ACOUSTICS AND THE HVAC INDUSTRY

For an acceptable indoor air quality (IAQ) design, temperature and humidity control is required. When the overall scope is broadened to maintaining a good indoor environmental quality (IEQ), proper acoustical design is also essential.

When acoustics is considered within the HVAC industry, the stringent requirements of theaters, libraries, recording studios, etc. often come to mind. Many people think sound is not an issue unless a job has a NC or RC requirement. In reality, all projects have noise requirements because of critical areas such as conference rooms, classrooms, or a manager or president's office.

This document will provide the reader with the fundamentals of acoustics and discuss the "Do's" and "Don'ts" of acoustically sound HVAC system design.

FUNDAMENTALS OF ACOUSTICS

Sound is caused when the motion of an object causes the molecules of a fluid medium (for our purposes, air) to vibrate. Human perception of sound is a function of the amplitude, frequency (wavelength) and duration of these vibrations.

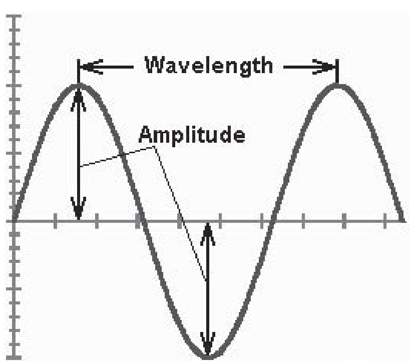

FIGURE 1.– AMPLITUDE VS. WAVELENGTH

Amplitude of Sound (dB)

Humans hear sound as a result of the pressure that sound waves exert on the eardrums. This Sound Pressure Level (SPL or Lp) is expressed in decibels. A decibel is a unit for expressing the ratio of two power-related quantities equal to 10 times the common logarithm of this ratio.

Eq. 1: $Lp = 10 \cdot \log_{10} (P^2_{(rms)}/P_o^2)$ dB

Where: $P_o = 20\ \mu Pa = 2 \times 10^{-5}\ N/m^2$

Equation 1 can also be expressed as:

Eq. 2: $Lp = 20 \cdot \log_{10} (P_{(rms)}/P_o)$ dB (ref. 20 μPa)

Sound Power Level (abbreviated PWL or Lw) is also expressed in dB, which often causes it to be confused with Lp. Sound power level is expressed as:

Eq. 3: $Lw = 10 \cdot \log_{10}(W_{(rms)}/W_o)$

Where: $W_o = 10^{-12}$ watts

It is very important to understand the difference between sound pressure and sound power. Sound pressure levels describe the "effect" of the sound. The sound pressure depends on both the sound source and the environment of the sound source and the receiver. All of the criteria or ratings used to describe the "effect" of the sound are based on sound pressure level.

Sound power levels describe the "amount" of sound produced by a source. In other words, the sound power is a characteristic of only the source and not the environment. This makes sound power the ideal way to specify equipment and to compare one supplier's offering to another's.

In order to keep Lp and Lw straight it is helpful to think of a 100-watt light bulb. The 100W output is analogous to the sound *power* level of a source. The light seen at any point in the room is analogous to sound *pressure* level that reaches the listener. The light seems brighter the closer the viewer stands to the bulb just as the sound seems louder the closer the listener stands to the source. But just as the wattage of the bulb does not change, neither does the sound power level of the source. Only the sound pressure level changes based on environmental factors.

It is also important to note that Lw cannot be measured directly; it has to be calculated from Lp measurements. Since accurate sound pressure levels are, at best, difficult to measure in the field due to reflection off nearby objects and interference from other sound sources, the only way to verify the sound power level of a unit is through a laboratory test or the sound intensity method.

Frequency of Sound (Hz)

While the sound level in dB is a measure of the amplitude of sound, frequency is a measure of the wavelength of the vibrations causing the sound (see Figure 1). Sounds with a short wavelength have a high pitch and sounds with a long wavelength have a low pitch. The frequency of sound is expressed in cycles per second or hertz (Hz).

HVAC specifications typically list equipment sound levels according to an octave band spectrum consisting of eight octave band center frequencies as shown in Table 1.

