HVAC ACOUSTICAL APPLICATION GUIDELINES

Dan Int-Hout
Chief Engineer

Krueger Corporate Headquarters
1401 N. Plano Rd. • Richardson, TX 75081
Phone: (972) 680-9136 | Fax: (972) 497-0450 | E-Mail: kruegerinfo@krueger-hvac.com
www.krueger-hvac.com

OVERVIEW

The application of accepted acoustical factors to the known sound power of HVAC devices is a 'tried and trued' method of engineering for acceptable sound levels in spaces. The factors involved are better defined than at any time in the past, where tables were often provided without clearly referenced calculation paths, or critical sound path elements were missing entirely. The ARI Acoustical Application Standard 885-1998, 'Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets', is now available over the internet (www.ari.org). A new ASHRAE publication 'Application of Manufacturers' Sound Data' (Ebbing et.al, 1999) is available which describes in detail the acoustical parameters of many types of equipment. The latest ASHRAE Handbooks provide increasingly tested acoustical application data.

Air terminals are the most noise sensitive of all HVAC products since they are almost always mounted in or directly over occupied spaces, and are the primary focus of the 885 standard. They usually determine the residual background noise level from 125 Hz to 2,000 Hz. There are two types of 'Air terminals': those that control the amount of airflow in response to a signal to control zone temperature ('boxes'), and those that distribute or collect the flow of air (Grilles, Registers, & Diffusers, GRDs). On some occasions, the two functions are combined. As these two elements are the final components in many built-up air delivery systems and those closest to the building occupants, both are critical components in the acoustical design of a space. There is also a critical interplay between acoustics and the primary function of these devices; providing a proper quantity of well mixed air to the building occupants.

Other devices in the HVAC system are also contributors to the sound levels in spaces, and most of the application factors in ARI 885 apply to them as well. In addition, different terminal devices, such as fan coils, water source heat pumps, unit ventilators and air induction units also behave acoustically the same as 'boxes'. The sound from air handlers, rooftops and vertical pack units can also be applied to many of the acoustical factors found in ARI 885. What makes 885 so useful is the fact that all tables in the 1998 Standard have a clearly identified calculation procedure, allowing exact duplication of the tables in a computer program or spreadsheet.

In order to discuss these types of devices, we must understand the issues regarding sound levels in occupied spaces.

SOUND POWER & SOUND PRESSURE

The sound level in an occupied space, sound pressure, can be measured directly with a sound level meter, or estimated from published sound power after accounting for room volume and other acoustical factors. Sound level meters measure the sound pressure level at the microphone location. Estimation techniques calculate sound pressure level at a specified point in an occupied space. Measured sound pressure levels in frequency bands can then be plotted and analyzed, and compared with established criteria for room sound levels.

Sound power, on the other hand, cannot be measured directly (except using special Acoustic Intensity techniques) and is a measure of the acoustical energy created by a source. It is normally determined in special facilities and reported for devices under stated conditions. Sound Power Level (Lw) values for Air Terminal devices are usually reported as the sound power level in each of several octave bands with center frequencies as shown below. Sound Power Levels are given in decibels (dB) referenced to a base power in watts, typically 10–12 watts. Sound power levels can also be reported for full or 1/3 octave bands, but usually as full octave bands, unless pure tones (narrow bands significantly louder than adjacent bands) are present.

ARI and ASHRAE provide guidance in both measuring sound power levels, and in estimating the resultant room sound pressure. ASHRAE handbooks provide detailed acoustical guidance. ARI Standard 885 is an application standard which provides tables and equations for determining acoustical deducts based on the ASHRAE guides, as well as additional information provided by manufacturers.

OCTAVE BAND DESIGNATIONS

	Octave Band Designations							
Center Frequency	63	125	250	500	1000	2000	4000	8000
Band designation	1	2	3	4	5	6	7	8

Table 1: Octave Band

ARI Standard 885-90 provides details on how to calculate sound pressure levels in spaces. For proper design, the specifying engineer or acoustician must determine acceptable product radiated and discharge sound power levels in order to arrive at an acceptable space sound pressure level. This process may be accomplished either by starting with the room pressure levels and applying attenuation factors to determine the maximum allowed sound power, or by taking a product sound power level and determining the sound pressure level that will result in the space. This document will outline the later process. There are programs available to perform the reverse analysis (see references).

NC – NOISE CRITERIA

NC curves were developed to represent lines of equal hearing perception in all bands and at varying sound levels, and allow a single number rating of that sound spectrum. The highest recorded sound pressure in any given frequency band determines the single number NC rating. This is called a “tangent” process. Sound Pressure Levels (Lp) are measured directly by sound level meters at one or more points in a room. A product’s estimated Sound Pressure Level (Lp) performance curve can be estimated by subtracting space (or other appropriate) sound attenuation effects from the unit’s reported sound power (Lw). Currently, most Air Outlet and Inlets (GRDs) sound performance is reported by subtracting a 10-dB attenuation from all octave band sound power levels, and determining the NC rating. This room effect approximates a 3,000 cu. ft. room, 10 ft. from the source, for a diffuser, whose sound levels peak in the mid frequencies. (This is defined in the ASHRAE / ARI Standard 885 space effect calculation described below). VAV boxes require a more detailed analysis. Most air terminal products are currently specified and reported as single number NC ratings.

SPACE EFFECT

The use of a 10 dB “room effect”, while in common practice for predicting diffuser room sound levels for grilles and diffusers, and accepted for many years, is not as accurate a prediction as is possible using newer techniques. The ASHRAE Handbooks and ARI Standard 885 present an equation for determining the “space effect” (often referred as the ‘Shultz effect’, after the originator) based on both room volume and the distance from the observer to a point sound source:

Space Effect = (25) - 10 Log (ft.) - 5 Log (cu. ft.) - 3 Log (Hz)

Where:
ft. = Distance from observer to source
cu. ft. = Room volume
Hz = Octave band center frequency

This yields a range of deductions which differ in each octave band, as shown in Table 2 (next page).

