An Introduction to Sound Level Data for Mechanical and Electrical Equipment

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(Figures, tables and equations in this publication may at times be a little difficult to read, but they are the best available. ***DO NOT PURCHASE THIS PUBLICATION IF THIS LIMITATION IS NOT ACCEPTABLE TO YOU****.)*

1. INTRODUCTION. This course contains sound pressure and sound power data for mechanical equipment commonly found in many commercial buildings. Where possible, the noise data have been correlated with some of the more obvious noise influencing parameters, such as type, speed, power rating, and flow conditions. The noise levels quoted in this publication are suggested for design uses; these noise levels represent approximately the 80 to 90 percentile values. That is, on the basis of these sample sizes, it would be expected that the noise levels of about 80 to 90 percent of a random selection of equipment would be equal to or less than the design values quoted in the manual, or only about 10 to 20 percent of a random selection would exceed these values. This is judged to be a reasonable choice of design values for typical uses. Higher percentile coverage, such as 95 percent, would give increased protection in the acoustic design, but at greater cost in weight and thickness of walls, floors, columns, and beams. On-site power plants driven by reciprocating and gas turbine engines have specific sound and vibration problems, which are considered separately elsewhere.

2. SOUND PRESSURE AND SOUND POWER LEVEL DATA. In the collection of data, most noise levels were measured at relatively close-in distances to minimize the influence of the acoustic conditions of the room and the noise interference of other equipment operating in the same area.

2.1 NORMALIZED CONDITIONS FOR SPL DATA. Note: All measurements were normalized to a common MER condition by selecting a distance of 3 feet and a Room Constant of 800 ft^2 as representative. SPL data measured at other distances and Room Constants were brought to these normalized conditions by procedures of discussed elsewhere.

2.2 SOUND POWER LEVEL DATA. For equipment normally located and used outdoors, outdoor measurements were made and sound power level data are given. Usually, more measurements and a more detailed estimate of the measurement conditions were involved in deriving the PWL data, so they are believed to have a slightly higher confidence level than the normalized SPL data.

2.3 A-WEIGHTED SOUND LEVELS. In the tables and figures that follow, A-weighted sound levels are also given. Where sound pressure levels are given, the A-weighted sound level is in pressure; where sound power levels are given, the A-weighted value is in sound power. A-weighted sound levels are useful for simply comparing the noise output of competitive equipment. For complete analysis of an indoor or outdoor noise problem, however, octave band levels should be used.

2.4 MANUFACTURERS' NOISE DATA. Whenever possible, and especially for new types of equipment, the manufacturer should be asked to provide sound level data on the equipment. If the data show remarkably lower noise output than competitive models or are significantly lower than the data quoted in this publication, the manufacturer should be asked to give guarantees of the noise data and to specify the conditions under which the data were measured and/or computed.

3. PACKAGED CHILLERS WITH RECIPROCATING COMPRESSORS. These units range in size from 15-ton to 200-ton cooling capacity. The noise levels have been reduced to the normalized 3 foot distance from the acoustic center of the assembly. In terms of noise production, the measured compressors are divided into two groups: up to 50 tons and over 50 tons. The suggested 80- to 90-percentile noise level estimates are given in Figure 1 and in Table 1 for the two size ranges selected. Although major interest is concentrated here on the compressor component of a refrigeration machine, an electric motor is usually the drive unit for the compressor. The noise levels attributed here to the compressor will encompass the drive motor most of the time, so these values are taken to be applicable to either a reciprocating compressor alone or a motor-driven packaged chiller containing a reciprocating compressor.

4. PACKAGED CHILLERS WITH ROTARY-SCREW COMPRESSORS. The octave band sound pressure levels (at 3 foot distance) believed to represent near-maximum noise levels for rotary-screw compressors are listed in Table 2. These data apply for the size range of 100- to 300-ton cooling capacity, operating at or near 3600 RPM.