Table 1 – Typical Sound Power Level Criteria

	Octave Band Center Frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000
Lw (dB)	88	96	91	83	76	72	68	67

The wavelength of sound at a certain frequency is the speed of sound in air (approximately 1,100 feet per second) divided by the frequency of the sound.

Example 1:

Wavelength of sound at 250 Hz ≈ 1100/250

≈ 4.4 feet

Human Perception of Sound

The dynamic range of human hearing for continuous sounds is shown as a function of frequency in Figure 2. The principle regions of speech and music are also shown for comparison. The lower end of the auditory range is called the *threshold of hearing* (usually stated as 0 dB at 1 kHz). The upper end of our hearing is the *threshold of pain* (usually stated as about 130 dB at 1 kHz).

When considering the dynamic range of sound pressures humans can hear, the necessity of expressing it on a logarithmic scale becomes obvious:

0 dB = 0.00002 N/m2

130 dB = 64 N/m2

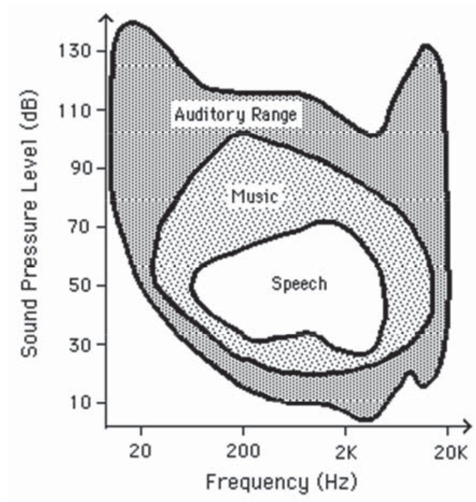

FIGURE 2 – RANGE OF HUMAN HEARING

Note that humans can hear a much wider range of frequencies than is required for speech communication, but are most sensitive to sounds in the speech frequency region of 250 to 4000 Hz. This means that office environments should have enough of background noise in this frequency region to provide adequate acoustic privacy between workstations. Sound from the HVAC equipment is what most often provides this acoustic privacy.

The perception of *loudness* is not directly related to the measured physical sound pressure level; it is summarized in Table 2.

Table 2 – Human Sensitivity to Sound Level Difference

Objective Amplitude	Subjective Loudness
3 dB change	Just Perceptible
5 dB change	Noticeable
10 dB change	Doubling of loudness

Adding Decibels

If two sources (e.g. the fan from Table 1) each produce L_w = 88 dB in the 63 Hz octave band, would the combined L_w be 176 dB, or 98 dB if they were running at the same time?

Neither. As Table 2 shows, a doubling of loudness only amounts to a 10 dB amplitude change, which is fortunate since the threshold of pain is 130 dB, far below 176 dB! Also, since decibels are expressed on a logarithmic scale they must be added logarithmically.

The formula for adding two decibel levels is:

Eq. 4: SPL1+2 = 10 · log10(10SPL1/10 + 10SPL2/10)

Example 2:

SPL_1 = 50 dB SPL_2 = 50 dB

$10^{SPL1/10}$ = 100,000 10^{SPL2}/10 = 100,000

SPL1+2 = 10 · log10(200,000)

= 53 dB

Therefore, combining two identical sound sources does not double the loudness. In fact, when two equivalent sound sources are added together the amplitude of the sound only increases by 3 dB - a barely perceptible increase. This is true whether the levels of the two sources are 20 db, or 120 db.

Figure 3 can be used when adding two sound sources of different decibel levels. For example, when adding sound sources which differ by 4-9 dB, only 1dB should be added to the higher of the two source values. When the sources differ by 10dB or more, the higher source will dominate and nothing should be added to it.