Space Effect, Point Source									
Room Volume	Distance	Octave Band Mid Frequency							D 1.7
		63	125	250	500	1000	2000	4000	8000
2000 CuFt (55 m^3)	@ 5 Ft (1.5m)	-4	-5	-6	-7	-7	-8	-9	-10
	@ 10 Ft (3m)	-7	-8	-9	-10	-11	-11	-12	-13
	@ 15 Ft (4.6m)	-9	-10	-10	-11	-12	-13	-14	-15
2500 CuFt (70 m^3)	@ 5 Ft (1.5m)	-4	-5	-6	-7	-8	-9	-10	-11
	@ 10 Ft (3m)	-7	-8	-9	-10	-11	-12	-13	-14
	@ 15 Ft (4.6m)	-9	-10	-11	-12	-13	-14	-15	-15
3000 CuFt (85 m^3)	@ 5 Ft (1.5m)	-5	-6	-7	-7	-8	-9	-10	-11
	@ 10 Ft (3m)	-8	-9	-10	-10	-11	-12	-13	-14
	@ 15 Ft (4.6m)	-10	-10	-11	-12	-13	-14	-15	-16
5000 CuFt (140 m^3)	@ 5 Ft (1.5m)	-6	-7	-8	-9	-9	-10	-11	-12
	@ 10 Ft (3m)	-9	-10	-11	-12	-12	-13	-14	-15
	@ 15 Ft (4.6m)	-11	-12	-12	-13	-14	-15	-16	-17

Table 2: Space Effect

ARI STANDARD 885-98

ARI Standard 885-98, “Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminal and Air Outlets”, provides the most current application factors for converting rated sound power to a predicted room sound pressure level. It replaces the 1990 version, and has slightly different calculations for radiated sound, includes updated ASHRAE tables and a new Environmental Effect. The 1998 standard is the basis by which most air terminal manufacturers now convert sound power, as measured in reverberant rooms per ARI Standard 880-98, to a predicted room sound pressure level. The 885 standard provides a number of equations and tables available elsewhere, but puts them all in one document, and includes some unique tables as well. It also includes examples and diagrams to make the process easier to use. The 1998 version replaces the earlier standard as manufacturers’ update their catalogs. The most important elements of the standard are described here.

ENVIRONMENTAL ADJUSTMENT FACTOR

In order to use the 885 Standard, sound power must be corrected for differences between reverberant room and free field calibrations when reverberant rooms, such as specified by ARI 880, generate a sound power base. This Environmental Adjustment factor is listed in the ARI Standard 885. There is a slight difference between the factors presented in the 1990 and 1998 versions of the standard. Recent research conducted as part of an ASHRAE project (TRP 755) has led to a slight reduction in this factor.

In order to utilize sound power obtained in reverb rooms, an environmental adjustment factor must be applied to manufacturers’ data. This is“necessary because at low frequencies, all real occupied spaces behave acoustically more like reverberant rooms than open spaces (free field). In other words, manufacturers’ sound power data that is based on ILG/RSS with a free field calibration must be adjusted to match actual operating conditions found in the field. This applies to all participants in the ARI 880 certification program. For data gathered using ARI 880 (or the referenced method of test, ASHRAE 130), the environmental adjustment factor must be subtracted from the manufacturers’ sound power level data in order to use the adjustments provided in ARI Standard 885. This is true for all sound power data obtained in reverberant rooms. The old and the new factors are shown below in Table 3.

	Octave Band Center Frequency						
	125	250	500	1000	2000	4000	8000
ARI 885-90	3	2	1	1	1	1	1
ARI 885-98	2	1	0	0	0	0	0

Table 3: Environmental Adjustment Factor

DETERMINING ACCEPTABLE RADIATED SOUND POWER LEVELS

To predict the radiated sound power levels in a space for a project, the attenuation from both the space effect and the plenum/ceiling combination must be subtracted from the unit sound power levels in each octave band. It was noticed that estimations of sound pressure using the 1990 version of the 885 standard were often optimistic when evaluated in real spaces. Following ASHRAE sponsored research, the Standard was modified in 1998. The technique is different in the 1998 version of the standard than it was in the earlier version. Both methods will be discussed, as most manufacturers currently utilize the earlier standard in their product performance catalogs.

RADIATED SOUND, ARI 885-90

The 1990 version of ARI 880 uses a combination of Space Effect and a combined Plenum/Ceiling effect to predict the total attenuation from a sound source located in a plenum above a room.

SPACE EFFECT

The space effect attenuation is determined by the space effect equation, defined previously. From a design perspective, the worst case would be for a receiver who is typically located 5 ft. above the floor, and within a few feet of the source. The sound source in this case would be a terminal located in the ceiling plenum. The distance factor will equal the distance from the receiver to the terminal, in a direct line. This distance is seldom less than 5, or more than 15, ft. from the noise source. Most manufacturers' catalog data, however, assumes a distance of 10 feet, with a relatively large room. Space effect is determined using table 2 (or the associated equation).

PLENUM/CEILING EFFECT

Attenuation for the plenum and ceiling are combined due to the difficulty of separating these for evaluation. Experience has shown that the STC rating (Sound Transmission Class) for ceiling tiles, which is based on a two room pair test, is poorly correlated with observed data for a noise source located above a ceiling (and should never be used!). The ARI 885-90 Plenum Ceiling Effect Table B-12 (Table 4, below) is derived from a number of manufacturers' observations, and is only found in the ARI Standard. This table assumes that the plenum space is at least 3 ft. deep, is over 30 ft. wide or lined with insulation, and that there are no penetrations directly under the unit.
From the 1990 ARI Standard, the following attenuation values, or transfer functions, should be used for the plenum/ceiling effect.

Ceiling/Plenum Effect Attenuation Values, Radiated sound						
Tile Type	125	250	500	1000	2000	4000
1, Mineral Fiber	9	10	12	14	15	15
2, Glass Fiber	8	8	8	10	10	14
3, Solid Gypsum Board	15	21	25	27	26	27

Table 4: ARI 885-90 Table B-12: Plenum/Ceiling Cavity Effect, dB.

Once the space effect and the plenum/ceiling effect have been determined, they are subtracted from the sound power level to determine the room sound pressure levels estimated for an observer in the space. This must be done for each octave band.

Eq 1a: Lp = Lw RAD – S – P/C – Env

Where:
Lp = Predicted Room Sound Pressure Level
Lw RAD = Unit Radiated Sound Power Level
S = Space Effect
P/C = Plenum/Ceiling Effect
Env = Environmental Effect

RADIATED SOUND, ARI 885-98

The report of the recently completed ASHRAE Research Study TRP 755 'Sound Transmission Through Ceilings from Air Terminal Devices in the Plenum' has added significantly to the understanding of acoustical effects of sound sources located in ceiling plenums. It is expected that these values will apply to many sound sources located in plenums, in addition to air terminals. The report recommended that some changes should be made to ARI Standard 885-1990 in order to bring the Standard into agreement with the results of the research project.

In addition to the recommended change in the environmental Effect discussed earlier, the report recommended a significant change in the prediction of radiated sound from plenum located sound sources. The TRP-755 study showed that there is a potential problem when applying the space effect 'draw away' (Table 2, above) with plenum located sound sources. Rather, the study suggests that plenum located sources are an area source, and that distance is not as significant as previously thought. As a result, a single table was proposed, which has been adopted in the 1998 revision of Standard 885.