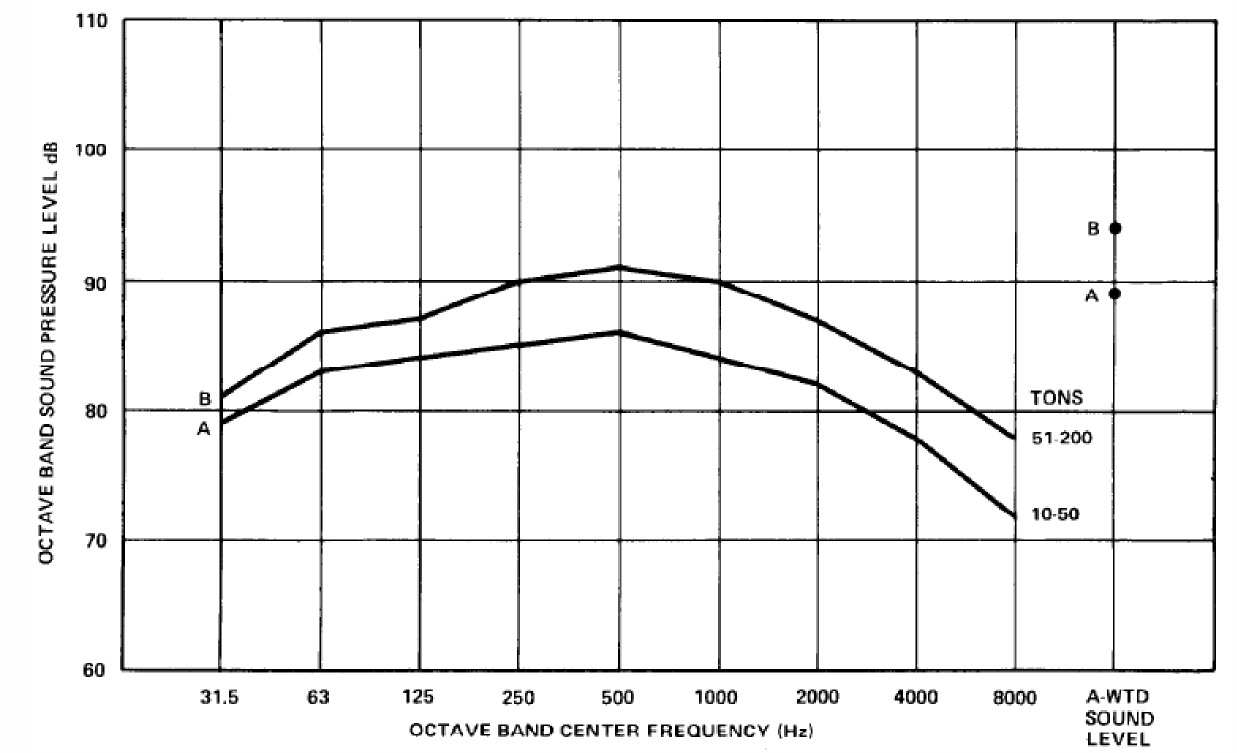

Figure 1

Sound pressure levels of reciprocating compressors at 3-ft. distance

Octave Frequency Band (Hz)	Sound Pressure Level, dB	
	10-50 Tons Cooling Capacity	51-200 Tons Cooling Capacity
31	79	81
63	83	86
125	84	87
250	85	90
500	86	91
1000	84	90
2000	82	87
4000	78	83
8000	72	78
A-weighted, dB(A)	89	94

Table 1

Sound pressure levels (in db at 3-ft. distance) for packaged chillers with reciprocating compressors

Octave Frequency Band (Hz)	Sound Pressure Level, dB 100-300 Tons Cooling Capacity
31	70
63	76
125	80
250	92
500	89
1000	85
2000	80
4000	75
8000	73
A-weighted, dB(A)	90

Table 2

Sound pressure levels (in db at 3-ft. distance) for packaged chillers with rotary screw compressors

5. PACKAGED CHILLERS WITH CENTRIFUGAL COMPRESSORS. These compressors range in size from 100 tons to 4000 tons and represent the leading manufacturers. The noise levels may be influenced by the motors, gears, or turbines, but the measurement positions are generally selected to emphasize the compressor noise. The noise levels given in Figure 2 and Table 3 represent the 80- to 90- percentile values found when the units were divided into the two size groups: under 500 tons and 500 or more tons. The low-frequency noise levels reflect the increased noise found for off-peak loads for most centrifugal machines. These data may be used for packaged chillers, including their drive units. For built-up assemblies, these data should be used for the centrifugal compressor only and the suggestions below followed for combining the noise of other components.

