location to calculate the energy-averaged level in each octave-band over the frequency range of interest. Thus, the average octave-band spectrum of the noise can be determined for comparison to a specified room noise criterion.

Those directly involved in the specification and commissioning process are encouraged to use ARI 885-90 procedures for most HVAC applications to avoid commissioning controversies.

CHAPTER 10
GRILLES REGISTERS & DIFFUSERS

Ted Carnes
Warren Blazier



OVERVIEW

Grilles registers and diffusers (GRD) are the termination of an HVAC system. These devices are among the most important sources of noise in an HVAC system, because they are located within conditioned spaces and usually in close proximity to the occupants. Grilles, registers and diffusers can be classified as follows:

- A Diffuser is typically mounted flush with a room ceiling and supplies conditioned air that entrains and mixes with room air to temper the thermal environment in the surrounding space.
- A Grille is typically mounted in a sidewall, and discharges air into the space with some means of deflection control, but with less mixing than obtained with diffusers. A Grille may also collect air from the space (a return).
- A Register is a grille with an integral flow balancing damper.

In general, diffusers and grilles will contribute *something* to the sound level in the receiver environment, because of "regenerated" noise due to air turbulence created by elements of the interior design, which control the air discharge pattern and optimize the entrainment and mixing of room air. The magnitude of this contribution depends upon the geometry of the design, the uniformity of the inlet air profile, and the volume flow rate. As a rule, there is a direct relationship between diffuser generated noise and the *effectiveness* of air distribution, in that noisy diffusers usually result in better air mixing than quiet ones. Thus, a compromise is generally required between the magnitude of diffuser-generated noise and the quality of air mixing.

In addition, grilles, registers and diffusers serve as a conduit between the air distribution system and the room, by transmitting noise already present in the connected ductwork. For this reason, when sound rating these elements of the duct system it is necessary to utilize a source of “quiet” air to avoid penalizing the device for noise unrelated to the product.

ACOUSTICAL CHARACTERISTICS

Typical grilles, registers and diffusers (GRD) generate noise principally in the octave mid-band frequency range between 500 and 4000 Hz. This range also corresponds to the major portion of the frequency region associated with human speech. Thus, GRD-generated noise can degrade the quality of speech communication by “masking”, if permitted to exceed a certain threshold depending upon the vocal effort used by the speakers (normal vs. raised speaking voice, shouting, etc.).

On the other hand, this speech-masking effect sometimes can be beneficial in open-plan office layouts, where a certain degree of conversational-privacy between speakers is usually desirable. However, it must be emphasized that GRD-generated noise should only be used for this purpose in constant-volume air distribution systems, where the “masking” sound can be maintained at a constant, predetermined level. It should not be counted upon in variable air-volume system applications, because the level of GRD-generated noise varies as approximately the fifth-power of the flow volume, and thus the speech-masking capability rapidly diminishes as flow-volume is reduced.

MANUFACTURERS’ SOUND TESTING, RATING PROCEDURES AND PRACTICES

SOUND TEST PROCEDURE

At present, Grilles and Diffusers are customarily sound-tested in accordance with ASHRAE Standard 70-1991 (“*Method of testing for rating the performance of air outlets and inlets*”). This Standard covers both aerodynamic and acoustical measurements. ASHRAE Standard 70-1991 references ANSI S12.31-80 (“*Precision Methods for the Determination of Sound Power Levels of Broad-band Noise Sources in Reverberant Rooms*.”) for acoustical measurements, using the reference-source comparison method. (See Chapter 17 for a description of the substitution/comparison method for determining sound power level). ASHRAE Standard 70-1991 also specifies the mounting configuration and installation set-up for testing the various types of products covered by the standard. Typically, data is obtained at the octave mid-band frequencies from 125 through 8000 Hz.

However, a significant observation of the procedure is that the specified air supply to the test unit is from a long, straight run of duct that ensures a uniform inlet velocity profile, which is seldom duplicated in a typical field installation of the product. Thus, the test configuration excludes any inlet system-effects, and the laboratory test data usually reflect a quieter acoustical performance than will be found in a field application, where the inlet supply duct may be close-coupled and frequently offset from the centerline of the unit.

SOUND RATING PROCEDURES AND CERTIFICATION

There is presently no recognized industry procedure for sound rating Grilles, Registers and Diffusers. *(The earlier Air Diffusion Council (ADC) 1062 series of testing/rating standards, and companion certification program, have been inactive since 1985, and should not be referenced in specifications*).

The cognizant trade association is now the Air-Conditioning and Refrigeration Institute (ARI). This organization is currently developing a new rating program based on ARI Standard 890-94 ("*Rating of Air Diffusers and Air Diffuser Assemblies*"), but it has not been implemented at this point in time. ARI Standard 890 references ASHRAE Standard 70 as the method of test.

PRESENT MANUFACTURERS' SOUND RATING PRACTICES; THE 10 DB "ROOM - FACTOR"

In the absence of an approved ARI sound rating procedure (see above), most manufacturers' are not rating the sound performance of diffusers. There are several potential flaws in the earlier ADC methodology that can lead to significant differences between data obtained with earlier test procedures and that obtained under the more stringent ASHRAE requirements. Data should be used which has been collected using the most current testing procedures.

Raw test data on the acoustical performance of a GRD device is typically expressed in terms of octave-band sound *power* levels. Thus, it is necessary to convert the data to sound *pressure* levels before comparison to a particular noise criterion specified for the application in question. (See Chapter 2 for a discussion of the relationship between sound *power* level and sound *pressure* level).

An appropriate conversion from sound power to sound pressure requires knowledge of the acoustical properties of the receiving space, including the room size and acoustical absorption present, the number of diffusers serving the space, and the distance from the source(s) to the specified receiver location. Because these factors are highly variable from application to application, in order to simplify the publication of cataloged sound rating data most manufactures assume a "default" value for the conversion factor. Presently, most manufacturers have adopted a default value of 10 dB, *independent of frequency,* which they then use to calculate a diffuser NC (Noise Criteria, see chapter 3). However, the use of this 10 dB "room factor" can lead to prediction errors in an application, because it does not adequately take into account the effects of room size and the number of diffusers serving the receiving room space.

Note that for most diffusers and grilles, the NC value is approximately equivalent to an RC (N) of the same numerical value; however, use of the RC sound descriptor is really not appropriate for sound rating of individual components of the complete HVAC air distribution system. For example, the combination of a terminal unit with a RC (R) individual discharge sound rating and a diffuser with a RC (H) individual sound rating will often provide a RC (N) overall rating for a balanced sound spectrum in the receiving room.

For example, if a single diffuser serves the receiving room, a conversion factor of 10 dB between source sound *power* level and room sound *pressure* level would be appropriate for a room with a 9 ft ceiling height and floor area of approximately 550 ft^2, based on ASHRAE-sponsored research by Schultz[1].

However, this corresponding room size (20'x25') is larger than most private office applications for a single diffuser, and the use of a 10 dB conversion factor in many instances may result in an *under-prediction* of room sound level by several NC points. Table 10-1 lists appropriate values of the conversion factor for private office applications as a function of room size[2]. (Note that in cases where *two* diffusers might be used to serve a private office as large as 300 ft^2, the conversion factor used for diffuser selection is 3 dB less than if a single diffuser is used. This occurs because each diffuser contributes one-half the permissible *total* sound power delivered to the receiving space).

Approximate Room Size	Number of Diffusers	Lw - Lp @ 500 Hz (dB)	Lw - Lp @ 1000 Hz (dB)
125 ft^2	1	6	7
300 ft^2	1	8	9
300 ft^2	2	5	6

Table 10-1 Conversion factor for private offices with a 9 ft ceiling height.

On the other hand, whenever four or more diffusers are used in a distributed ceiling array to serve large areas such as open-plan offices, a modification of the Schultz single-source equation is necessary to take into account the presence of multiple sound sources in a common plane. The appropriate equation applicable to a distributed array of ceiling diffusers may be found in Appendix B, Section B1.7, of ARI Standard 885-90. Table 10-2 illustrates appropriate values of the conversion factor using this equation that correspond to a room with a 9 ft ceiling height, where the variables are the array spacing and floor area served by each diffuser. As in the previous example, values are shown for the 500 Hz and 1000 Hz octave bands, because this is the frequency region that usually establishes the value of the "NC" (or perhaps future "RC") sound rating typically specified by the user.

Array Spacing ft	Floor Area/Diffuser ft^2	Lw - Lp @ 500 Hz	Lw - Lp @ 1000 Hz
10 x 10	100	4	5
12 x 12	150	5	6
15 x 15	250	6	7
20 x 20	400	7	8

Table 10-2 Conversion factor for open-plan offices with a 9 ft ceiling height (four or more diffusers).

[1] Schultz, T. J., "Relationship between sound power level and sound pressure level in dwellings and offices" ASHRAE Transactions 91 (1).
[2] Conversion factors for the 500 Hz and 1000 Hz octave bands are illustrated, because it is in this frequency region that the "NC" rating of a GRD noise spectrum will be usually determined for most diffuser types.

Few, if any, manufacturers of GRD's publish octave-band sound *power* level data in their catalogs at the present time. Publishing pertinent octave-band data would significantly increase the size of a diffuser catalog, but this may become less of a problem as PC-based electronic catalogs become available. However, such information may be presently available, if requested or specified. Most manufacturers catalog data will continue to be expressed as "NC" ratings based on a 10 dB conversion factor until an industry standard ultimately is adopted for sound rating GRD products.

BALANCING DAMPERS

Discharge noise from diffusers can be minimized if the supply duct pressure is maintained as low as feasible. Balancing dampers, such as those illustrated in Figure 10-1, are used to counter excessive duct pressures, but can generate significant noise due to the turbulence created by the pressure reduction necessary to achieve design air volume at the diffuser terminals. This can result in increased noise of 10 NC points, or more, above the normal diffuser NC rating, particularly if the dampers are located in either the diffuser inlet or within 1-2 duct diameters upstream, and are throttled significantly.

Pressure reducing valves (PRV's) and balancing dampers located at branch take-offs can be effective in achieving design air volume, but only if there is sufficient attenuation from lined ductwork installed ahead of the diffuser terminals to attenuate the damper-generated noise.

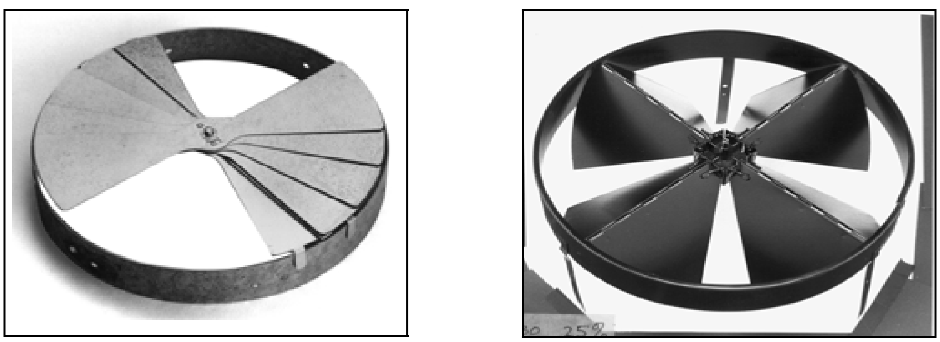


Figure 10-1 Typical Balancing Dampers

Even when fully open, these devices can add 5 NC points to a diffusers sound output, if located in the diffuser such as for a register. As the balancing damper is closed down, the noise level increase is usually proportional to the cube of the ratio of the diffuser's rated total pressure drop to total pressure in the supply ductwork upstream of the damper. This relationship is illustrated in Table 10-3.