Type	Tile Type	Density		Thickness		Weight		Octave Band Mid Frequency						
#		kg/m^3	lb/f^3	mm	in.	kg/sqm	Lb/Sf	63	125	250	500	1000	2000	4000
1	Mineral Fiber	313	20	16	0.63	5		13	16	18	20	26	31	36
2	Mineral Fiber	156	10	16	0.63	2.5		13	15	17	19	25	30	33
3	Glass Fiber	44	3	16	0.63	0.7		13	16	15	17	17	18	19
4	Glass Fiber	60	4	50	1.969	3		14	17	18	21	25	29	35
5	Glass Fiber, TL Backed	60	4	50	1.969	3		14	17	18	22	27	32	39
6	Gypsym Board Tiles	692	43	13	0.512	9		14	16	18	18	21	22	22
7	Solid Gypsum Board	692	43	13	0.512	9		18	21	25	25	27	27	28
8	Solid Gypsum Board	688	43	16	0.63	11		20	23	27	27	29	29	30
9	Double Gypsum Board	720	45	25	0.984	18		24	27	31	31	33	33	34
10	Double Gypsum Board	688	43	32	1.26	22		26	29	33	33	35	35	36
11	Concealed Spline	313	20	16	0.63	5		29	23	24	24	29	33	34

Data from ASHRAE TRP 755.

Table 5: ARI 885-98 Combined Ceiling/Space Effect Table

The values in table 5 are to be used, in conjunction with the 885-98 environmental effect values, for predicting sound levels for devices located in a ceiling plenum cavity.

Once the combined ceiling/space effect has been determined, they (and the environmental adjustment factor) are subtracted from the sound pressure level to determine the room sound pressure levels. This must be done for each octave band.

Eq 1b: Lp = Lw RAD – S – P/S – Env

Where:
Lp = Predicted Room Sound Pressure Level
Lw RAD = Unit Radiated Sound Power Level
P/S = Plenum/Ceiling Effect

Env = Environmental Effect

The effect of these changes to ARI 885 has been to increase the estimated radiated sound pressure NC rating by about 5 NC, for manufacturers who previously used larger room volume and distances in predicting room NC levels from the 1990 version of the Standard (almost all manufacturers used at least a 5000 cuft room, @10 ft distance). These resultant higher NC levels, however, should be able to be used directly, without a 'rule of thumb' addition. (Should another rating system be adopted in place of NC, this too will be able to be used without additional 'safety factors'). It will also bring the observed results in mock-ups in line with NC estimations presented in manufacturer's catalogs.

DETERMINING ACCEPTABLE DISCHARGE SOUND POWER LEVELS

Discharge sound (sometimes called airborne sound) is the sound that travels down the duct and discharges into the room along with the conditioned air. The procedure for determining the maximum acceptable discharge sound power levels requires the addition of the space effect, end reflection, duct insertion, flow division (or branch power division), and elbow and tees, or some combination of these, to the maximum acceptable room sound pressure levels. If more than one outlet supplies air to a room, separate evaluations should occur for each discharge path. This is done for each octave band.

SPACE EFFECT

The discharge sound space effect is determined in the same manner as the radiated sound space effect (from 885-90). The sound source in this case might be the outlet (i.e., grille, register, or diffuser) supplying air to the space, or may be sound from an upstream noise source (damper or fan) which passes through the outlets, or a sum of both. (Table 2)

END REFLECTION

When the area across the airstream expands (or contracts) suddenly as the ductwork terminates or ends at the outlet to the occupied space (or enters a branch fitting), a significant amount of low frequency sound is reflected back into the ductwork. This is called end reflection. The amount of end reflection varies based on the inlet size and type of duct. (The end reflection table changed slightly in the 1998 version of ARI 885, as a different reference was used, based on an ISO standard.)

Mean Duct Width		Octave Band Mid Frequency				
in.	**mm.**	63	125	250	500	1000
6	[150]	17	12	6	3	1
8	[200]	15	9	5	2	0
10	[250]	13	8	3	1	0
12	[300]	12	6	3	1	0
16	[400]	9	5	2	0	0
20	[500]	8	3	1	0	0
24	[600]	6	3	1	0	0
28	[700]	5	2	1	0	0
32	[800]	5	2	0	0	0
36	[900]	4	1	0	0	0
48	[1200]	3	1	0	0	0
72	[1800]	1	0	0	0	0

Table 6: End Reflection from Table B-11, ARI 885-98, dB.

Krueger Corporate Headquarters
1401 N. Plano Rd. • Richardson, TX 75081
Phone: (972) 680-9136 | Fax: (972) 497-0450 | E-Mail: kruegerinfo@krueger-hvac.com
www.krueger-hvac.com

DUCT INSERTION LOSS

The addition of lined ductwork results in significant attenuation of higher frequency sound. The amount of attenuation varies with duct size and lining thickness. ARI 885 contains several tables helpful in determining the appropriate attenuation values. Each section of ductwork inserted downstream of the terminal must be evaluated. For example, one might have separate duct insertion attenuation values for straight lined discharge duct, branch duct (if lined), and flex duct from the branch to the outlet. The ARI Standard, as well as the 1999 Systems ASHRAE Handbook, provides tables of insertion loss per foot of duct based on inside duct dimensions and insulation thickness. (Note – the 1995 ASHRAE Systems Handbook tables should not be used, as they have subsequently been found to be in error for small duct sizes!)

TABLE D8. Sound Attenuation in Straight Lined Sheet Metal Ducts of Rectangular Cross Section in dB/ft [dB/0.3m] Lining Thickness: 1in.[25mm]: No Airflow. Db/ft [dB/0.3m]