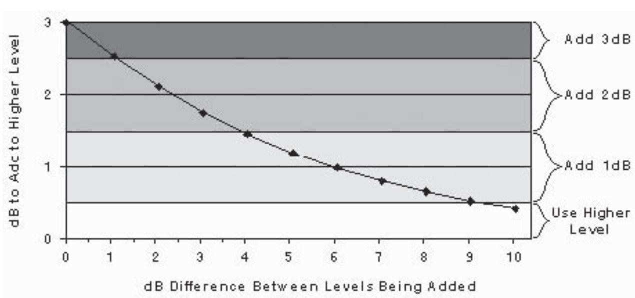

FIGURE 3 – ADDING DECIBELS

Single Number Ratings

HVAC sound is often the major source of background noise in indoor spaces and there are several single-number ratings that use the octave band levels to evaluate the suitability of HVAC sound. All of these rating methods depend on the system design, which includes the room shape, room finishes, equipment room layout, the air distribution system and VAV units, as well as the sound produced by the AHU. For this reason, manufacturers cannot guarantee the performance of air-handling units against any of the following criteria.

NC: Noise Criteria Method: The NC rating system was developed in 1957 as a means of evaluating background noise in interior spaces and has been widely used for the past 30 years, although ASHRAE does not recommend its use today. To determine the NC level, the octave band sound pressure levels in the space are plotted on a family of NC curves as shown in Figure 4. The NC level of the space is determined by the highest penetration of the data on the curves. For example, both curves plotted in Figure 4 represent NC level 40 because all of the data points fall on or below the NC-40 curve.

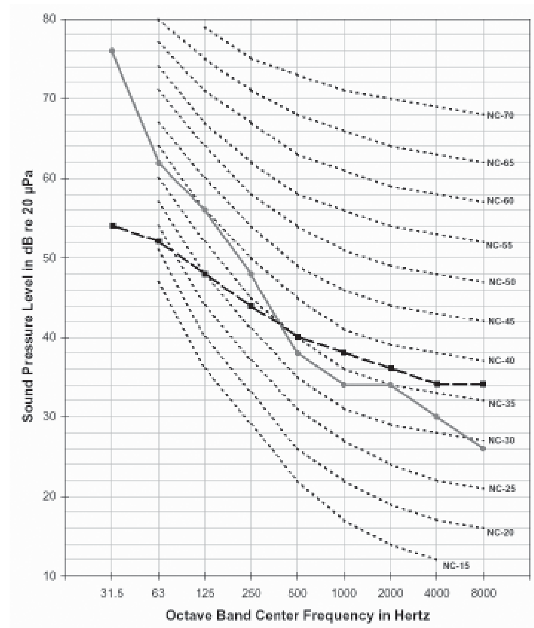

FIGURE 4 – NC CURVES

The NC Method has a few disadvantages. Since the curves do not extend below 63 Hz, the 76 dB data point in the 31.5 Hz octave band does not affect the NC rating, even though it represents an unacceptable level. This method also allows for excessive noise above 2,000 Hz due to the shallow slope of the curves. Finally, the NC method does not account for the “quality” of the sound. In other words, while both data sets in Figure 4 represent NC-40, the solid curve would have a rumbling quality due to the excessive low frequency sound, while the dashed curve would have a hissing quality due to the higher levels in the upper octave bands.

RC: Room Criteria Method: The RC rating was standardized in 1995 and is intended to establish HVAC system design goals. The shape of this family of curves represents well-balanced spectrum. Two additional octave bands were added on the low end (16 and 31.5 Hz center frequencies) to evaluate low-frequency sound. This rating assesses background HVAC noise in spaces, both in terms of its effect on speech (or speech privacy) and on the subjective quality of the sound. Until recently this was the method recommended by ASHRAE.

NCB: Balanced Noise Criteria Method: The NCB method (ANSI S12.2: Beranek 1989) is used to evaluate room noise, including that from occupant activities. The NCB method also extends the low frequency octave bands down to 16 Hz. The NCB rating is based on the Speech Interference Level (SIL = the average of the 500, 1000, 2000 and 4000 Hz octave band sound pressure levels) with added tests for rumble and hiss. Although the method is a significant improvement over the old NC curves, this rating does not see widespread use in the HVAC industry.