Ratio of : Total Pressure in Duct/ Diffuser Pressure Drop	**1**	**1.5**	**2**	**4**
Increase in NC points	**0**	**4 - 5**	**8**	**16**

Table 10-3 Expected increase of close coupled balancing dampers with damper throttling.

EFFECT OF NON-UNIFORM INLET VELOCITY PROFILE

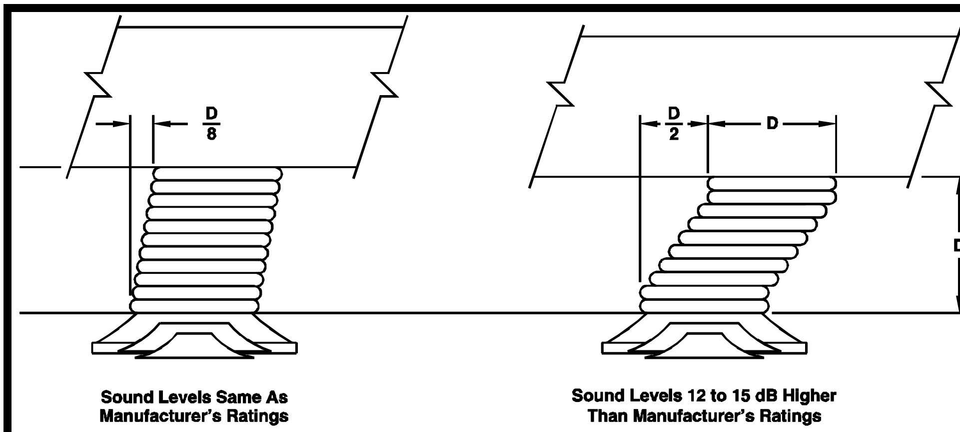


Figure 10-2 Effect of entering velocity profile.

Ceiling mounted diffusers, with the limited plenum space available in many installations, can be severely affected by the inlet velocity profile as illustrated in Figure 10-2. When a flexible connecting duct is offset, or makes a sharp turn at the inlet to a diffuser, both a significant increase in diffuser noise and a poor air distribution profile can result. Where possible, the installation such as shown in Figure 10-2 should be avoided altogether, as the short length of flexible duct cannot attenuate noise generated by the inlet connection or inlet damper. If the supply duct velocity exceeds 500 fpm, a 45-degree entry or conical tap type connection should be used. (Extractors, which protrude into the main air stream, are not normally recommended but may be considered for special applications.) A minimum of 5 ft of non-metallic flexible duct should be employed. This will allow smooth radius bends and straight sections before the terminating GRD. See one of the many available references on good duct designs[3].

For right angle connections, inlet plenums with turning vanes and straightening grids are available, and effective (but are seldom specified or installed). Poor inlet conditions can add up to 15 NC points to the rated sound level of a properly selected diffuser. An average effect for a 90 degree inlet connection is 3 to 5 NC points.

The effect of poor inlet conditions depends partially on the type of diffuser installed. For example, perforated diffusers with face mounted deflectors are very sensitive to velocity pattern variations created by inlet flow profiles. Devices with deflectors in the neck are much less sensitive. As a rule, the diffusers with the lowest reported pressure drop for a given flow rate are the most sensitive to inlet flow profiles.

UNCERTAINTIES IN THE APPLICATION OF MANUFACTURERS' DATA

The possible uncertainties in the application of manufacturers' data for grilles, registers, and diffusers are provided in Table 10-4 below. These are based upon utilized test standards and results found in testing programs.

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[3] One such reference is HVAC System Duct Design, SMACNA, Chantilly, VA, 1990 (Third Printing November 1993). See Chapter 10 "Provisions for Testing, Adjustment, and Balancing" and Chapter 11 "Noise Control".

GRILLES, REGISTERS AND DIFFUSERS						
OCTAVE MID-BAND FREQUENCY	MEASUREMENT UNCERTAINTY OF STD. TEST PROCEDURE[4] dB	REPEATABILITY OF TEST DATA[5] dB	PRODUCTION VARIABILITY[6] dB"	CUMULATIVE UNCERTAINTY DUE TO ALL PREVIOUS FACTORS [7] dB	VARIABILITY OF RANDOM SAMPLE RE. CATALOG[8] dB	BANDS MOST AFFECTED BY SYSTEM EFFECTS[9]
63	*	*	*	*	*	*
125	± 3 dB	± 1 dB	± 1 dB	± 3 dB	± 1 dB	
250	± 2 dB	± 1 dB	± 2 dB	± 3 dB	± 2 dB	5 dB
500-4000	± 1.5 dB	± 1 dB	± 2 dB	± 3 dB	± 2 dB	+5 to 1 5 dB
8000	± 3 dB	± 1 dB	± 3 dB	± 4 dB	± 3 dB	

Table 10-4: Estimated variability of sound power data and bands most affected by system effects. Note: Sound power data is seldom available, as manufacturers typically list NC values only. For most diffusers, NC is set by sound power in the 500-4000 Hz range.

RECOMMENDED SELECTION AND INSTALLATION PRACTICES

SELECTION

Although an NC 35 criterion is frequently specified as an acoustical design goal in the selection of GRD devices for many building applications, in open-plan office layouts a somewhat higher level of background sound (around NC 40) is occasionally specified to enhance acoustical privacy by masking the intelligibility of conversational speech between occupants outside the local area of the occupant listener.

In fact, the predominant sound produced by typical air diffusers lies within the speech-frequency range and is frequently considered as an option to the alternative introduction of masking noise by electronic means.

However, it must be understood that the level and spectrum shape of the masking background sound is relatively critical and only constant-volume air distribution systems can conceivably be used for such an application. If the diffuser sound is too low, the degree of masking is inadequate; if too high, or of an inappropriate spectrum shape, the result may be annoying and objected to by the occupant.

The over sizing of diffusers, while providing a quiet installation, often results in a poor air distribution pattern, due to "dumping" of air leaving the diffuser and poor room temperature control. It is far better to reduce the size and capacity of the diffusers and increase their number to obtain an acceptably quiet installation.

[4] Based on ANSI S12-31 for pure tones and assumes that the expected range is 1 Standard Deviation.
[5] Repetitive testing of same unit in same laboratory.
[6] Subject to manufacturing tolerances permitted in Quality Control.
[7] Based on the square-root of the sum of the squares of all uncertainties.
[8] Catalog data for diffusers is based on production samples of common sizes interpolated for all sizes.
[9] See pages 10-5 through 10-6 for a discussion of system effects.

INSTALLATION

Duct connections to grilles and diffusers should have no tight bends at the immediate approach to the inlet. (See Figure 10-2 and the discussion above for guidance along with footnote 3 and also manufacturers' installation guidelines and instructions). If balancing dampers are required, they should be located at some distance from the supply outlet, preferably close to a branch take-off, and followed by at least 6 feet of acoustically-lined or non-metallic insulated flexible duct to control the transmission of damper-generated noise.

FLEXIBLE DUCTS

Flexible (flex) ducts provide some "attenuation" of upstream system noise, by leaking sound into the surrounding ceiling plenum due to their relatively low wall sound transmission loss and also possibly acoustical impedance mismatch effects which would reflect some sound energy back toward the sound source. Because the ceiling beneath the plenum also provides some sound attenuation between the plenum and the room below, the upstream system noise transmitted to the room from the grille or diffuser outlet is reduced. However, flexible ducts, because of their "rough" interior surface, can also become significant noise generators, if flow velocities greater than about 700 fpm are utilized. Thus, in some cases the attenuation provided by flexible ducts can be offset by the regenerated self-noise, if the duct flow velocities are excessive. Flex ducts should not exceed 8' in length and should be free of sharp bends.

VIBRATION ISOLATION

Grilles, registers and diffusers are "passive" mechanical devices which require no external vibration isolation. Noticeable vibration and rattling of them are usually a product of excessive air turbulence from upstream of the air outlet devices. Both air outlet and inlet devices are susceptible to structureborne vibration coupled from improperly vibration isolated rotating mechanical equipment and ductwork, or from piping with excessive turbulent flow and mechanical vibration attached rigidly to building structural systems such as the ceiling grid.

SOME PRACTICAL NOISE CONTROL OPTIONS

- Reduce the air volume per diffuser by adding additional diffusers. (Down sizing of the diffusers may be necessary, if the discharge flow pattern is adversely affected).
- Install a larger diffuser (over sizing diffusers, however, may reduce air distribution performance).
- Change to a quieter diffuser design.
- Move balancing dampers from the diffuser inlet back to the branch duct take-off point.
- Install rigid rather than flexible duct at the connection to the diffuser, and use turning vanes to ensure uniform flow into the diffuser neck. (Be sure that the flexible duct is not, however, required to provide attenuation of upstream noise sources.) If flexible duct is used, keep the inlet to the diffuser free of sharp bends and maintain low inlet velocities.

SUGGESTED ACOUSTICAL SPECIFICATIONS FOR GRD'S

Sound data on all GRD devices shall be determined in accordance with ASHRAE Standard 70-1991. The octave-band sound *power* levels obtained shall be made available upon request. If the manufacturer's simplified catalog data are expressed in NC points, then the magnitude of the acoustical "room-factor" used in the determination of NC shall be stated.

The GRD selection shall be based on the specified NC room criterion adjusted for the difference between the stated assumed room attenuation and that expected at the location of the diffuser, plus expected installation effects.

Note that it is suggested to <u>not</u> use RC for sound rating only the GRD components alone. See discussion beginning on Page 10-3.

For example, if the specified room criterion is a NC 35, and the manufacturer has used a 10 dB room-factor in cataloging the data, then the following selection guide should be used:

- For a room with 325 sq. ft per diffuser (resulting in a '7 dB room')
- With a typical inlet of flexible duct (which adds approximately 5 NC points)
- Select a GRD sound rated at NC 27 as shown below.

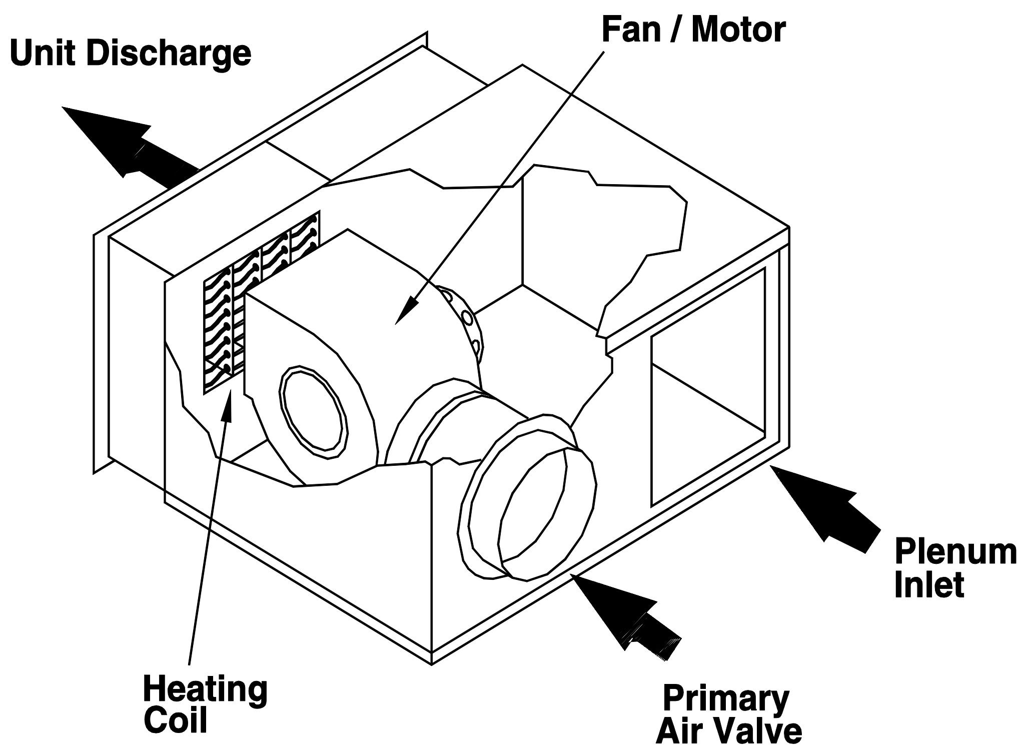
NC 35 (Room requirement);

- 3 NC points (Correction for Room Size) and

- 5 NC points (Inlet Condition Correction).