Dimensions		Octave Band Center Frequency (Hz)						
in x in	mm x mm	125	250	500	1000	2000	4000	8000
6 x 6	150 x 150	0.6	1.5	2.7	5.8	7.4	4.3	2.0
6 x 10	150 x 250	0.5	1.2	2.4	5.1	6.1	3.7	1.9
6 x 12	150 x 300	0.5	1.2	2.3	5.0	5.8	3.6	1.9
6 x 18	150 x 460	0.5	1.0	2.2	4.7	5.2	3.3	1.9
8 x 8	200 x 200	0.5	1.2	2.3	5.0	5.8	3.6	1.9
8 x 12	200 x 300	0.4	1.0	2.1	4.5	4.9	3.2	1.8
8 x 16	200 x 410	0.4	0.9	2.0	4.3	4.5	3.0	1.8
8 x 24	200 x 460	0.4	0.8	1.9	4.0	4.1	2.8	1.8
10 x 10	250 x 250	0.4	1.0	2.1	4.4	4.7	3.1	1.8
10 x 16	250 x 300	0.4	0.8	1.9	4.0	4.0	2.7	1.8
10 x 20	250 x 410	0.3	0.8	1.8	3.8	3.7	2.6	1.7
10 x 24	250 x 760	0.3	0.7	1.8	3.7	3.5	2.5	1.7
12 x 12	300 x 300	0.4	0.8	1.9	4.0	4.1	2.8	1.8
12 x 18	300 x 460	0.3	0.7	1.7	3.7	3.5	2.5	1.7
12 x 24	300 x 610	0.3	0.6	1.7	3.5	3.2	2.3	1.7
12 x 36	300 x 910	0.3	0.6	1.6	3.3	2.9	2.2	1.7
15 x 15	380 x 380	0.3	0.7	1.7	3.6	3.3	2.4	1.7
15 x 22	380 x 560	0.3	0.6	1.6	3.3	2.9	2.2	1.7
15 x 30	380 x 760	0.3	0.5	1.5	3.1	2.6	2.0	1.6
15 x 45	380 x 1140	0.2	0.5	1.4	2.9	2.4	1.9	1.6
18 x 18	460 x 460	0.3	0.6	1.6	3.3	2.9	2.2	1.7
18 x 28	460 x 710	0.2	0.5	1.4	3.0	2.4	1.9	1.6
18 x 36	460 x 910	0.2	0.5	1.4	2.8	2.2	1.8	1.6
18 x 54	460 x 1370	0.2	0.4	1.3	2.7	2.0	1.7	1.6
24 x 24	610 x 610	0.2	0.5	1.4	2.8	2.2	1.8	1.6
24 x 36	610 x 910	0.2	0.4	1.2	2.6	1.9	1.6	1.5
24 x 48	610 x 12320	0.2	0.4	1.2	2.4	1.7	1.5	1.5
24 x 72	610 x 1830	0.2	0.3	1.1	2.3	1.6	1.4	1.5
30 x 30	760 x 760	0.2	0.4	1.2	2.5	1.8	1.6	1.5
30 x 45	760 x 1140	0.2	0.3	1.1	2.3	1.6	1.4	1.5
30 x 60	760 x 1520	0.2	0.3	1.1	2.2	1.4	1.3	1.5
30 x 90	760 x 2290	0.1	0.3	1.0	2.1	1.3	1.2	1.4
36 x 36	910 x 910	0.2	0.3	1.1	2.3	1.6	1.4	1.5
36 x 54	910 x 1370	0.1	0.3	1.0	2.1	1.3	1.2	1.4
36 x 72	910 x 1830	0.1	0.3	1.0	2.0	1.2	1.2	1.4
36 x 108	910 x 2740	0.1	0.2	0.9	1.9	1.1	1.1	1.4
42 x 42	1070 x 1070	0.2	0.3	1.0	2.1	1.4	1.3	1.4
42 x 64	1070 x 1630	0.1	0.3	0.9	1.9	1.2	1.1	1.4
42 x 84	1070 x 2130	0.1	0.2	0.9	1.8	1.1	1.1	1.4
42 x 126	1070 x 3200	0.1	0.2	0.9	1.7	1.0	1.0	1.4
48 x 48	1220 x 1220	0.1	0.3	1.0	2.0	1.2	1.2	1.4
48 x 72	1220 x 1830	0.1	0.2	0.9	1.8	1.0	1.0	1.4
48 x 96	1220 x 2440	0.1	0.2	0.8	1.7	1.0	1.0	1.3
48 x 144	1220 x 3660	0.1	0.2	0.8	1.6	0.9	0.9	1.3

*Based on measurements of surface-coated duct liners of 1.5 lb/ft^3 [24 Kg/m^3]density. Liner density has a minor effect over the range of 1.5 to 3lb/ft^3 [24-48 Kg/m^3] .

Reprinted with the permission of the Americal Society of Heating Refrigerating and Air Conditioning Engineers. 1999 ASHRAE Applications Handbook, Chapter 46, Table 8.

Table 7: Rectangular, 1" Lined Duct, dB / ft. from ARI 885-98.

FLEXIBLE DUCT INSERTION LOSS

The effect of flex duct is significant, more than an equivalent amount of lined duct, especially in the critical low frequencies. Data from a number of manufacturers conducted at an independent laboratory (in accordance with ASTM procedures) was collated, and found to be surprisingly similar. This data was presented in a table in the ARI 1990 Standard. The table was subsequently 'tweaked' to allow it to be computed with an algorithm, and presented in ARI 885-98 (along with the equation and constants):

Duct Diameter Inches [mm]	Length L Feet [m]	Insertion Loss dB Octave Band Mid Frequency							
		63	125	250	500	1000	2000	4000	8000
4 [100]	10 [3.0]	8	10	9	27	33	38	24	17
	5 [1.5]	4	7	5	16	24	27	18	11
	3 [0.9]	3	5	4	12	20	23	15	9
5 [125]	10 [3.0]	8	10	12	28	33	37	23	15
	5 [1.5]	4	6	7	17	23	25	16	10
	3 [0.9]	3	5	5	13	19	21	13	8
6 [150]	10 [3.0]	8	10	15	28	33	35	22	13
	5 [1.5]	4	6	9	18	22	24	15	9
	3 [0.9]	3	5	6	13	17	19	11	7
7 [175]	10 [3.0]	8	10	16	29	33	34	21	12
	5 [1.5]	4	6	10	18	21	22	13	8
	3 [0.9]	3	5	7	14	16	17	10	6
8 [200]	10 [3.0]	8	10	18	29	32	32	20	10
	5 [1.5]	4	6	10	18	20	21	12	7
	3 [0.9]	2	4	7	14	15	16	8	6
9 [225]	10 [3.0]	7	9	18	28	32	31	19	9
	5 [1.5]	4	6	11	18	19	19	10	6
	3 [0.9]	2	4	8	14	13	14	7	5
10 [250]	10 [3.0]	7	9	19	28	31	29	18	8
	5 [1.5]	3	5	11	18	18	18	9	5
	3 [0.9]	2	4	7	14	12	13	6	4
12 [300]	10 [3.0]	6	8	17	26	29	26	15	7
	5 [1.5]	3	4	9	16	16	15	7	4
	3 [0.9]	1	3	6	12	10	11	4	3
14 [350]	10 [3.0]	4	6	13	23	26	23	12	6
	5 [1.5]	2	3	7	14	14	13	6	4
	3 [0.9]	1	2	4	10	9	9	4	3
16 [400]	10 [3.0]	2	4	7	19	24	20	8	6
	5 [1.5]	0	2	2	11	12	11	5	3
	3 [0.9]	0	1	0	8	8	8	4	2

Data based on solid core (non-perforated or woven), 1" insulation, and plastic jacket
These data are compiled from several sources and should therefore be used as a guide.