RC Mark II: Room Criteria Method: Based on findings and experience from ASHRAE sponsored research (Broner 1994) the RC method was revised to the RC Mark II method (Blazier 1997). The method is rather complicated and is best performed in a computer program or spreadsheet. This is the method recommended by ASHRAE for evaluating indoor acoustics. However, due to its complexity, it is doubtful that it will ever receive widespread industry acceptance.

The above methods are useful diagnostic tools that are intended to assist the HVAC engineer with system design and analysis. Unfortunately they are often misapplied by using them to specify equipment such as AHUs. Because these criteria take into account not only the sound source, but also the path of the sound and the design of the space, designers should not specify NC or RC levels for AHUs. Rather they should calculate the sound power levels for the AHUs that are required to meet the NC or RC criteria of the space.

dBA: A-weighted Sound Pressure Level: The A-weighted sound pressure level is a single-number rating system that has gained widespread use for analyzing sound in outdoor environments. The A-weighted level can be measured directly using a weighting network that is built into most sound level meters. The relative response of the A-weighting network is shown in Figure 5.

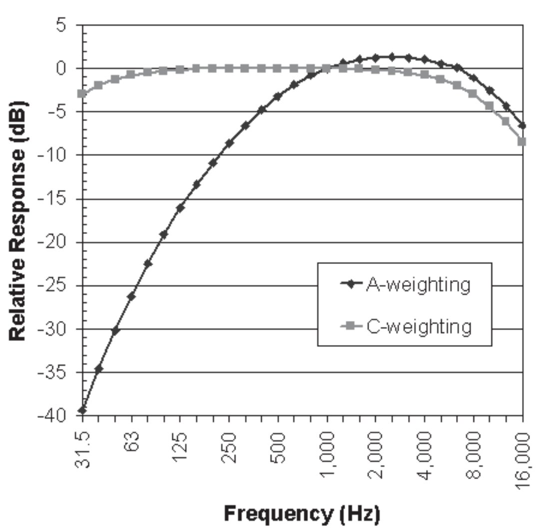

FIGURE 5 – A & C-WEIGHTING NETWORKS

The C-weighting network is also shown in Figure 4. Note that the C-weighting does not weight the low frequencies as much as the A-weighing. The difference between the A and C levels is sometimes used to quantify the low frequency content of the sound (sort of a "poor man's" octave band analysis).

The A-weighted level has been correlated with the annoyance due to noise from various outdoor sound sources, such as highway traffic. For this reason, most outdoor noise criteria are written in terms of the dBA. Although outdoor HVAC equipment could always be better evaluated in octave bands, specifications are often written in terms of the A-weighted level (dBA).

For an AHU, typically only the outside air, exhaust air or casing radiated sound components would be rated in terms of dBA, not the ducted sound components. Sound power is not typically rated in dBA because the A-weighting network is intended to rate the annoyance or "effect" of outdoor noise sources, and Lw is a *characteristic* of the noise source, not the *effect* of the noise.

For the outside air, exhaust air and casing radiated sound components, LwA ratings are valid because the conversion between sound power and sound pressure for these components is typically a "distance attenuation." Since distance attenuation doesn't vary with frequency, A-weighting can be applied directly to the sound power levels to determine the sound pressure level (Lp in dBA) at some distance from the unit. However, the same is not true of ducted sound components such as supply and return sound power levels because the attenuation caused by appurtenances such as ducts, silencers, plenums, etc. is dependent on frequency. Unfortunately, equipment suppliers have often provided dBA sound power levels for ducted sound components because they are lower. This practice has confused and deceived the market more than it has informed.

AHU TEST STANDARDS & PROCEDURES

In response to the confusion that designers have had regarding sound in the past, the American Refrigeration Institute (ARI) developed ***ARI Standard 260*** *– Sound Rating of Ducted Air-Moving and Conditioning Equipment*. The standard is intended to establish a universal method for air-handling unit manufacturers to measure and report the sound power levels of their products.

In the past, air-handler suppliers did little, if any, sound testing to validate their acoustical models. If a designer was considering three or four manufacturers for the same job, each may very well have used a different method to arrive at the sound power levels they reported. It then fell to the designer or an acoustician to sort through all of the data and come to a conclusion as to which equipment met the specification. Now however, when manufacturers report "sound data rated in accordance with ARI Standard 260," the designer or acoustician is able to make an apples-to-apples comparison of the data.