Results in a NC 27 Maximum Allowable Sound Rating for Selection.

CHAPTER 11
AIR TERMINALS
Ted Carnes
Warren Blazier



OVERVIEW

Within the context of this Chapter, the term “Air Terminals” refers to a factory made damper assembly consisting of supply inlet duct connection(s), outlet duct connection(s) or integral air outlets, unit section(s) and fan assembly (when present). Air terminals provide control for individual zones (individual temperature control areas) of conditioned primary air through the use of automatic or manual dampers, valves, heating or cooling coils, nozzles and fan assemblies.

In general there are two categories of air terminals; *constant air volume* (CAV), and *variable air volume* (VAV). Both types have essentially the same components; a flow-rate regulating device, (typically a damper, although it may be a bellows or other mechanism), and some method of control. In the simplest of units, the process of controlling the air flow-rate generates sound that is proportional to the static-pressure drop across the damper mechanism.

In the more complex units that contain induction nozzles or fans, additional sound is generated. Thus, the relationships between the inlet duct static-pressure, the primary-air flow-rate across the damper, and the secondary-air flow-rate induced by nozzles or an integral fan, determine the sound generated by the unit and transmitted to the receiving space.

VAV air terminals are among the most acoustically critical of all HVAC components, because they are frequently located directly above occupied spaces. Thus, these units may make a significant contribution to the background sound level in the receiving space, particularly at octave mid-band frequencies from 125 Hz to 2000 Hz. In a RC type sound analysis (see chapter 3), the additional sound produced by the air outlets, such as grilles, registers and diffusers, must also be taken into account. (The acoustical characteristics of these latter components are discussed in Chapter 10).

NOISE CHARACTERISTICS OF AIR TERMINALS

The basic function of an air terminal device is to reduce the *total* air pressure available in the inlet duct to the pressure required to deliver the specified flow volume through the downstream system resistance. Since these devices perform this function by the introduction of an obstruction in the airflow path, sound is generated by the aerodynamic turbulence created in the process. In most cases, the sound power level, L_w, produced by the device will be proportional to both the airflow rate and the static-pressure drop across the internal damper assembly. In practice, this means that the sound power level, *in dB per octave band*, is a logarithmic function of the static-pressure drop and the flow-rate across the damper[1].

MANUFACTURERS SOUND RATING PROCEDURES AND PRACTICES FOR AIR TERMINALS

Most manufacturers make data available in two forms for these devices: Full octave-band sound *power* levels at mid-band frequencies from 125 Hz through 4000 Hz, and single-number "Application Data", presented in the form of an "NC" sound *pressure* level rating. (Refer to Chapter 3 for a discussion of how NC ratings are obtained).

The available data is representative of the device *only*, and does not include any external system effects. Data is available over a range of stable operating conditions, for both discharge sound (transmitted down the duct) and radiated sound (through either the unit casing or out of the induction port, depending upon the design).

APPLICABLE ACOUSTICAL TEST AND APPLICATION STANDARDS

- ANSI S12.31 -1980 "*Precision Methods for the Determination of Sound Power Levels of Broad-band Noise Sources in Reverberant Rooms*."
- ANSI S12.32 -1980 "*Precision Methods for the Determination of Sound Power Levels of Narrow-band Noise Sources in Reverberant Rooms*."

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[1] This can be expressed mathematically as follows: $L_W \approx C_0 + K_1 Log(P) + K_2 Log(Q), dB$

Where: P = static-pressure drop across the unit
Q = flow-rate across the damper
C_0, K_1 and K_2 are constants determined by test, specific to each octave band and specific to a given unit design

- ARI 880-94 "*Industry Standard for Air Terminals*" (and ASHRAE Standard 130P "*Method of Testing for Rating of Ducted Air Terminal Units*").
- ARI 885-90 "*Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets* ".

SOUND MEASUREMENT

Broad-band noise data must be obtained in a reverberant room meeting ANSI Standard S12.31, in order to qualify for testing in accordance with the ARI Standard 880-94 rating requirements. However, If the measurement of narrow-band tonal components is required, such as sometimes produced by fan motors, then the test facility must be qualified in accordance with the ANSI Standard S12.32 procedure.

For most product types, the specified operating point of the test unit for determining the ARI Standard rating is 1.5" inlet static pressure at 2000 fpm nominal inlet velocity. Series fan terminals, however, are tested at both maximum rated fan cfm and at reduced fan cfm corresponding to a velocity of 2000 fpm at the primary air inlet.

MEASUREMENT FACILITIES

Ratings based on octave-band sound *power* levels are determined in a reverberant-room test facility using the reference-source comparison method discussed in Chapter 17. However, a number of manufacturers have also constructed non-standardized mockup demonstration facilities for measuring the room sound *pressure* levels of a unit in a typical field installation. These rooms, which are also used for 'in-situ' sound evaluations by listeners, usually have a suspended acoustical ceiling to permit installation in typical plenum configurations, and simulate a normal office space. *There are no standards covering these types of facilities*.

FREQUENCY RANGE OF MEASUREMENTS

The range of normal concern for most air terminal devices is the 125 Hz through 4000 Hz octave mid-band frequencies. The 63 Hz octave-band is seldom reported, because many manufacturers of these products have reverberation chambers too small to qualify for measurements in this frequency range. However, limited data available from test facilities that *can* qualify indicates that the level in the 63 Hz octave-band is usually less than that in the adjacent 125 Hz octave-band. Thus, the absence of 63 Hz octave-band data for these products is not likely to affect the magnitude of the sound rating.

TEST SET- UP AND OPERATING CONDITIONS

ARI Standard 880-94 specifies the installation details for sound testing air terminal devices in a reverberant room. Discharge sound power levels are determined by locating the unit outside the test room and connecting it to the reverberant test chamber using a specified length of acoustically lagged 20 Ga. duct installed flush with the inside wall of the room. The specified length of duct is 2.5 equivalent hydraulic diameters, typically lagged with gypsum board.

Radiated sound is measured with the device suspended in the central area of the reverberant test room, and the connected inlet and discharge ductwork is heavy gauge sheet metal, lagged with gypsum board or other enclosure materials, and ducted outside the test room. However, if the unit has an air induction inlet, this is left exposed within the test room.

ARI Standard 880-94 specifies several pressure and air flows for rating the unit. The ARI Certification Program entails annual check tests on randomly selected samples of participating manufacturer's products and the publication of a directory of performance for 'certified units'. At present, the ratings are for both discharge and radiated sound at the single operating point of 1.5 in w.g. inlet static-pressure and nominal 2000 FPM inlet velocity for most product types, but include fan-only conditions for fan-powered terminals.

Many manufacturers also catalog, or make available upon request, octave-band sound power level data at operating points other than the one used for ARI certification. These points are usually derived by testing at a sufficient number of different operating points to develop a reliable estimation procedure based upon a multiple variable regression analysis of the test data.

TESTING IN A MOCK-UP INSTALLATION

Some acoustical consultants, engineers and developers require full-scale mock-up testing, sometimes referred to as an 'in-situ' test. In most cases, the test space is configured in a realistic manner, incorporating the proposed ceiling system, a typical plenum depth (36" deep, unbounded at the sides above the room walls), and a representative room size. So many tests have been conducted in rooms with a nominal 2400 cu ft volume (200-250 sq. ft of floor area depending upon the effective ceiling height) that this physical size is becoming a "de-facto" standard.

Octave-band sound pressure level measurements are usually made in such a mock-up facility at a point directly beneath the air terminal, about 5 ft. above the floor. This is usually a worst case situation. (The ARI Standard 885-90 Applications Procedure recommends the space-average of four locations in the room, but this is seldom followed in most instances.) The measured sound pressure levels are then usually converted to a simplified "NC" rating and evaluated on a 'pass-fail' basis. (See Chapter 3 for a description of how NC ratings are determined.) The RC sound descriptor is becoming more important in order to evaluate to some degree sound quality issues.

UNCERTAINTIES IN THE APPLICATION OF MANUFACTURERS' DATA

The possible uncertainties in the application of manufacturers' data for air terminal units are provided in Table 11-1 below. These are based upon utilized test standards and results found in testing programs.

AIR TERMINAL UNITS FAN POWERED MIXING SHUT-OFF VAV BOXES						
OCTAVE MID-BAND FREQUENCY	MEASUREMENT UNCERTAINTY OF STD. TEST PROCEDURE [2] dB	REPEATABILITY OF TEST DATA[3] dB	PRODUCTION VARIABILITY[4] dB"	CUMULATIVE UNCERTAINTY DUE TO ALL PREVIOUS FACTORS[5] dB	VARIABILITY OF RANDOM SAMPLE RE. CATALOG[6] dB	BANDS MOST AFFECTED BY SYSTEM EFFECTS[7]
63	*[8]	*	*	*	*	*
125	± 3 dB	± 2 dB	+5, -2 dB	± 6 dB	+5, -2 dB	+5 dB
250	± 2 dB	± 2 dB	± 4 dB	± 5 dB	± 4 dB	+5 dB
500-4000	± 1.5 dB	± 1 dB	± 3 dB	± 3 dB	± 3 dB	
8000	± 3 dB	± 2 dB	± 3 dB	± 5 dB	± 3 dB	

Table 11-1 Estimated variability of sound power data and bands most affected by system effects.

APPLICATION OF AIR TERMINAL ACOUSTICAL DATA TO BUILDING SYSTEMS

The presentation of air terminal application data is not currently specified by any standard, although application guidelines are provided in ARI Standard 885-90. Most manufacturers present simplified estimated "NC" ratings, based upon whatever room conversion factors they deem appropriate. Most manufacturers are also using the guidelines provided in ARI Standard 885-90, which is presently the most comprehensive source of available information on the appropriate conversion from sound *power* level to room sound *pressure* level for air terminal devices.

Since, manufacturers are presently free to choose whatever application factors they desire, there is little commonality among suppliers. Although most manufacturers may state what assumptions were used in converting the product sound power level data to the simplified single-number "NC" rating, it must be recognized that these assumptions are not always appropriate to the application. Manufacturers are subjected to strong marketing pressures to present data in as favorable a light as possible but not necessarily at the most realistic

[2] Based on ANSI S12-31 for pure tones and assumes that the expected range is 1 Standard Deviation.
[3] Repetitive testing of same unit in same laboratory.
[4] Subject to manufacturing tolerances permitted in Quality Control.
[5] Based on the square-root of the sum of the squares of all uncertainties.
[6] Fan Powered Units have higher variability than Shut-off VAV Units due to differences from fan motors.
[7] See pages 11-7 through 11-9 for a discussion of system effects.
[8] * No test procedures are in place to determine 63 Hz data for these products.

conditions. Such marketing-driven uses of the available knowledge have become so prevalent that ARI is considering proposals to standardize application assumptions.