Table 8: Flexible Duct Insertion Loss

OTHER DISCHARGE FACTORS

Flow Division. When the airstream is divided, the sound carried in each downstream branch is less than the sound upstream of the branch take-off. This shows the percent of total airflow carried by the branch. The appropriate level of attenuation can then be determined from a simple equation:

Eq 2: $$dB = 10 \text{ Log}(N)$$

Where: dB = Octave Band Sound Reduction
N = Number of Duct Splits

Elbows and Tees. A certain amount of attenuation of higher frequency sound is gained when an airstream enters an elbow or tee duct connection. The attenuation is considered by ARI Standard 885–90 to be negligible If the elbow is round and unlined. Attenuation of rectangular tees is determined by treating the tee as two elbows placed side by side. The tables below indicate the approximate attenuation of 90° elbows with and without turning vanes.

Insertion Loss of Unlined and Lined Elbows W/O turning vanes.
From ASHRAE Applications, 1999, Chap 46, Table 13
Unlined Duct

Width		Octave Band Mid Frequency							
in.	cm	63	125	250	500	1000	2000	4000	8000
5-10	10-25	0	0	0	1	5	8	4	3
11-20	26-70	0	1	5	5	8	4	3	3
21-40	71-100	0	5	5	8	4	3	3	3
41-80	101-200	1	5	8	4	3	3	3	3

Lined Duct

Width		Octave Band Mid Frequency							
in.	cm	63	125	250	500	1000	2000	4000	8000
5-10	10-25	0	0	0	1	6	11	10	10
11-20	26-70	0	1	6	6	11	10	10	10
21-40	71-100	0	6	6	11	10	10	10	10
41-80	101-200	1	6	11	10	10	10	10	10

Insertion Loss of Unlined and Lined Elbows With turning vanes.
From ASHRAE Applications, 1999, Chap 46, Table 15
UnLined Duct

Width		Octave Band Mid Frequency							
in.	cm	63	125	250	500	1000	2000	4000	8000
5-10	10-25	0	0	0	1	4	6	4	4
11-20	26-70	0	1	4	6	4	4	4	4
21-40	71-100	0	4	6	6	4	4	4	4
41-80	101-200	1	4	6	6	4	4	4	4

Lined Duct

Width		Octave Band Mid Frequency							
in.	cm	63	125	250	500	1000	2000	4000	8000
5-10	10-25	0	0	0	1	4	7	7	7
11-20	26-70	0	1	4	7	7	7	7	7
21-40	71-100	0	4	7	7	7	7	7	7
41-80	101-200	1	4	7	7	7	7	7	7

Table D11. Attenuation of Unlined Round Elbows
Radiused Elbow (90°), dB

Diameter		Octave Band Mid Frequency							
in.	cm	63	125	250	500	1000	2000	4000	8000
5-10	10-25	0	0	0	1	2	3	3	3
11-20	26-70	0	1	2	2	3	3	3	3
21-40	71-100	0	2	2	3	3	3	3	3
41-80	101-200	1	2	3	3	3	3	3	3

From ASHRAE Applications, 1999, Chap 46, Table 14

Table 9: Elbow tables from ARI 885-98.

Once all of the attenuation factors have been determined, they are subtracted from the device's sound power level to determine the sound pressure levels in the space. This must be done for each octave band, and again the Environmental Adjustment factor must be subtracted as well.

Eq 3: Lp = Lw DIS – S – ER – I – D – T/E – ENV

Where:

Lp	=	Sound Pressure Level
Lw DIS	=	Discharge Sound Power Level
S	=	Space Effect
ER	=	End Reflection
I	=	Duct Insertion
D	=	Flow Division
T/E	=	Tee/Elbow
ENV	=	Environmental Factor

DETERMINING ACCEPTABLE TOTAL SOUND IN A SPACE

Once the Radiated and Discharge sound pressure paths and effects are known, the resulting room sound level can be evaluated. Other factors may play a part in determining the final room sound levels. All these factors must be included to achieve an accurate prediction or analysis. The results may be very complex.

For a typical project, room sound levels can be computed using a simple spreadsheet. One is available on the Krueger website (www.krueger-hvac.com), named 885-calc.xls. A sample discharge sound analysis is shown in Figure 1. The top line represents reported sound power, the lower the predicted room sound pressure.

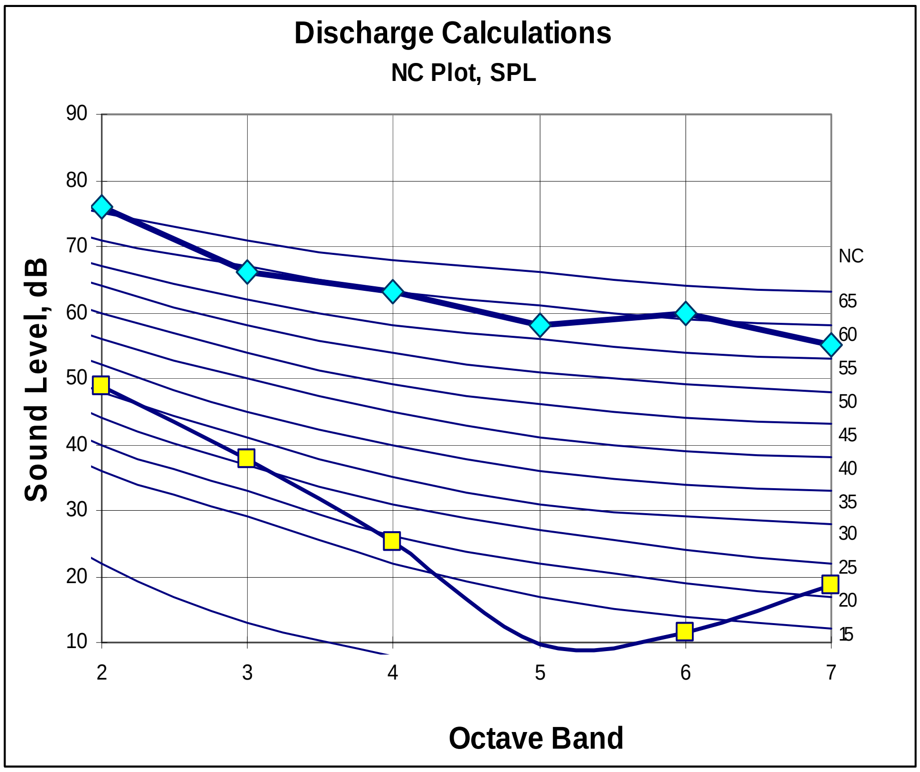

Figure 1: Discharge calculations

For a radiated analysis, using both the old (885-90) and new (885-98) methods, with a mineral tile ceiling, the analysis is shown in Figure 2.