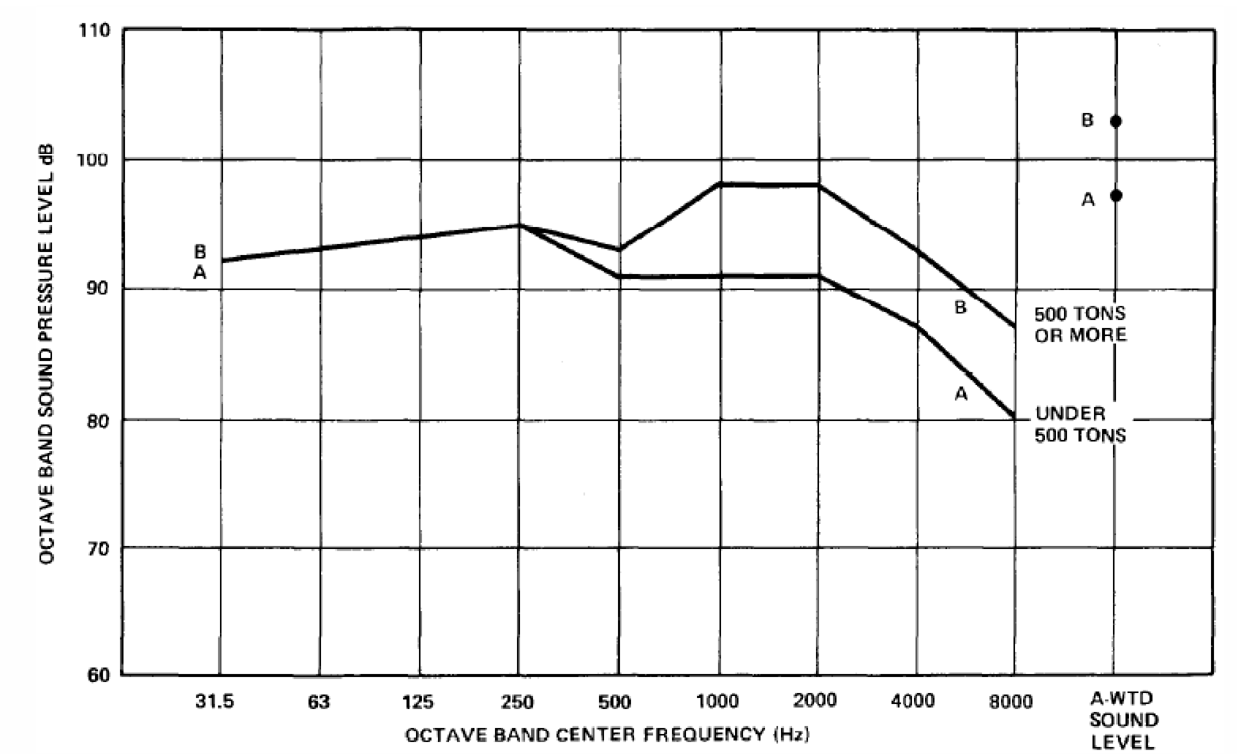

Figure 2

Sound pressure levels of centrifugal compressors at 3-ft. distance

Octave Frequency Band (Hz)	Sound Pressure Level, dB	
	Cooling Capacity Under 500 Tons	Cooling Capacity 500 Tons or More
31	92	92
63	93	93
125	94	94
250	95	95
500	91	93
1000	91	98
2000	91	98
4000	87	93
8000	80	87
A-weighted, dB(A)	97	103

Table 3

Sound pressure levels (in db at 3-ft. distance) for packaged chillers with centrifugal compressors

6. BUILT-UP REFRIGERATION MACHINES. The noise of packaged chillers, as presented in the preceding paragraphs, includes the noise of both the compressor and the drive unit. If a refrigeration system is built up of separate pieces, then the noise level estimate should include the noise of each component making up the assembly. Compressor noise levels should be taken from the packaged chiller data. Sound level data for the drive units (motors, gears, steam turbines) should be taken from the appropriate tables in the manual or obtained from the manufacturers. Decibel addition should be used to determine each octave band sum from the octave band levels of the various components. The acoustic center should be assumed to be at the approximate geometric center of the assembly, and distances should be extrapolated from that point. For very close distances (such as 2 to 3 feet) to each component, assume the total sound levels apply all around the equipment at distances of 3 feet from the approximate geometric centers of each component, although this assumption will not provide exact close-in sound levels.