RECOMMENDED DESIGN AND INSTALLATION PRACTICES FOR AIR TERMINAL DEVICES

Problems common to all types of VAV air terminal devices include:

INLET CONDITIONS

- Poor inlet conditions (such as sharp bends or manual damper immediately before the unit, as illustrated in Figure 11-1) can cause significant errors in the sensed airflow through the unit. If these conditions exist within one equivalent diameter, the sensed flow can be significantly in error (20% or more) and result in more than the desired flow through the unit. This results in higher sound levels than predicted for both the box and the diffusers downstream.
- If flexible duct is used at the inlet, it is virtually acoustically *transparent* and can result in higher than predicted radiated sound beneath the unit. This occurs because the sound power generated by the damper in the terminal device is transmitted in both directions from the unit, and can "break out" through the transparent walls of flexible inlet ductwork. While this is seldom a problem, when inlet pressures exceed 2" w.g. and inlet ducts are 12" diameter and larger breakout noise may become significant. Break-out noise at the inlet to a unit is not presently covered in any application guides or standards.
- A sharp bend in the duct at the inlet can also generate significant noise.

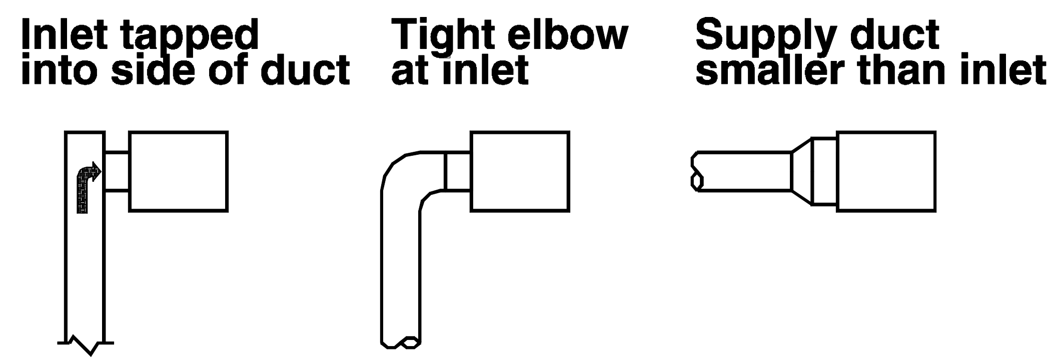

Figure 11-1: Inlet Effects

- ***Solution***

Provide as little flexible duct at the inlet as possible; make it straight, and keep air velocities well below 2000 FPM, to minimize regenerated noise. However, If ductwork upstream of the branch take-off is not acoustically lined, the insulated flexible duct may offer the only acoustical attenuation of upstream noise. Thus, this is somewhat of a "catch-22" like situation when inlet static-pressures exceed 2 in w.g.

OUTLET CONDITIONS

A poor aerodynamic configuration at the unit outlet can generate excessive noise. Tee fittings too close to a terminal can create additional radiated and ductborne noise due to excessive turbulence. The discharge air velocity profile from fan-powered units interacting with a close-coupled downstream balancing damper can create localized areas of high turbulence and increased noise generation.

- ***Solution***

Several duct diameters of straight duct (preferably lined) should be provided before any abrupt transitions, bends or splits, to allow the discharge flow profile to expand and fill the duct, minimizing potential regenerated noise.

VIBRATION ISOLATION

Air Terminals seldom require *external* vibration isolation. Most fan powered units have some form of internal motor isolation included as part of the packaged design.

PRODUCT SIZING

In general, at a given flow rate, a larger unit will be quieter than a smaller unit. There are several problems with over sizing units, however. Units should be selected to operate *within* the manufacturer's recommended air flow and pressure ranges, with best acoustical results achieved in the maximum range of 1500 to 1800 fpm inlet velocity at the full design flow rate. Primary-air inlet velocities below 300 FPM generally cannot be controlled well, as the flow sensors and transducers used become unreliable. Fans throttled to operate at airflow below their normal capacity range may surge or stall. If the speed is lowered below about 600 RPM, reduced bearing life will occur, because of a lack of reliable lubrication when sleeve bearings are used. Also electric heaters can present problems with minimum velocities where overheating of the heating elements may occur when oversized units are used.

On the other hand, inlet duct velocities above about 2500 FPM can result in high self-generated inlet noise, high pressure loss, and may exceed the flow sensor's maximum range of control.

- ***Solution***

Select units which are in the *middle* of their published operation range with around two-thirds of rated airflow rate and 2" wc inlet static pressure being recommended upper operating limits.

VARIATIONS IN ROOM SOUND LEVEL

In VAV applications, room sound levels due to air terminal devices will vary as a function of the airflow necessary to handle the thermal load. When the thermal load is nominally constant, as in most interior spaces, the room sound level is also relatively constant. However, when the thermal load varies the room sound level varies as well. This is most noticeable in perimeter spaces as the sun moves across a building during the daytime. Areas exposed to the sun have not only the highest thermal loads, but also the highest sound levels. When units are selected to meet the design noise criteria under peak load conditions, the reduced air delivery at lower thermal loads may result in the loss of beneficial masking background noise, with a corresponding degradation in the quality of speech privacy perceived in the environment.

- ***Solution***

Series-flow (constant-volume) fan-powered terminals often can be used to minimize the *impact* of a loss in speech privacy at reduced thermal loading. This is possible, because both the air volume circulating in the external system and the sound generation of the unit are relatively constant. However, when the quality of speech privacy is a *critical* factor in the room-acoustics design, the use of a separate electronic sound-masking system provides the best assurance of meeting the specified requirements. Note that while these terminals are constant volume discharge that the inlet primary air usually uses a VAV valve or damper with the balance of the air provided by the induction air port.

OTHER PRACTICAL NOISE CONTROL OPTIONS

- Maintain the lowest practical duct static-pressure.

- Maintain duct static-pressures as constant as possible. Dampers and orifice plates installed upstream of VAV air terminals to reduce high duct pressures at branch takeoffs near the supply fan can reduce the operating static-pressure of the unit, and thus reduce the sound level at rated airflow. Such pressure-reduction devices should *not* be located closer than five duct diameters upstream of the unit, to avoid introducing errors in the terminal's inlet velocity sensor. However, it must be pointed out that the pressure-reduction across dampers and orifice plates at branch take-off varies with the square of the velocity of through-flow. Therefore, at *low* airflow through the VAV terminal, nearly the *full* pressure in the main duct may be present at the box inlet, because the external pressure-reduction devices cease to be effective.

- If discharge noise is a problem, or if there is no lining downstream of the terminal, the addition of about 5 ft. of non-metallic core insulated flexible duct at each diffuser outlet can often serve as a 'flexible sound-attenuator'. Flexible duct is generally overlooked as a potential noise reduction device, because acoustical performance data is not commonly available. However, insertion loss and breakout factors for typical non-metallic core fiberglass insulated flexible duct are currently listed in ARI Standard 885-90. In the application of flexible duct, it is important to ensure that the installation is as straight as possible; badly kinked flexible duct can add significant pressure drop and self-generated noise to the system.

- If radiated noise is a problem in units *without* induction ports, lagging the box, and the first few feet downstream with drywall, using a construction adhesive and screws on 6" centers is sometimes effective (refer to Chapter 5, Figure 5-5 for a more detailed description). The relative effectiveness of this lagging is greatest on the larger size units, because the area around the controls that cannot be lagged is a smaller percentage of the total exposed surface area. However, such lagging treatment is *not* beneficial on units having induction ports, because the amount of sound radiated directly through these ports is usually significantly greater than that radiated by the box casing. Try to locate the unit away from open return air grilles.

- Whenever possible, utilize lined ductwork, flexible duct, and multiple branches downstream of the terminal device. Try to maintain at least 2-3 diameters of lined duct *downstream* of the boxes before making the first diffuser takeoff. Use rigid round ductwork *upstream* of the terminal device.

- Avoid the location of fan-powered or air-to-air induction terminals in regions directly above acoustically sensitive spaces when low background noise is desired.

FAN SPEED-CONTROL

Fan powered air terminals with a fan-speed controller will modulate air volume much more quietly than with damper controls. Almost all series and parallel fan-powered air terminals have a fan-speed controller installed as a standard feature. However, it is most important to ensure that the speed controller and motor are properly matched by the manufacturer, because at low rpms excessive motor hum (typically pure tones at 120 Hz and higher harmonics) may result if the combination is mis-matched. In addition, at motor speeds of 600 RPM or less, the centrifugally lubricated sleeve bearings used on most units will have shortened life due to insufficient lubrication.

FLOW SENSORS

Duct pressures change in VAV systems in response to changing load requirements throughout the building. In order for VAV air terminals to operate independently of any inlet pressure variations (called pressure-independent operation) they require flow sensors in the unit to measure actual flow rates and signal the controller to regulate the damper position. A side benefit of pressure-independence is a much more stable acoustical performance.

The flow sensor, however, is not without problems. It generates both additional pressure drop and noise. A sensor with optimum accuracy at *low flow* rates (where the unit is the most quiet) often has higher self-generated sound and pressure losses at the higher airflow volumes. Any flow sensor design must make a compromise between pressure drop, noise and signal strength. Flow-averaging probes are used to allow operation at less than ideal inlet conditions, and at lower airflow than a pitot-tube type sensor. Thus, changing the type of inlet sensor can significantly affect the sound generation of a unit. ARI certified units are required to have the most common type of flow sensor device installed when rating the unit. Special controls, however, may require special inlet sensors, and this is a factor to consider in an acoustically critical application. When in doubt, always specify that the submitted sound data be obtained with the flow probe to be used on the project installed.

EFFECT OF CONTROLS ON NOISE

The types of controls used on an air terminal device may affect the resulting sound level in the receiving space:

- Different control types have differing minimum-flow capabilities. Pneumatic-type, true proportional, continuous signal / response controls are often better at controlling low flow rates than most digital controllers, whose typical response at the low end tends to be a *step function* (often with surprisingly large steps). As airflow varies at minimum conditions, sound levels will also vary. More importantly, air pressures in the ductwork near the unit can be caused to fluctuate, creating noise fluctuations in adjacent units.

- The minimum capabilities of different types of controls can affect the selection of unit sizes. For example, a pneumatic unit can be selected *one size larger* than a DDC (Direct Digital Control) unit when controlling at the desired minimum value. This will result in quieter comparative performance at the higher airflow.

- Some analog electronic and DDC controls have the capability of varying the fan speed in fan-powered air terminals. Thus by maintaining the fan speed as low as possible reduced noise generation will be present.

In summary, both the acoustical signature and the selection process of VAV Air Terminals are affected by the control selection for a unit.

EXTERNAL DUCTWORK CONSIDERATIONS

Inadequate Length Downstream Of Fan-Powered Terminals

Very short diffuser distribution plenums downstream of fan-powered air terminals can result in fan instability in the form of stall or surge. The fan is located on the discharge side of many fan-powered units, and several diameters of downstream duct are required to achieve low turbulent, properly established and stable airflow. If an abrupt transition occurs before fully-developed airflow is established, the excessive turbulence created results in a fluctuating discharge pressure seen by the fan, which is sufficient in some cases to cause the fan to become unstable. Significant problems of this nature occur in many retrofit applications, where adequate space to obtain fully-developed airflow downstream of the fan is unavailable. As a consequence, fan surge is frequently encountered, and this affects not only the air performance, but introduces annoying, fluctuating sound levels in the receiving room.