ARI Standard 260

It is important to understand that ARI-260 is not a test procedure, it is a rating standard. The actual test procedures are based on AMCA *Standard 300 – Reverberant Room Method for Sound Testing of Fans*. In the past it was common practice to specify, "AHUs tested in accordance with AMCA-300." However, this statement can lead to confusion and errors because AMCA-300 only considers the bare fan performance and doesn't account for the acoustical effect of placing the fan in the cabinet of an AHU. AMCA-300 also allows extrapolation of untested data points, and therefore requires much less testing than ARI-260, which does not allow extrapolation.

> **Suggested Specification Text:** Sound power levels of air-handling units shall be rated in accordance with ARI Standard 260.

Another point of confusion arises with the issue of "rating" versus "certification." There is no certification procedure in ARI-260 as there is in other standards such as ARI-410 or ARI-430. Since ARI-260 does not require third-party verification or round robin testing of a standard unit, it is <u>incorrect</u> to specify an "AHU *certified* in accordance with ARI-260."

ARI-260 is primarily intended to establish a standard method for rating the ducted sound components of equipment, but it also includes procedures for measuring sound at free openings (such as OSA or exhaust air) and casing radiated sound. There are several key requirements that must be met to claim conformance with ARI-260:

- Testing must be conducted in accordance with procedures referenced in ARI-260.
- A duct end correction must be applied to the ducted data.
- Adjacent "mapped" sound points on a constant speed fan line must be less than 5 dB apart in any 1/3-octave band.
- Sound data for units that were not tested can be interpolated, but not extrapolated, from the results of units that were tested, provided the difference between the tested units is no more than 5 dB in any octave band. (This is a very stringent requirement that forces manufacturers to test many different sizes.)
- Manufacturers must account for the acoustical effect of appurtenances such as coils, filters, economizers, mixing boxes, diffusers and discharge plenums.

While ARI-260 is a superior alternative to the confusion of the past, there are still a few weaknesses in the standard that have to be addressed. The first is that the casing radiated test is difficult and unrealistic. There are few, if any test labs in North America that can test AHUs larger than 60,000 cfm without violating some of the recommended procedures in ARI-260. A more appropriate test for casing radiated sound would be based on the sound intensity method (AMCA draft standard 320).

The second drawback to ARI-260 is that it does not require the measurement of airflow concurrently with sound testing. Typically, the AHU airflow is mapped using an AMCA certified code tunnel (AMCA-210). The unit is then moved to the sound test lab and the operating points are "mapped" by adjusting the system resistance (sometimes by loading the unit internally) to get to the same fan RPM and TSP. The CFM is then assumed to be the same as the AMCA-210 test. Testing of forward curved fans has shown that this is not always a good assumption. ARI-260 testing requires points near the stall line and quite often the unit can jump to the left side of the fan curve, creating different sound characteristics and erroneous results.

Uncertainty of Test Data

Measurement uncertainties exist for any test procedure, and sound testing is no different. The uncertainties in test data taken in accordance with the ARI-260 test methods can range from 1.5 to 4 dB depending on the octave band.

Sound tests are difficult to repeat exactly due to uncertainties that stem primarily from sampling the sound field in the test room and measuring the unit static pressure and airflow. Therefore, a repeatability uncertainty must also be taken into account.

Small variations in manufacturing can also affect the sound performance of equipment. Even within tolerances permitted by most quality control programs, production related uncertainties can amount to variations in sound performance of 3 or 4 dB between two "identical" units.

After taking all of these variables into account, designers should expect cumulative uncertainties in the range of the values shown in Table 3: ±6 dB in the 63 Hz band and ±4 dB in the other seven octave bands. This should be kept in mind when specifying sound power levels for equipment as well as when comparing the sound power levels of different manufacturers' equipment.