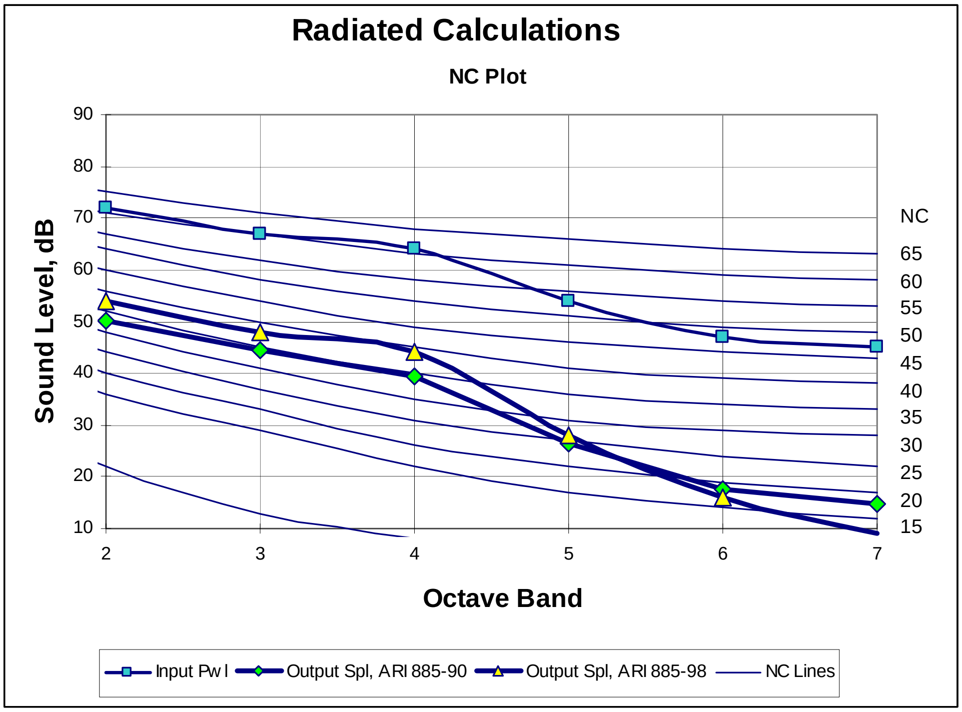

Figure 2: Radiated sound analysis

In this example, two paths are shown (radiated and duct borne discharge), but in practice, changes in designs may make previously insignificant path become predominant. As shown, radiated sound will predominate. But should duct lining be eliminated, and no flex duct employed, discharge sound may be much more important than radiated. Poor duct design may cause duct breakout to be the highest sound heard in the space.

The ARI 885 Standard provides guidance on all the possible paths. Not shown here is background sound, which is often at an NC–35 or greater in occupied spaces.

All the sound paths must be combined to predict the room sound level. When combining path elements, the math is done using log addition, not algebraically. Logarithmic (log) addition requires taking the antilog of the dB in each band, adding them together, and then taking the log of the answer. While this sounds complicated, figure 3 on the next page shows an easier way of estimating the result.

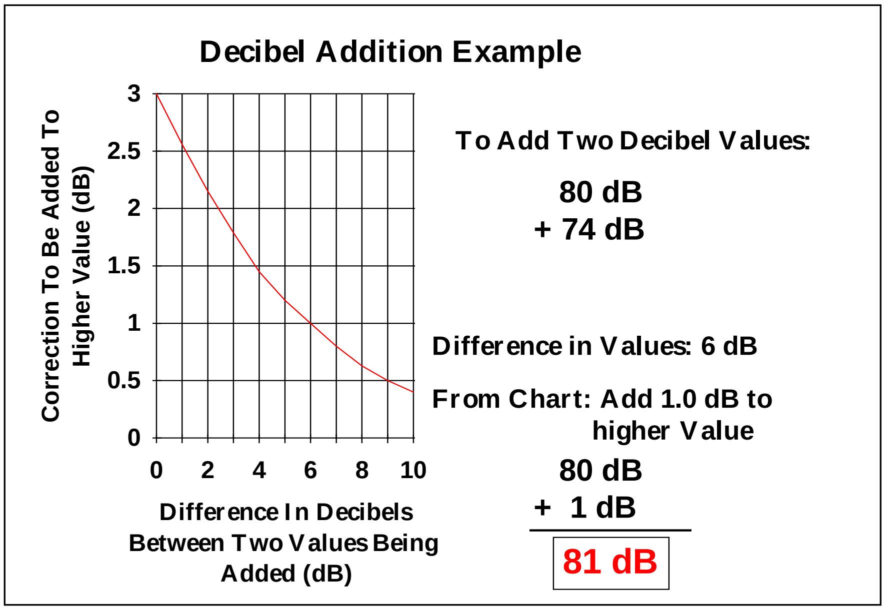

Figure 3: Log Addition

The referenced spreadsheet also has a room sound calculator that performs a three-way calculation combining Discharge, Radiated and diffuser sound levels to calculate a room sound pressure level.

NC vs. RC

Historically, most sound levels are both specified and reported as either dBA or NC. A dBA value is the average of all sound frequencies weighted against a standard curve, and is as a result essentially useless as a sound descriptor or diagnostic. NC (Noise criteria) is a better descriptor but has some shortcomings, especially as a predictor of speech privacy. The use of RC (room criteria) has been proposed as a better descriptor, and has been substituted for NC in recent ASHRAE handbooks.

In the mid frequencies, NC and RC result in similar ratings. RC has the advantage of an additional Letter (or Letters) to describe high (H) and low frequency rumble (R) and vibration (V) characteristics. Unfortunately, when used to rate the performance of VAV terminals, or any device that is predominant in the low frequencies, significant differences between products can be masked by this rating scheme. Other descriptors have also been proposed including RC MkII and NCB. These often require measurements in the 63 Hz or lower octave bands. The 20,000 CuFt reverb room required for these measurements isn't available, and VAV terminals do not produce significant sound levels in these lower frequencies, in any case.

As a result, VAV box manufacturers (and manufacturers of other devices containing fans) will likely continue to rate products in NC units. But room sound levels will be specified by acousticians in RC. The only way to realistically specify sound levels, therefore, is to specify maxim allowed sound power, by octave band. Maximum Sound Power can be determined by taking a desired room sound level and adding predicted attenuation elements to those values.

Figure 4 shows the combined Discharge, Radiated and Diffuser sound levels on both NC and RC graphs.