Indoor Air-Quality (IAQ) Impact on Noise

The use of glass-fiber and other porous duct-lining materials has become an important concern with regard to indoor air-quality (IAQ). Porous materials, if allowed to become dirty within a high relative-humidity environment, become an ideal breeding ground for microbial growth. Although ASHRAE is presently developing HVAC system design guidelines to minimize this potential problem, the current trend, driven by IAQ concerns, often is to eliminate the use of porous, acoustically absorbent duct liner materials in air-distribution systems. This has a significant impact on the system noise level coupled to the receiving space through air terminal devices and outlets. As a result, the elimination of porous, acoustically absorbent duct liner materials significantly impacts the system designer and component manufacturers, who must jointly find new methods for coping with up to 20 dB higher system noise levels than encountered at the present time. Hopefully, there is a rational solution to this potential problem that will not require the complete elimination of acoustically absorbent materials in *all* areas of the air-distribution system. One possibility is to use facing materials of plastic or metal-foil sheeting to protect the porous surface from direct exposure to the airstream. The disadvantage is that such facings, unless they are *very thin*, will substantially degrade the acoustical efficiency in the mid- to high-frequency region -- where the need for sound absorption is the greatest!

These concerns have been seen to produce serious sources of noise problems in regions of the country with high humidity climates. Often the noise problems have not been discovered until after commissioning and building occupancy where remedial noise control solutions are difficult to find and often expensive to put into place. Inline duct silencers are often tried with less than desired results with the introduction of self-noise generation by the silencers.

CEILING PLENUM EFFECTS

The combined acoustical effects of a suspended ceiling, with a typical plenum space above, has been empirically developed from "in-situ" tests, conducted primarily on fan-powered terminals, in several Industry mock-up facilities. The results are incorporated in tabular form in ARI Standard

885-90. A number of acoustically-sensitive projects using fan-powered terminals have been successfully installed, based on the procedures recommended in this application standard.

However, recent research funded by ASHRAE[9] suggests that some terminal types, other than fan-powered boxes, may require different attenuation factors to convert from the product sound power level, determined in a reverberant test facility, to room sound pressure level in a typical receiving room environment. This recently completed research study results have not been sufficiently analyzed to the extent that new application guidelines have yet been formulated. Until this is resolved a full scale mock-up is still the best alternative in critical situations.

SPECIFIC AIR TERMINAL APPLICATION NOISE CHARACTERISTICS

REHEAT AIR TERMINALS

VAV Air Terminals often have factory supplied attached hot water or electric coils to raise the temperature of the air supplied to the space as illustrated in Figure 11-2. The acoustical implications of these devices include:

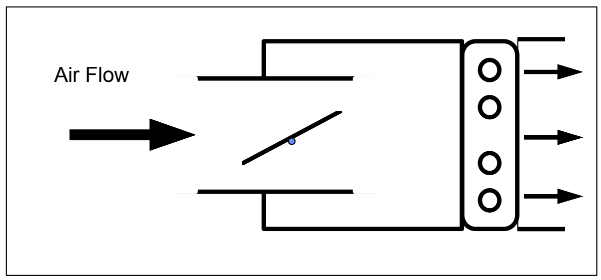

Figure 11-2 VAV Terminal with Reheat Coil

Water Coils

Because of the additional pressure drop introduced by the presence of water coils, the pressure drop across the control damper of a VAV unit that is required to obtain the desired flow volume is reduced. This typically reduces the damper-generated noise for a given pressure in the inlet duct. Water coils, themselves, seldom produce regenerated noise, unless they are mis-matched in size to the attached VAV unit. The beneficial effect of water coils attached to a VAV unit is not routinely reported in manufacturers' acoustical data.

Electric-Heat Units

Electric heating coils typically have little acoustical effect. However, electric heater air distribution baffles, which can introduce a pressure drop, are installed to equalize airflow across the heating coils. Although the baffles may generate some additional noise, this tends to be offset by the lower pressure drop required across the unit. Thus damper generated noise is reduced. In many cases electric-heat equipped units must be greater in length to comply with code restrictions imposed upon damper location upstream of heating elements. Consequently, these units are usually quieter than standard units, because of a greater length of integral acoustically lined duct.

DUAL- DUCT AIR TERMINALS

[9] ASHRAE project RP-755 "Sound Transmission Through Ceilings from Air Terminal Devices in the Plenum". Principal Investigator: A.C.C. Warnock, National Research Council Canada.

These units are used to mix two different streams of air flowing into a *common* outlet duct as illustrated in Figure 11-3. Because there is considerably more exposed sheet metal area in dual-duct units, the radiated sound is usually higher than that of an *equivalent* single-duct unit.

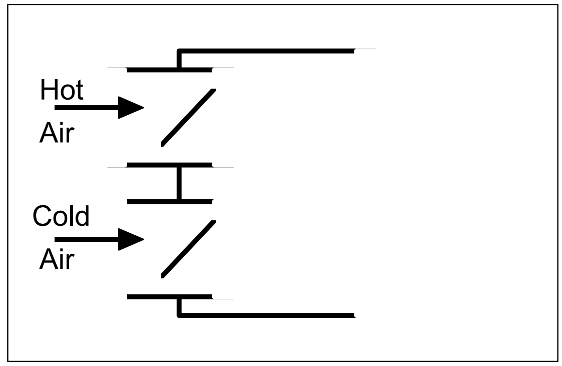

Figure 11-3 Dual-Duct Air Terminals

Acoustical data is seldom provided for the combined flow, because it is not covered in the present test procedures. However, It can be calculated by logarithmic summation of the contributions of each inlet duct. The worst case condition is typically when *one* inlet is furnishing the full design airflow at the maximum expected inlet static-pressure.

Discharge mixers as illustrated in Figure 11-4, if supplied, will add increased pressure drop and alter unit sound levels. The increased pressure drop will lower valve-generated sound, by reducing the velocity across the damper for a given inlet pressure. The mixer will tend to introduce some self-generated noise, but if factory fabricated, acoustical data should be available. However, if field installed, this factor will be an unknown.

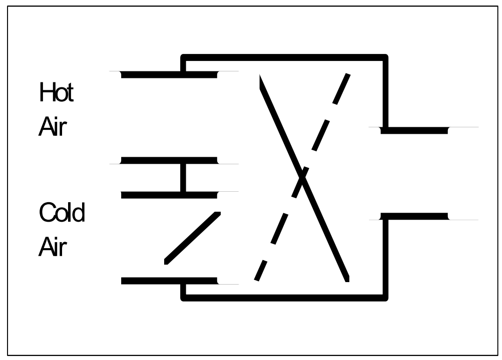

Figure 11-4 Dual Duct with Mixers

BY-PASS UNITS

By-pass units, as illustrated in Figure 11-5, are not truly VAV boxes in the sense of the commonly accepted definition, because they do not vary the quantity of air delivered by the system supply fan. They are used in conjunction with constant-volume systems, where variable-volume air delivery to a particular zone is desired. Supply air is directed to either the room diffusers, or partially bypassed to the ceiling plenum (or sometimes ducted back to the central fan) in proportion to a thermostat signal. These units are not intended to be used as VAV/pressure-reducing air control devices, although they often are used. This typically leads to an early damper failure and noise problems, because By-Pass terminals are often rated at much lower inlet velocities and pressures than conventional VAV air terminals.

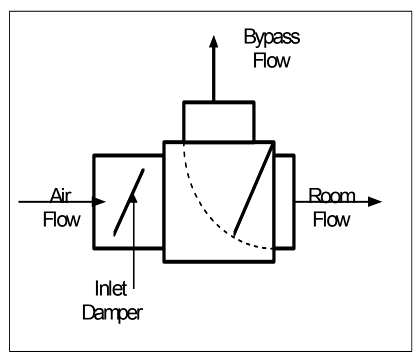

Figure 11-5 By-Pass Units

If a By-Pass unit is not properly installed or balanced, both acoustical and flow-control problems will result. By-Pass units used at airflow exceeding the manufacturers' nominally rated flow (typically 1500 fpm maximum) or inlet pressure (typically 0.3" wc), may actually fail in operation. In general, even at rated airflow rates and pressures, By-Pass units generate significantly more sound than typical VAV terminals.

AIR-TO-AIR INDUCTION AIR TERMINALS

Air-to-air induction air terminals are employed to mix plenum air with cold supply air, using the velocity-pressure in the supply duct nozzle to induce plenum air, providing a nearly constant airflow with variable temperature to the space without mechanical mixing. This design requires fairly high inlet duct pressures to develop the velocity necessary for adequate induction. At least one port is open to the plenum to obtain the induction air. As a result, the potential for radiated noise problems is fairly great, and care must be taken in the selection and location of these units to minimize complaints.

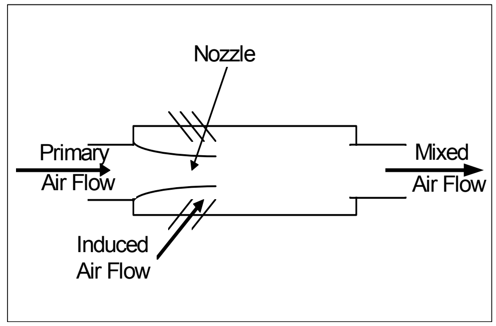

Figure 11-6 Air-to-Air Induction Terminal

FAN-POWERED AIR TERMINALS

These units are available in both parallel (variable-airflow, “side pocket”) and (constant-airflow) series flow versions. These air terminals can create low-frequency noise problems when located above sensitive occupied spaces.

Parallel, VAV Fan-Powered Air Terminals

Parallel air terminals, as illustrated in Figure 11-7, deliver a varying flow volume to the room in the cooling mode and a constant flow volume in the heating mode. The fan is typically sized to supply about one-half the VAV maximum primary airflow volume and the level of fan noise is often less than with a comparable series-flow terminal. The most significant problem with noise occurs when the fan starts on demand for heat to the space and the character of sound noticeably changes. Sound attenuators installed on the induction inlet port are sometimes effective with parallel units, because the sound escaping from the inlet port usually dominates that of the direct casing radiated noise.

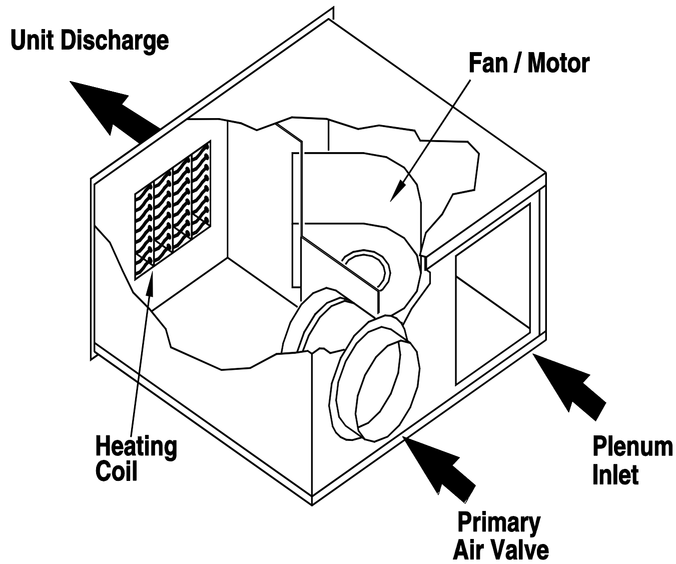

Figure 11-7 Parallel, VAV fan-powered air terminal

Series Flow, Constant-Volume Fan-Powered Air Terminals

In series flow air terminals, the fan operates continuously whenever air is delivered to the room. As illustrated in Figure 11-8, the fan is located at or near the discharge of the unit. Consequently, the level of discharge noise is relatively constant, as is the airflow rate, regardless of the thermal load. When acoustically lined duct is used following the unit discharge, excessive noise is seldom a problem. However, applications that do not permit the use of downstream duct liner frequently experience problems with excessive discharge noise.