Table 3 – Uncertainties in Sound Measurement

Air-Handling Units				
Octave Mid-Band Frequency	Measure-ment Uncertainty of Std. Test Procedure dB	Repeatabil-ity of Test Data dB	Production Variability dB	Cumulative Uncertainty Due to All Previous Factors dB
63	±4	±2	±4	±6
125	±3	±1	±3	±4
250	±2	±1	±3	±4
500-4000	±1.5	±1	±3	±4
8000	±3	±0.5	±3	±4

SYSTEM DESIGN

There are very few constants when it comes to acoustics, however one thing true in every case is that it's always less expensive to design and install a system correctly the first time than it is to make the system quiet after it is installed. The following section will examine the good practices to use and the pitfalls to avoid when designing an acoustically friendly HVAC system.

Determine the Critical Path

Sound can travel several different paths from the source (the AHU) to the receiver (the room or space). It is important to know which of these paths is most likely to cause acoustical problems so that it can be dealt with in the design phase of the project.

Figure 6 shows four paths sound can take from the source to the receiver:

A. Discharge – sound from the supply air duct.

B. Breakout – sound coming through the duct walls (a common problem on poorly designed rooftop systems).

C. Return – sound from the return air duct.

D. Radiated – sound radiated through the walls of the building.

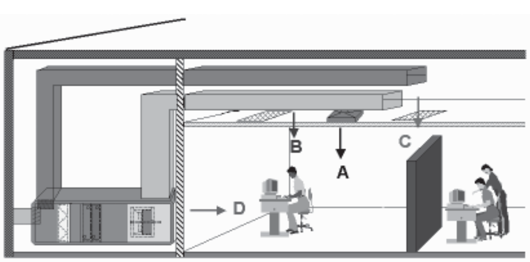

FIGURE 6 – CRITICAL SOUND PATHS

Consider the following four AHUs. If all four units generate the same CFM and all other components of the system are identical, which would produce the lowest discharge sound power? Which would produce the lowest casing radiated sound power?

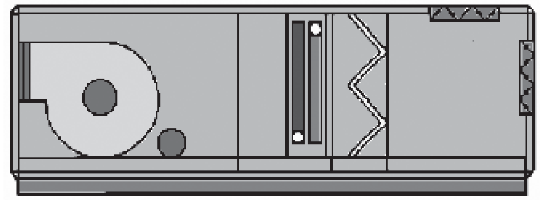

FIGURE 7 – 20" FC FAN

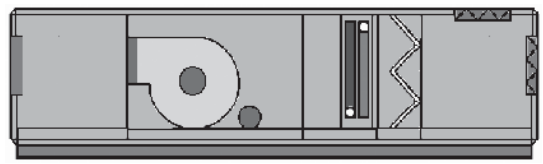

FIGURE 8 – 20" FC FAN WITH DISCHARGE PLENUM

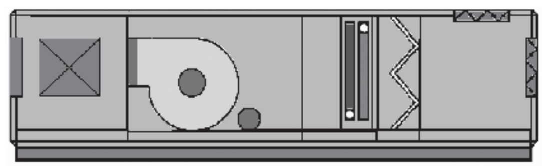

FIGURE 9 – 20" FC FAN W/DUAL OUTLET DISCH. PLENUM

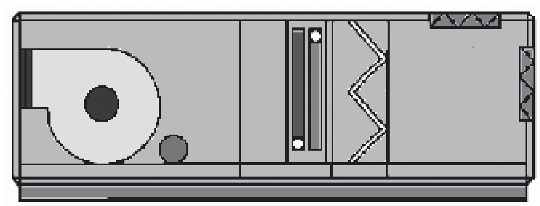

FIGURE 10 – 22" FC FAN

Actual testing in accordance with the procedures listed in ARI-260 indicates that the AHU illustrated in Figure 9, produced the lowest discharge sound power levels due to the attenuating effect of the side-outlet discharge plenum. The AHU in Figure 7 produced the highest discharge sound power levels.

Casing radiated sound tests yielded different results. The AHU in Figure 10 generated the lowest radiated sound power levels because the 22" FC fan had the lowest fan sound power. Ironically, the AHU in Figure 9 generated the highest casing radiated sound power. That's because the side-outlet discharge plenum, while adding enough attenuation to the discharge path to surpass the lower bare fan PWL, also added static pressure to the system. The additional static pressure caused the 20" fan in this unit to work harder, and therefore produce more sound than the other two 20" fans.