Room Sound Calculations

This page adds the sound from the VAV box in the previous examples to the sound from a selected diffuser

Project / Zone:							Comments:
Octave Band	2	3	4	5	6	7	
Mid Frequency, Hz	125	250	500	1000	2000	4000	
VAV Box Discharge Sound pressure	49	38	25	10	11	19	From Discharge Sheet
VAV Box Radiated Sound Pressure	54	48	44	28	16	9	From Radiated Sheet, 885-98
Diffuser Sound Power	50	45	44	46	44	27	User input, Sound Power (PWL) rating, from catalog or software
# of Diffusers	1						# of diffusers near listener
Room Attenuation	5	6	7	8	9	10	Room Absorption, from Discharge Sound Sheet
Other (Custom)	0	0	0	0	0	0	Optional User Input
Environmental Adjustment Factor	2	1	0	0	0	0	From ARI 885-98, Should always be used
Diffuser Sound Pressure	43	38	37	38	35	17	Diffuser Sound Power less attenuation factors
Total Room Sound Pressure	55	49	45	39	35	21	Log add Diffuser, Dischg and Rad sound pressures. db= 10*log((10^(Disch/10)+(10^(Rad/10))+10^(Diff/10)))
Output NC	40						
		R?	H?				
Output RC, Letters	40	N	N				RC can have an "H" or an "R" designation (Hiss or Rumble). N = Neutral.

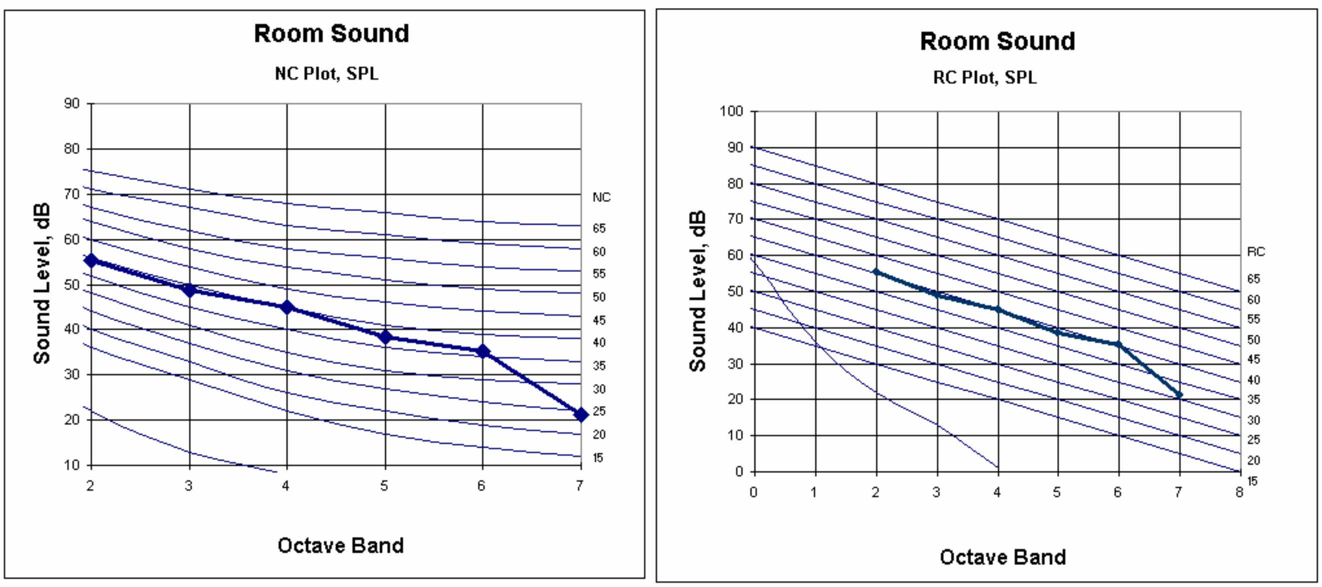

Figure 4: Combined radiated, discharge and diffuser sound

Sound from VAV boxes and diffusers combine to create the room sound pressure level. Since they peak in different bands, however, they often complement each other. In many cases, a Series Fan Terminal will be predicted to have an RC–35R in a space, but when combined with an RC–35N diffuser, will result in a room sound pressure level of RC–38N, which is optimum for providing speech privacy in open plan spaces.

Most people cannot differentiate between two sources that differ by less than 3 dB. If the background sound is an NC–35, and the device in question is predicted at an NC–35, it is likely that the space will be at an NC–38 (although this is dependent on which octave bands are critical), but most people cannot hear the difference.

Krueger Corporate Headquarters
1401 N. Plano Rd. • Richardson, TX 75081
Phone: (972) 680-9136 | Fax: (972) 497-0450 | E-Mail: kruegerinfo@krueger-hvac.com
www.krueger-hvac.com

SOUND DESIGN

Engineers can minimize the sound contribution of air terminals to an occupied space through good design practice. Whenever possible, terminals should be located over areas less sensitive to noise. This includes corridors, copy rooms, storage rooms, etc. Quiet air terminals facilitate the location of terminals over unoccupied space as with these units larger zones are possible resulting in fewer terminals. This also reduces first cost and improves energy efficiency. The use of lined duct work or manufacturers' attenuators downstream of air terminals can help attenuate higher frequency discharge sound. Flexible duct (used with moderation) is also an excellent attenuation element. Sound will be reduced when appropriate fan speed controllers are used to reduce fan rpm rather than using mechanical devices to restrict airflow. This form of motor control is often more energy efficient. The air terminal and the return air grille location should be separated as far as possible. Radiated sound can travel directly from the terminal through the return air grille without the benefit of ceiling attenuation.

Designing systems to operate at low supply air static pressure will reduce the generated sound level. This will also provide more energy efficient operation and allow the central fan to be downsized. Sharp edges and transitions in the duct design should be minimized to reduce turbulent airflow and its resulting sound contribution.

ARI 885-98 RECOMMENDATIONS

ARI 885-98 has provided recommended application assumptions for reporting of estimated room sound levels in Appendix E:

VAV Boxes:

Radiated Sound, Ceiling Plenum Noise Sources:

Total deduct from Sound Power to predict room Sound Pressure (Includes Environmental effect)
Assumes a 3ft Deep Plenum with non bounded sides.