Radiated sound levels vary significantly with variations in both duct pressure and thermal load. Attempts to quiet the sound radiating from the induction port with attenuators are seldom successful, because the added pressure drop requires an increase in fan speed to restore air delivery, and this essentially offsets any anticipated benefit. Another effect of adding inlet sound attenuators (and with some "quiet" designs), is a condition sometimes referred to as "fan shift". This is a condition where the unit delivers more air when in the full-cooling mode, than when in either the partial cooling or heating mode. If the induction port offers a significant restriction to induced air flow, then there will be a change in airflow to the room when the primary valve varies from minimum to maximum flow rate. This can cause objectionable changes in diffuser noise levels as thermal loads change.

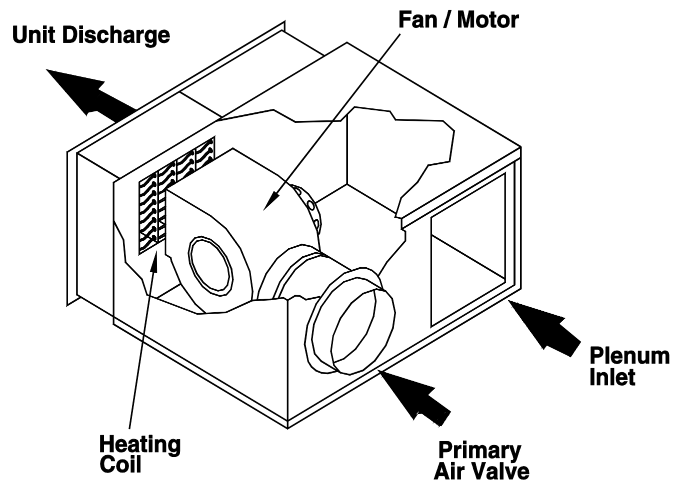

Figure 11-8 Series flow, constant-volume fan-powered air terminal.

Series fan air terminals are usually equipped with a low-cost silicon-controlled rectifier (SCR) speed-controller installed in the unit for balancing purposes. The SCR device is usually a "chopper" type of speed-controller that reduces the fan speed and thus fan-generated noise, but has the potential of increasing motor 'hum' if the motor and speed controller are not matched properly. This can produce annoying tonal components at 120 Hz and higher harmonics that are difficult to attenuate.

If fan speed is lowered much below 600 rpm, excessive motor bearing wear will result. Also, if the start-up sequence allows primary air to enter the unit prior to fan start, the unit may have insufficient start-up torque to overcome the tendency for these units to rotate backwards. It is not uncommon to discover that series boxes have been oversized for the application, as a result of a desire for quiet operation. The reduced airflow further causes the motors to run hotter, reducing expected motor life, and thereby increasing operational costs.

LOW-PROFILE FAN-POWERED AIR TERMINALS

Low-profile fan-powered air terminals are available in both parallel and series types. Usually designed to have an installed height no greater than 10.5", these air terminals have limited capacity, if compared to the more typical taller versions. When installed in shallow, rather than in the typical 3 ft depth plenum spaces, the radiated noise via the ceiling path can be significantly increased.

RETRO-FIT AIR TERMINALS

Many manufacturers offer a number of solutions for retro-fitting existing systems. There are four general types, and each has unique acoustical characteristics, and are difficult to predict. In most cases, these devices are used to operate at lower system pressures than originally required (to reduce energy costs), and result in reduced noise levels:

- Type 1: Internally-mounted, design specific, mechanical constant-volume replacement units. These are custom designed to replace specific high-pressure mechanical dampers with externally-powered (electric or pneumatic) VAV dampers.
- Type 2: Externally-mounted VAV conversion units. These are add-on units, typically round ducts with internal dampers and flow sensors, which are added at the inlet of existing units, which are then gutted of their mechanical flow controllers. The old unit casings then act as mixers or sound attenuators.
- Type 3: Slip-in add-on VAV damper. Designed to be slid into existing ductwork, these are designed to add a VAV control zone where one didn't exist before.
- Type 4: Sleeve-Type VAV dampers. These are essentially a section of duct with an installed VAV damper and flow sensor. Used in both new and existing buildings, these are custom made to fit the application.

SUGGESTED SPECIFICATIONS FOR AIR TERMINALS

An ideal specification for air terminals should specify the octave-band sound power level *limit* spectrum at design static-pressure, and include a statement of the factors to be used in conversion from sound power level to room sound pressure level. The specification should also specify the room noise criterion used as the basis for system design. The use of ARI Standard 885-90, "Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets" is recommended for guidance in establishing realistic sound power to sound pressure conversion factors.

It is important to understand that the design room noise criterion applies to the system as a whole, and only indirectly to the noise produced by the specified component. However, it follows that the contribution of the component to the total system noise in the room cannot exceed the spectrum of the specified room noise criterion at any point. Thus, there are two ways to develop a specification for air terminal devices:

- The first is to specify the maximum octave-band sound *power* levels of the unit, by working *backwards* from the design room sound *pressure* level criterion for the system as a whole and adding the conversion factors from sound *pressure* to sound *power* level using the guidelines recommended in ARI Standard 885-90.
- The second method is to specify the system design room sound *pressure* level criterion, and make a unit selection based on the manufacturer's catalog ratings that have been derived by using suggested "default" factors provided in ARI Standard 885-90.

The second method is easier for the specifying engineer to use, but its reliability depends upon how close the "default" conversion values used match the details of the intended application. Therefore, it is essential that the manufacturer clearly list the assumptions made in cataloging unit performance in terms of room sound *pressure* level noise criteria.

For example, the following factors must be considered in cataloging the acoustical performance of a VAV air terminal device for discharge noise:

1) The total number of air outlets served by the terminal device.
2) The number of these outlets serving a typical room.
3) The dimensions of this typical room.
4) The length and dimensions of acoustically lined duct downstream of the unit discharge.
5) The size and typical length of flexible branch duct connected to an air outlet.
6) The specified room noise criterion to be met.

7) The location in the room where the specified room noise criterion is to be qualified.

The following factors must be considered in cataloging the performance of a VAV air terminal device for casing and induction port (when present) radiated noise:

1) The height of the ceiling plenum.
2) The type of ceiling beneath the unit.
3) The dimensions of the room beneath the unit.
4) The size of any openings in the ceiling for plenum return, if any.
5) The specified room noise criterion to be met.
6) The location in the room where the specified room noise criterion is to be qualified.

The logarithmic addition of the discharge and radiated sound in the room, attributable to the air terminal device, will establish whether or not the unit meets the specified design room noise criterion. The criterion is met if this addition results in octave-band sound pressure levels that do not exceed the spectrum of the specified criterion at any point.

An example of the first method, *backward* calculation, is as follows:

VAV TERMINAL SOUND/PRESSURE SPECIFICATION

Project Guide Specification for Manufacturers' Sound Data

The pressure drop through the units shall not exceed scheduled values, including hot water coils. The unit manufacturer shall furnish certified sound data for both casing radiated and discharge sound levels as tested per ARI Industry Standard 880-94. Both Discharge and Radiated sound shall result in room Sound Pressure levels not to exceed those listed below, with a tolerance of + 2 dB in any band for less than 20% of the units. The Sound Pressure shall be determined in accordance with ARI 885-90 from certified sound power levels determined in accordance with ARI 880-94, with the listed assumptions:

Select Desired Noise Criteria

Desired Room Sound Pressure level (dB) by Octave Bands to Meet Project:
For a Design Sound Criteria of RC of 40(N)

	Octave Band Mid-band Frequency (Hz)					
	125	250	500	1000	2000	4000
Desired Room Sound Pressure Level (dB):	60	55	45	40	35	33

Application Assumptions

Nominal Terminal Unit Inlet Size is 12” Diameter,
Nominal Duct Inlet Static Pressure is 1.0” wc, and
Environmental Adjustment Factor is to be used (see ARI 885-90 for discussion).

Discharge Sound

Sound Power Division based on Five (5) Feet of Acoustically Lined Duct,
Two (2) Power Splits or Divisions,
Five (5) Feet of 8" Diameter Flexible Duct to Each Diffusers, and
End Reflection based on a 8" Diameter Round duct.

Radiated Sound

ADC/ARI Standard 885 Ceiling / Plenum with No Duct Breakout

	Octave Band Mid-band Frequency (Hz)					
	125	250	500	1000	2000	4000
Ceiling/Plenum Attenuation (dB):	9	10	12	14	15	15

Room Absorption based on a 2400 Cu. Ft. room, 5 Ft. from the source:

	Octave Band Mid-band Frequency (Hz)					
	125	250	500	1000	2000	4000
Room Effect (dB):	5	7	8	9	10	11

Maximum Sound Power Level

Based on these assumptions, Neither the Unit Radiated Nor Discharge sound power levels shall exceed the following levels at an inlet static pressure of 1.0 " wc.:

Both Radiated and Discharge sound power levels shall be based on the most current ARI Certified data, as reflected in the most current Certified Product Directory.

Octave Bands (Hz)	125	250	500	1000	2000	4000
Radiated Lw (dB)	77	74	66	64	61	60
Discharge Lw (dB)	83	81	79	87	85	68

CHAPTER 17
OVERVIEW OF HVAC TEST PROCEDURES

Charles Ebbing
Warren Blazier

ARI
AMCA
ASHRAE
ASTM
CTI

SOUND TESTING OF HVAC PRODUCTS

Manufacturers' published information on product sound performance is usually based on specified test set-ups and/or operating conditions conducted in accordance with ARI, AMCA, ASHRAE, CTI, ASTM or other Industry Standards. Table 17-1 presents a guide to the acoustical test standards most often used by the HVAC industry. Representing four generic acoustical test methods, these are:

1. Reverberant Room Method for *Determining* Sound Power Level
2. Free Field Above a Reflecting Plane Method for Measuring Sound Pressure Level
3. Close-In Measurement Method for Measuring Sound Pressure Level
4. Acoustic Intensity[1] Measurements used mainly for diagnostic purposes

THE REVERBERATION ROOM METHOD FOR DETERMINING SOUND POWER LEVEL

OVERVIEW

Most HVAC acoustical rating data is determined in reverberant rooms. The HVAC equipment that is normally tested using the reverberation method is identified in Table 17-1 and in Figure 17-1.

[1] Acoustic Intensity is not currently used as a primary method of rating HVAC products, However its use is increasing for diagnostic and research measurements.