If the unit in this example were located next to a noise sensitive area such as a conference room, the AHU in Figure 10 would probably be selected to minimize the noise radiated to the space. On the other hand, if the unit was located next to an elevator shaft or in an equipment room with heavy masonry walls, the unit in Figure 9 might be selected to minimize the discharge sound power levels. Determining the critical path could make all the difference between an acoustically sound job, and expensive field retrofits.

Source Optimization vs. Path Attenuation

There are two choices when it comes to acoustic design – a quiet fan with no attenuation (source optimization) or a loud fan with heavy attenuation (path attenuation).

Source optimization is the ideal choice for a number of reasons. It is difficult to attenuate sound in the 63 through 500 Hz octave bands due to the wavelength of sound at those frequencies. By selecting a fan that is relatively quiet in those octave bands, the need for costly sound attenuators can be reduced or eliminated. Another advantage of source optimization is that selecting a fan with lower sound power levels acts to simultaneously attenuate all paths, whereas a louder fan may require attenuation along multiple paths. Finally, it's typical that the quieter a fan is, the more efficiently it is running. The more efficient a fan is for a given operating condition, the lower its brake horsepower, and therefore its operating cost is reduced. When combined with the elimination of sound attenuators, this reduction in operating costs more than offsets any higher first cost of the more efficient fan.

So, does that mean that path attenuation should never be used? No, path attenuation is effective in the 500 through 8000 Hz octave bands. Also, sound attenuators for frequencies in this range are smaller, and therefore less costly than those required for low frequency attenuation. Also, as we saw in the previous example, discharge plenums can provide relatively low cost sound attenuation for the supply path, particularly where air turbulence from the fan discharge would otherwise cause rumbling in the ductwork. Finally, path attenuation should be used where no other option exists; e.g. on an existing project where replacing the fan would cost more than installing attenuators.

General Guidelines for Fans and AHUs

Selecting the most efficient fan for the application is important, but selecting the proper type of fan based on static pressure and unit configuration is also important. Every fan has its own "sweet spot" and the designer should be familiar with them. For example, vane axial fans generate relatively low sound power in the first few octave bands, but the tonal sounds they can generate at higher frequencies can often be more objectionable than the broadband noise produced by centrifugal fans.

Regardless of the type of fan used, it should be selected at a good operating point. Fans that operate on the left side of the fan curve, such as the one illustrated in Figure 11, are in the stall region. This is a region of unstable airflow that can cause control and balancing problems as well as objectionable "surge" and "roar" sounds in the system. Fans should always be selected to operate on the right side of the fan curve as shown in Figure 12.

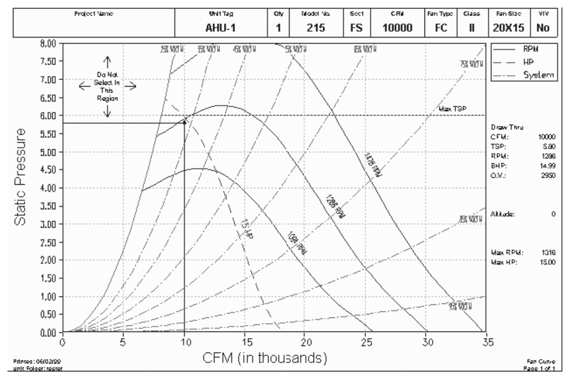

FIGURE 11 – POOR FAN SELECTION

When comparing fans it is also important to make an apples-to-apples comparison. The following factors can have an effect on the sound output of two similar-appearing fans:

- Outlet shape – square vs. rectangular.
- Wheel diameter.
- Outlet velocity.
- Number of fan blades.
- Fan speed – RPM.