	Octave Band Mid Frequency					
	125	250	500	1000	2000	4000
Type 1 - Glass Fiber	19	19	21	25	29	35
Type 2 - Mineral Fiber	18	19	20	26	31	36
Type 3 - Solid Gypsum Board	23	26	25	27	27	28

From Table D15

Discharge Sound, Noise Source in Occupied Space:

	Octave Band Mid Frequency						
	125	250	500	1000	2000	4000	8000
Small Box, 8 x 8 in duct (0.2 x 0.2 m)							
<300 cfm (<140 l/s)	25	28	38	53	58	31	28
Medium Box (12 x 12 Duct)							
300-700 cfm (140-330 l/s)	27	29	39	51	53	33	26
Large Box (15 x 15 Duct)							
>700 cfm (330 l/s)	29	30	40	51	51	35	29

From Table D 18

Table 10: Appendix E, ARI 885-98

The effect of these recommendations will be to raise almost all manufacturers' currently reported Radiated NC 'Application' sound levels by at least 5 NC. The effect on currently reported discharge sound levels will be minimal. Most manufacturers now use these recommended values to calculate application NC levels in their catalogs. In November 2002, ARI passed a rule requiring that the above assumptions be used to report estimated NC values when the ARI seal is shown on the same page. Until such a time as all manufacturers have adopted the newer (more conservative) levels, users should be especially careful in applying catalog NC values to projects.

SPECIFYING SOUND LEVELS

So far, we have discussed how to predict sound levels in a space, given a unit's octave band sound power level. Because of problems with NC and RC, what is needed is the opposite, a not-to-exceed sound power specification value. This can be accomplished by starting with a desired room sound pressure level, and then adding the expected acoustical deductions to that value, creating a maximum allowable sound power requirement. Below are some suggested room sound pressure levels:

	2	3	4	5	6	7
Suggested Room sound pressures	125	250	500	1000	2000	4000
High Speech Privacy	57	53	48	43	37	31
Low Speech Privacy	52	49	44	37	32	20
RC 40	60	55	45	40	35	33
NC40	55	50	44	41	39	38
NC35	52	45	40	36	34	32

Table 11: Room sound pressure examples

When the ARI 885-98 recommended Discharge Sound factors are applied to a high speech privacy sound requirement, the following results (from the spreadsheet 885-spec.xls, also available at the www.krueger-hvac.com website):

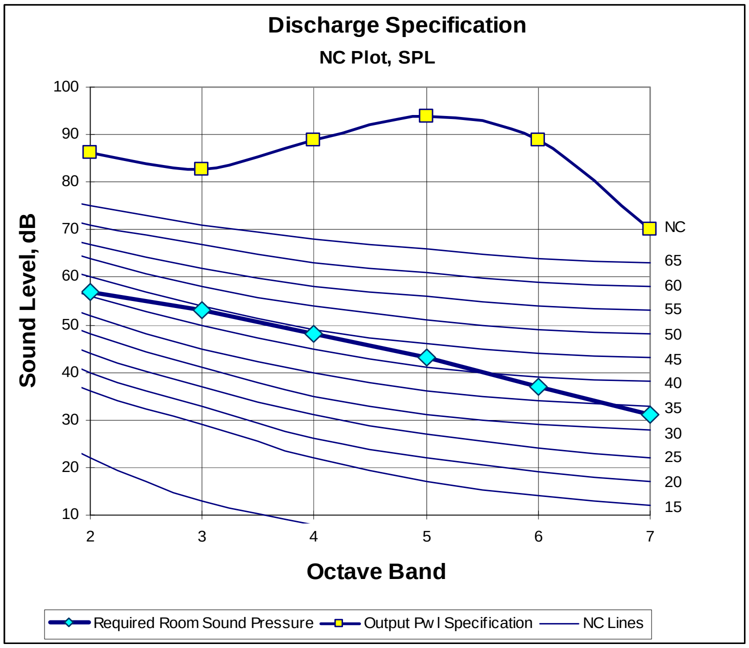

Figure 5: Discharge specification

For radiated sound, a similar plot is created:

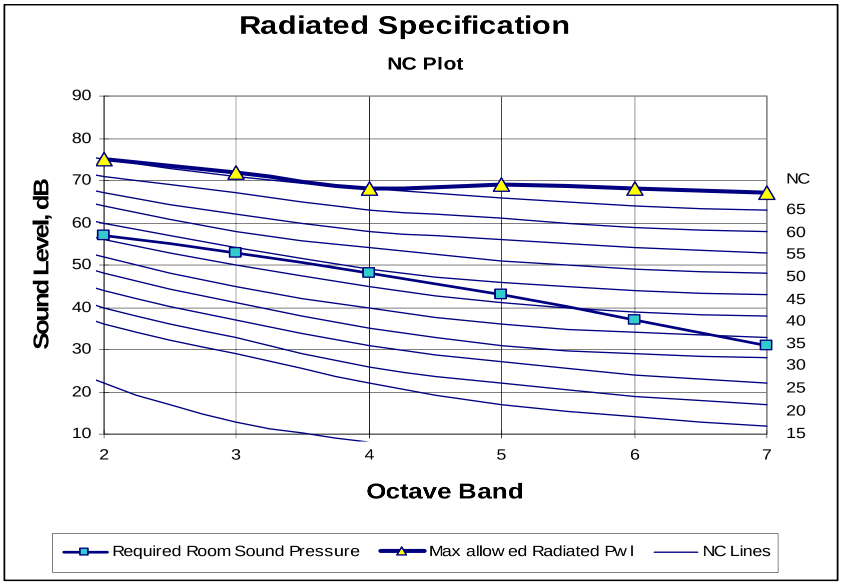

Figure 6: Radiated sound specification

A computer program, SoundSpec.exe is available at the Krueger website which will generate a text-file guide specification using these same calculations.

SUMMARY

The prediction of sound levels in a space which are the result of HVAC system component sound generation can be made with greater accuracy than ever before. The latest ARI 885 Standard (available at no charge from the ARI web site) has been shown to yield very accurate predictions in several manufacturers' mock-up facilities (controlled conditions) as well as in the field (where other acoustical contributions are more difficult to quantify). The latest ASHRAE handbooks, and the new ASHRAE book, also provide excellent information. The latest ARI 885 standard, moreover, provides a means of calculation of all included tables, allowing reproduction and system calculations in a spreadsheet or computer program.

Design engineers should utilize these new tools in determining the acoustical requirements of their projects, in order to ensure acceptable spaces at the lowest cost. Over-attenuation is expensive, and often reduces system performance. Noisy spaces, however, reduce space effectiveness and productivity. These methods allow a systematic, and proven, means of optimizing system design and product selection.

REFERENCES

ARI 885-1998, 'Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets', ARI, 1998 (WWW.ARI.ORG)

'Application of Manufacturers' Sound Data' , ASHRAE, Ebbing and Blazier, 1999 (WWW.ASHRAE.ORG)

Spreadsheet: 885-calc.XLS – Excel spreadsheet used to calculate main acoustical elements in the application of VAV terminals, from known sound power levels.

Program: CALC-NC.EXE – Computer program to perform the same functions of the spreadsheet above, with a graphical output.

Program: SounSpec.EXE – A 'back calculator' which runs the calculations in reverse, starting with desired room sound levels and producing a specification for maximum discharge and radiated sound power.