HVAC EQUIPMENT	STANDARD	STANDARD TITLE	Lw (Power) Lp (Pressure)	RATING LOCATION	APPLICATION STD/ *measurement locations*
Air Terminal	ARI 880 ASHRAE 130P	Air Terminals (ANSI/ARI 880-89)	Lw	Indoor	ARI 885
Air Diffuser	ARI 890 ASHRAE 70	Rating Of Air Diffusers And Air Diffuser Assemblies	Lw	Indoor	ARI 885
Fan Coil Unit	ARI 350	Sound Rating Of Non-Ducted Indoor Air-Conditioning Equipment	Lw	Indoor	NONE
Packaged Air Handler	ARI 260P	Sound Rating Of Ducted Air Moving and Conditioning Equipment	Lw	Indoor	NONE
Built Up Air Handler	AMCA 300	Sound Rating Of Ducted Air Moving and Conditioning Equipment	Lw	Indoor	NONE
Ventilating Fan	AMCA 300 AMCA 301	Reverberant Room Method for Sound Testing of Fans Methods for Calculating Fan Sound Ratings from Laboratory Test Data	Lw	Indoor	AMCA 302 AMCA 303
Packaged & Year-Round Air-Conditioners (Rooftop)	ARI 260P ARI 370	Sound Rating Of Large Outdoor Refrigeration and Air-Conditioning Equipment	Lw	Indoor Outdoor	NONE
Air Cooled Chiller < 10 tons	ARI 270	Sound Rating of Outdoor Unitary Equipment	Lw	Outdoor	ARI 275
Air Cooled Chiller > 10 tons	ARI 370	Sound Rating Of Large Outdoor Refrigeration and Air-Conditioning Equipment	Lw	Outdoor	NONE
WSHP Water Source Heat Pump	ARI 260P	Sound Rating Of Ducted Air Moving and Conditioning Equipment	Lw	Indoor	NONE
Packaged Heat Pump	ARI 260P	Sound Rating Of Ducted Air Moving and Conditioning Equipment	Lw	Indoor Outdoor	NONE
Vertical Air Conditioner	ARI 260P	Sound Rating Of Ducted Air Moving and Conditioning Equipment	Lw	Indoor Outdoor	NONE
Reciprocating Chiller	ARI 575	Method Of Measuring Machinery Sound Within Equipment Rooms	Lp	Indoor	***Avg Levels At 1 Meter From Sides Of Unit***
Screw Chiller	ARI 575	Method Of Measuring Machinery Sound Within Equipment Rooms	Lp	Indoor	***Avg Levels At 1 Meter From Sides Of Unit***
Centrifugal Chiller	ARI 575	Method Of Measuring Machinery Sound Within Equipment Rooms	Lp	Indoor	***Avg Levels At 1 Meter From Sides Of Unit***
Cooling Tower	CTI Code ATC-128	Code for measurement of sound from water-cooling towers"	Lp	Outdoor	***Usually Measure At 5 ft and 50 ft On Sides And Top***
Centrifugal Water Pum	ANSI/ HI 9.1-9.5	Pumps- General Guidelines	Lp	Indoor	NONE
Sound Attenuators	ASTM E-477-96[a] ISO 7235-199[b]	[a] Standard Test Method for Measuring Acoustical and Air Flow Performance of Duct Liner Materials	INSERTION LOSS	N/A	[b] Acoustics - Measurements Procedures for Ducted Silencers -Insertion Loss, Flow Noise and Total Pressure Loss

Table 17-1 Guide to Common Sound Standards

HVAC PRODUCTS TESTED USING REVERBERANT ROOMS
Built up Fans
Air Terminals
Exhaust Fans
Room Fan Coils
Roof Ventilators
Air Handling Units
Sound Attenuators
Air Cooled Water Chillers
Outdoor Condensing Units
Packaged Air Conditioners
Grilles Registers & Diffusers

Ratings are in Sound Power Level Lw

Figure 17-1 HVAC equipment normally tested in a reverberant-room facility using various test standards shown in Table 17-1. The example shown is the set-up for testing an outdoor air-cooled condensing unit in accordance with ARI 270-84.

Present procedures for testing in reverberant-room facilities are generally applicable only to the frequency range from 100 Hz to 10,000 Hz when measurements are made in one-third octave bands, and from 125 Hz to 8,000 Hz when measurements are made in octave-bands. ARI[2] has just completed a standard to set qualification requirements for the 63 Hz Octave Band and the latest draft of the ISO[3] reverberant room standard defines the requirements for the 63 Hz Octave Band testing.

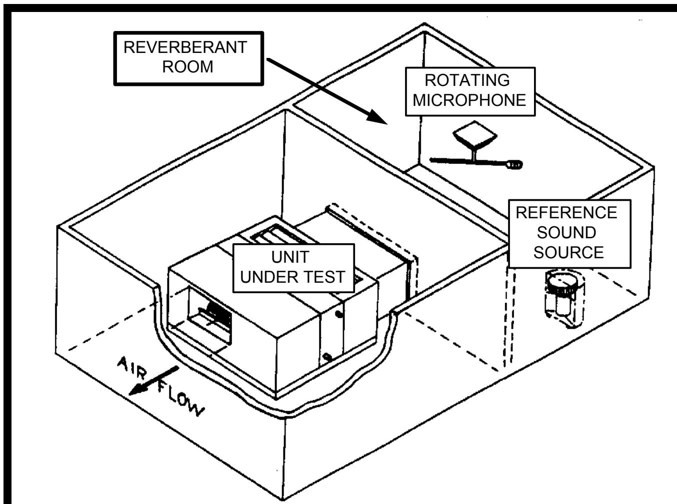

Fig 17-2 Testing for Discharge Sound

TESTING PACKAGED AIR HANDLING UNITS

Figures 17-2 through 17-4 show examples of the typical set-up for separately determining the sound power level of the discharge, inlet, and casing radiated noise of packaged air handling units using ARI Standard 260-P.

Testing For Discharge Sound

Figure 17-2 illustrates the test set-up for measuring discharge noise. The equipment is installed *outside* the test room and the discharge side is connected to the reverberant room. The length of discharge duct used in making this connection is typically equal to 2.5 equivalent duct diameters, which is the same length that would typically be used to determine fan air performance. Relief ducts between the rooms must be used to avoid catastrophically over pressurizing the rooms[4]. The operating characteristics of the fan, such as airflow, static pressure, speed, and brake horsepower are usually measured at the same time the fan sound power level data are obtained. This permits the correlation of the measured sound data with the point of operation on the fan curve. Care

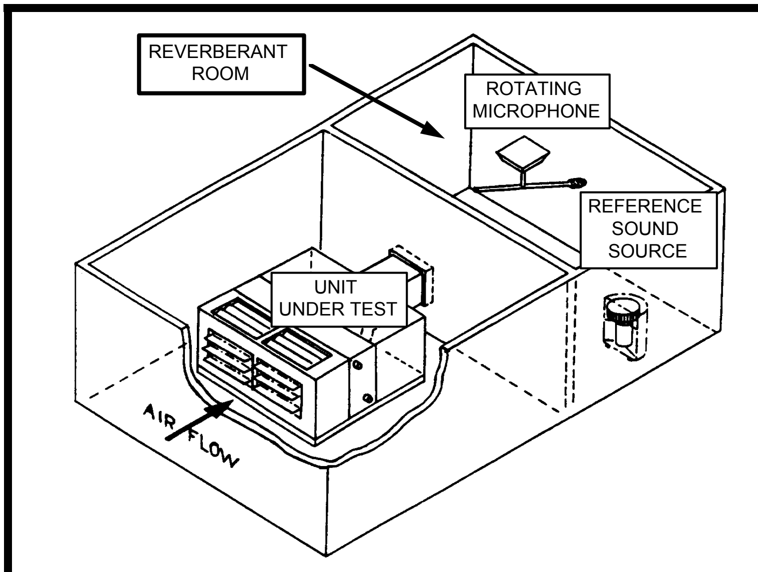

Figure 17-3 Testing for Inlet Sound

[2]ARI has recently completed ARI STANDARD 280, " Requirements for the qualification of reverberant rooms in the 63 Hz Octave Band." This standard outlines the qualification requirements for the 63 Hz Octave Band .
[3] ISO 3741 Draft Standard
[4] The reverberant room construction must be capable of safely withstanding the room pressurization necessary to operate the unit at the required static pressure.

is also taken to avoid flanking noise problems associated with inadequate vibration isolation between the test unit and the testing facility.

Testing For Inlet Sound

Figure 17-3 illustrates the test set-up for measuring inlet noise. The unit is located outside the test room and the inlet side is connected from the reverberant room to the room in which the fan is located. An acoustically lined transfer relief duct is connected between the two rooms to return the air to the fan and avoid over pressurization of the reverberant test room.[5]

Testing For Casing Radiated Sound

Figure 17-4 illustrates the test set-up for measuring casing-radiated sound[6]. The unit is located inside the reverberant room, and the supply and return air are ducted out of the test room. Heavy-gage, stiffened and lagged ductwork is used to control the sound radiation from the inlet and discharge ductwork within the test room. In this way, only the sound radiated by the casing of the unit is measured.

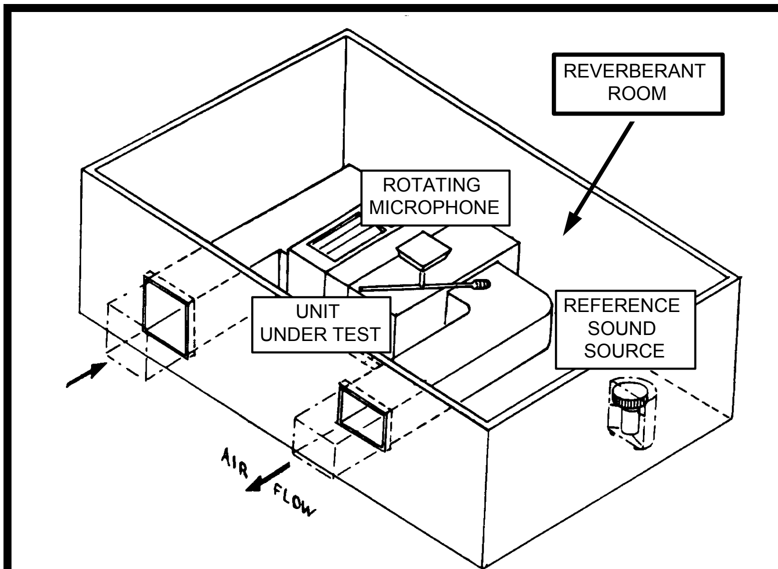

Figure 17-4 Testing for Casing Radiated Sound

TEST DATA ON LOW FREQUENCY SOUND

Current standards for testing in reverberant room specify measurements down to ***only*** 100 Hz. The meaningful determination of sound power level in the octave bands below 63 Hz requires a reverberation room that is substantially larger than 10,000 cubic feet. Thus, most manufacturers sound power level information does not include data at these frequencies, because of the prohibitive costs of construction and the difficulties encountered in qualifying the performance of the test room to the tolerances permitted by present standards.

Low frequency field application data may differ considerably for the installed systems because problems arise from poor aerodynamic design practices in connected ductwork, and/or the inappropriate selection of a fan size that cannot cope with system effects. In many cases the fan, becomes unstable and operates inefficiently with much added low frequency noise in the installed system.

[5] A *low-resistance* transfer duct into the inlet-room is also necessary to avoid artificial loading of the inlet side of the unit thus distorting fan performance and noise measurements. Unit loading is often controlled by a variable resistance in the relief exhaust duct from the discharge room.

[6] The accepted term "casing-radiated" sound may be a bit misleading to some readers. This is the sound radiating from the panels or external surfaces of the air handler.

USE OF REFERENCE SOUND SOURCE IN DETERMINING SOUND POWER LEVEL

Most ARI and AMCA test standards are based on use of a *reference* sound source (RSS)[7] in determining the sound power level of a device under test. The most common reference sound source is a small, direct-drive fan wheel that has no volute housing or scroll. The impeller is a forward-curved design with a choke-plate installed on the inlet face of the fan wheel. The choke-plate causes the fan to operate in a rotating-stall condition that is very noisy. Reference sound sources of this type are designed and built to strict specifications that result in a stable sound power level output having a relatively uniform broadband frequency spectrum. Typical types of reference sound sources are shown in Figure 17-5 and 17-6.

Figure 17-5 Horizontal axis reference sound source

Figure 17-6 Vertical axis reference sound source

However, because of the production of a pure-tone component in the rotating-stall mode of operation, the fan rotational speed determines the lowest frequency at which the unit can be used as a reference sound source. In general, a 3600 rpm unit should ***not be*** used below the 125 Hz octave-band; An 1800 rpm unit should ***not be*** used below the 63 Hz octave-band.