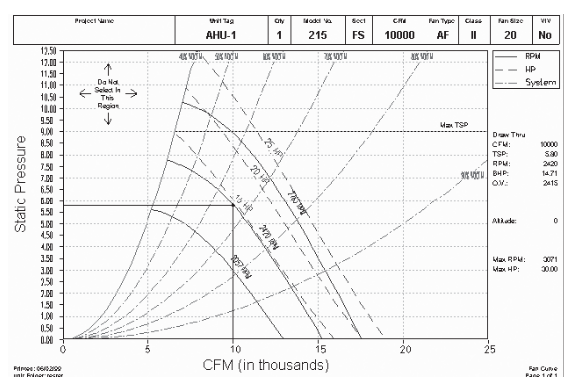

FIGURE 12 – GOOD FAN SELECTION

Placing a fan in an air-handling unit cabinet alters the acoustical characteristics of the fan. Improper cabinet design or sizing can result in poor acoustical performance. When possible, there should be a minimum clear space equal to ¾ of the fan wheel diameter at any unducted fan inlet, and a minimum clear space equal to 1-1/2 fan wheel diameter at any unducted fan discharge. These clear spaces allow the airflow patterns to even out, reducing air turbulence and the rumbling noise associated with it.

Duct transitions, particularly at the unit inlet and discharge openings should be gradual, 15° maximum to minimize air pressure drop and turbulence. Duct fittings such as elbows, take-offs and tees should also be designed for smooth transitions and low pressure drop. Refer to SMACNA guidelines or York International Form 100.00-AG2, *Acoustichecks for Air Handling Projects* for more information on proper duct design.

Rooftop units are especially susceptible to poor acoustical design. It is always recommended on a rooftop unit that a discharge plenum follow the fan. The duct fitting below the roofline should include as gradual a transition as possible, and should be as heavy a gauge as possible. This will minimize breakout noise, turbulent air, and oil canning. Round ductwork is especially resistant to oil canning and low frequency noise breakout and should be considered if this is the critical path to attenuate.

COMMON ACOUSTICAL ERRORS

The following is a list of design errors that can lead to poor acoustic performance, beginning with the three most common:

- **Improper or inefficient fan selection.**
- **Insufficient clearance at fan inlet.**
- **Duct fitting or sound attenuator too close to fan inlet or outlet.**
- Inadequate vibration isolation.
- Poor control system operation, causing fan instability.
- Adjustable pitch sheaves on motors > 5 hp.

IMPACT OF VALUE ENGINEERING

Cost cutting often leads to acoustical problems. A few common value engineering proposals and their potential acoustical impacts are shown in Table 4.

Table 4 – Acoustic Impact of Value Engineering

Value Engineering	Potential Acoustical Impact
Use smaller fan running at higher RPM	Higher PWL due to fan inefficiency and higher outlet velocity
Reduce duct cross sectional area	Higher PWL due to higher duct velocity
Reduce gauge thickness of duct walls	Greater duct flexibility makes duct rumble more likely. More breakout sound through thinner duct walls.
Use inlet vanes in lieu of VFDs on fans	Higher PWL at all conditions due to system effect of inlet vanes
Use ductboard in lieu of sheetmetal ducts	Greater breakout of sound through duct walls
Drywall in lieu of masonry walls in equipment rooms	More low frequency sound radiated to adjacent spaces
Use cheaper VAV boxes or grills	Poor quality design or construction can lead to higher noise levels
Use neoprene pads in lieu of spring isolators on fans	More vibration and therefore more sound transmitted through structure
Reduce the number of VAV zones	Fewer larger VAV boxes produce more noise in the space

(Schaffer, M, 1991)

In Summary…

Remember, all jobs have sound requirements, although some may be more critical than others. Remember too that it is always more economical to design an HVAC system for good acoustical performance than to correct acoustical problems in the field. With that in mind, the following checklist should be followed on every job:

1. Always check the air-handling unit selection by reviewing the fan curves, outlet velocities and unit layout.
2. Avoid high fan discharge velocities and high duct velocities.
3. Determine the critical sound path(s).
4. Watch out for poor transitions off the unit and high static pressure duct fittings.
5. Always make an apples-to-apples comparison. Are the fans and AHUs the same? Is the sound data rated in accordance with ARI-260?
6. **On sound sensitive projects, *get the experts involved early!***

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