To determine the sound power level spectrum of an unknown source, sound pressure level measurements are first made in the reverberant-room with the reference sound source operating. Then, the reference source is turned off and similar measurements are made with the HVAC equipment under test in operation. Because the sound power level spectrum of the reference source is a known, the measured differences in sound pressure level between the reference source and unknown source correspond to differences in sound power level between the two. Thus, the sound power level of the unknown source can be indirectly determined by adding these differences to the known sound power level of the reference source. This procedure for determining sound power level is known as the *substitution or comparison* method.

[7] Commonly used RSS are manufactured by ILG Industries, Inc. (Chicago, IL), Bruel & Kjaer Instruments, Inc. (Marlborough, MA), and Acculab, Inc. (Columbus, OH).

FREE-FIELD ABOVE A REFLECTING PLANE METHOD[8] FOR MEASUREMENT OF SOUND PRESSURE LEVEL

Essentially free-field[9] sound measurement techniques depend upon being able to measure the sound pressure levels *directly* radiated by the equipment under test, without contamination from any sidewall or ceiling reflected energy. This is accomplished by either measuring out-of-doors with the equipment installed on a large paved area, or in a hemi-anechoic[10] room that has highly absorbent sidewalls and ceiling. In either configuration, the accuracy of the method depends upon how closely the measuring space approximates a free field condition without significant reflections from any surface other than floor. Large outdoor equipment such as cooling towers are typically measured per the Cooling Tower Institute Code ATC-128 when installed on a large outdoor paved area. As shown in Figure 17-7, sound pressure level measurements are made at fixed distances from the sides and top of the unit. Sound power levels can also be determined from this data by the method described in Chapter 16.

HVAC PRODUCTS TESTED IN FREE FIELD ABOVE A REFLECTING PLANE

Cooling Towers

Ratings are in Sound Pressure Level Lw

Figure 17-7 Measurement of sound pressure level from a cooling tower in a free field above a reflecting plane in accordance with CTI Code ATC-128.

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8 An essentially free field above a reflecting plane means that there are no significant reflective surfaces close to the source, except for the reflective surface it rests on.

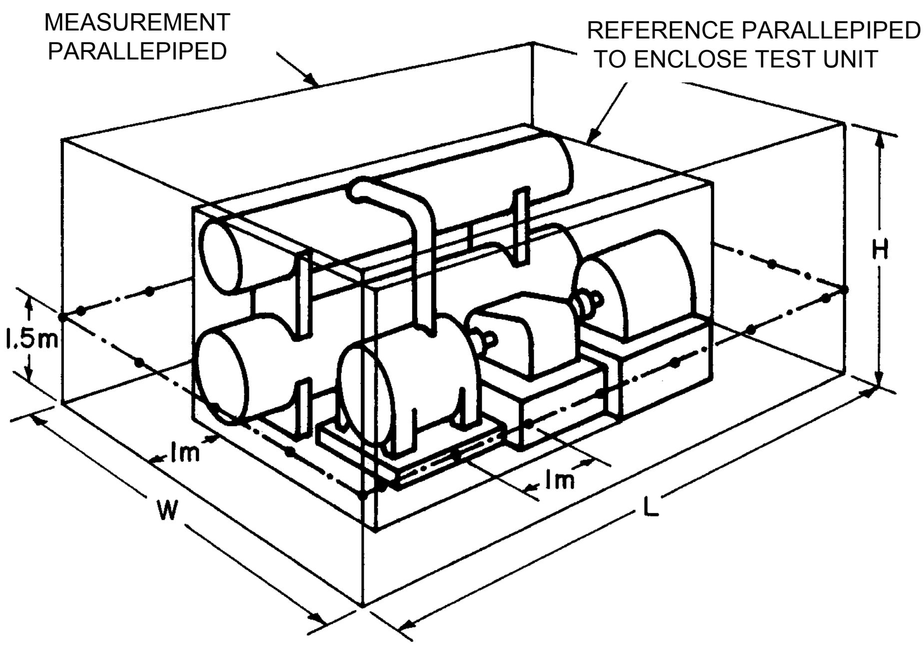
9 Testing rooms that have no significant reflecting surfaces are called free field or anechoic rooms. They are mainly used for research purposes in the HVAC Industry.

10 These test spaces require non-reflective wedges or panels on the walls and ceiling and the properties of these surfaces determine the lowest frequency that can be accurately measured.

"CLOSE-IN" METHOD FOR MEASURING SOUND PRESSURE LEVELS

HVAC PRODUCTS TESTED WITH CLOSE-IN METHOD OF MEASURING SOUND PRESSURE LEVELS

Water Cooled Chillers



Ratings are in Average Sound Pressure Level Lp

Figure 17-8 Measurement of sound pressure level from a water cooled chiller per ARI 575.

Sound pressure levels radiated by Water Chillers are measured per ARI 575 at close in distances of 1 meter from the chiller surfaces, and at 1.5 meters above the floor. ARI 575 does not specify that the measurements be conducted in a free field above reflecting plane environment, but most manufacturers measure in either large rooms or outdoors which are essentially free field [11]. Chapter 15 discusses the effect of mechanical equipment room size on the data, when the measurements are made indoors.

[11] Chapter 15 discusses the effect of mechanical equipment room size on these measurements.

SOUND INTENSITY METHOD FOR MEASURING SOUND POWER LEVEL

Recent advances in acoustical instrumentation now permit the direct measurement of sound power level[12]. This technique is largely used for research purposes and is seldom used for product rating. This may change over time as the state of the art improves.

By measuring the sound intensity with a two-channel probe over a surface surrounding a sound source, the sound power radiated by the source can be measured. One advantage of this method is that with few limitations, sound power measurements can be made in the presence of background noise and in ordinary rooms, thereby eliminating the need for a special testing environment. A second advantage is that by measuring sound intensity in the close-in area around a sound source, information on local sound directivity[13] can be obtained, that is not possible when using the reverberant-room substitution method of test. This can be particularly useful in the noise reduction aspects of product development. One disadvantage is that sound intensity measurements are sensitive to local air velocities that may be present in the vicinity of the probes and which introduce errors if the velocity is greater than 400 - 500 fpm.

Sound intensity measurement procedures are still experimental and require significant expertise to obtain meaningful and reliable rating data.

STANDARDS FOR TESTING DUCT SILENCERS

The present standards used for testing dissipative and reactive silencers are ASTM E477-96 in North America and ISO 7235-I99I elsewhere. There currently are no standard test methods available to test the performance of active silencers. Figure 17-9 illustrates a typical test set-up for determining a silencer's insertion loss, pressure drop and airflow generated noise using ASTM-E477-96.

To measure insertion loss, two separate measurements must be made. The sound pressure level in the reverberation room is measured while noise generated by loudspeakers enters the room in through a length of straight empty duct. The sound pressure is measured again after a section of the empty duct has been replaced with the silencer. The insertion loss is the difference between the two measured levels.

Silencer pressure drop is obtained by measuring the total pressure 5 diameters downstream of the silencer and subtracting it from the total pressure measured 2.5 diameters upstream. To ensure streamlined air flow for accurate measurements, a minimum of five diameters of straight duct on the inlet side of the silencer and ten diameters of straight duct following the discharge of the silencer are required.

The airflow generated noise is calculated from measured sound pressure levels in the reverberation room when originally quiet air passes through the system with the silencer installed.
The substitution method is used to calculate the sound power levels. End reflection corrections per AMCA must be added to the sound power based on the termination cross-section of

[12] ANSI Standard S1.12 or ISO 9614-1
[13] Sound directivity enables one to identify the noisiest radiating parts or surfaces of the source , and the noisiest radiating directions. See Figures 16-2 and 16-3 for example.

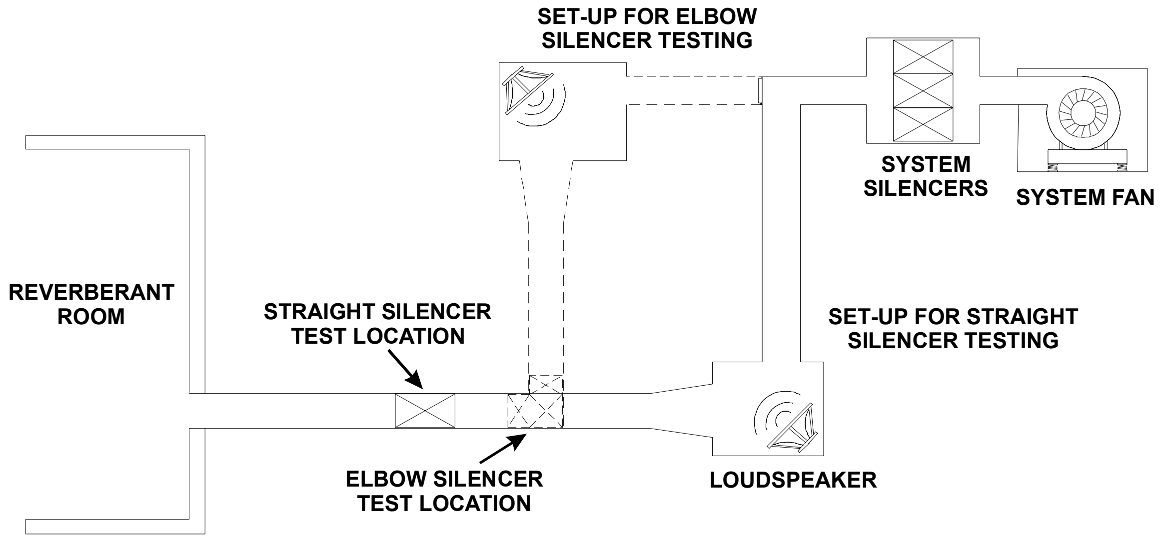


Figure 17-9 Typical silencer testing arrangement

the system duct at the reverberation room wall. Current standards do not account for possible acoustic losses from the system duct between the silencer and the reverberation room.

Variations of ± 3 dB in insertion loss values can be expected down to 63 Hz[14] even though there are presently no ANSI or ISO standards to qualify reverberation chambers below 100 Hz. Generated noise testing produces greater variation. A recent round robin showed five different laboratories reporting ± 3 to ± 6 dB over the octave band frequency range of 125 Hz to 8000 Hz when testing for generated noise according to ASTM E-477-96. Accurate data were not available below 100 Hz. Similar round robins are not known to the authors to be held for the ISO 7235-1991 Standard.

When evaluating elbow and transition silencer insertion loss and pressure drop test data it is critical to evaluate the reference empty duct section. Elbow silencers can be tested according to the ISO 7235-1991 standard using an empty duct fitting as the reference. The ASTM E-477-96 standard does not yet include provisions for testing elbow silencers. Manufacturer's could use a straight length or a sheet metal elbow (mitered or radiused, with or without turning vanes) as the reference empty duct. Each case will yield varying insertion loss data for the same silencer. Therefore for elbow and transition silencers, a description of the substituted duct element must be reported along with any performance data.

While many similarities exist, the ASTM and ISO standards produce differing results because of variations in loudspeaker location orientation, duct termination conditions and computation methodology.

[14] ASHRAE Applications Handbook, 1995 p. 43.19

VARIABILITY OF TEST DATA

Whether manufacturers ratings are approached in a typical field applications, depends mainly upon the details of product application as has been documented in the previous chapters of this book.

Which ever method of test used, variation and uncertainty are expected due to different laboratories having different size or instrumentation. Tables in each product chapter in this book gives estimates of the likely uncertainties involved.