7. ABSORPTION MACHINES. These units are normally masked by other noise in a mechanical equipment room. The machine usually includes one or two small pumps; steam flow noise or steam valve noise may also be present. The 3 foot distance SPLs for most absorption machines used in refrigeration systems for buildings are given in Table 4.

Octave Frequency Band (Hz)	Sound Pressure Level, dB All Sizes
31	80
63	82
125	82
250	82
500	82
1000	81
2000	78
4000	75
8000	70
A-weighted, dB(A)	86

Table 4

Sound pressure levels (in db at 3-ft. distance) for absorption machines

8. BOILERS.

8.1 NOISE DATA. The estimated noise levels given in Table 5 are believed applicable for all boilers, although some units will exceed these values and, certainly, many units will be much lower than these values. These 3 foot noise levels apply to the front of the boiler, so when other distances are of concern, the distance should always be taken from the front surface of the boiler. Noise levels are much lower off the side and rear of the typical boiler. The wise variety of blower assemblies, air and fuel inlet arrangements, burners, and combustion chambers provides such variability in the noise data that it is impossible simply to correlate noise with heating capacity.

8.2 BOILER RATING. Heating capacity of boilers may be expressed in different ways: sq. ft. of heating surface, BTU/hour, lb of steam/hour, or bhp boiler horsepower). To a first approximation, some of these terms are interrelated as follows:

33,500 BTU/hour = 1 bhp
33 lb of steam/hour = 1 bhp.

In this publication, all ratings have been reduced to equivalent bhp.

9. STEAM VALVES. Estimated noise levels are given in Table 6 for a typical thermally insulated steam pipe and valve. Even though the noise is generated near the orifice of the valve, the pipes on either side of the valve radiate a large part of the total noise energy that is radiated. Hence, the pipe is considered, along with the valve, as a part of the noise source. Valve noise is largely determined by valve type and design, pressure and flow conditions, and pipe wall thickness. Some valve manufacturers can provide valve noise estimated for their products.

10. COOLING TOWERS AND EVAPORATIVE CONDENSERS. The generalizations drawn here may not apply exactly to all cooling towers and condensers, but the data are useful for laying out cooling towers and their possible noise control treatments. It is desirable to obtain from the manufacturer actual measured noise levels for all directions of interest, but if these data are not forthcoming, it is essential to be able to approximate the directional pattern of the cooling tower noise. For aid in identification, four general types of cooling towers are sketched in Figure 3:

A.) The centrifugal-fan blow-through type;
B.) The axial-flow blow-through type (with the fan or fans located on a side wall);
C.) The induced-draft propeller type; and
D.) The "underflow" forced draft propeller type (with the fan located under the assembly.

10.1 SOUND POWER LEVEL DATA. Sound power level data are given for both propeller-type and centrigual-fan cooling towers.

10.1.1 PROPELLER-TYPE COOLING TOWER. The approximate overall and A-weighted sound power levels of propeller-type cooling towers are given by Equations 1 and 2, respectively: for overall PWL (propeller-type),

$$L_w = 95 + 10 \log (\text{fan hp}), \quad \text{(Eq. 1)}$$

and for A-weighted PWL,

$$L_{wa} = 86 + 10 \log (\text{fan hp}), \quad \text{(Eq. 2)}$$

where "fan hp" is the nameplate horsepower rating of the motor that drives the fan. Octave band PWLs can be obtained by subtracting the values of Table 7 from the overall PWL.

10.1.2 CENTRIFUGAL FAN COOLING TOWER. The approximate overall and A-weighted sound power levels of centrifugal-fan cooling towers are given by Equations 3 and 4, respectively: for overall PWL (centrifugal-fan),

$$L_w = 85 + 10 \log (\text{fan hp}) \qquad \text{(Eq. 3)}$$

for A-weighted PWL,

$$L_{wa} = 78 + 10 \log (\text{fan hp}). \qquad \text{(Eq. 4)}$$

Octave Frequency Band (Hz)	Sound Pressure Level, dB 50-2000 BHP
31	90
63	90
125	90
250	87
500	84
1000	82
2000	80
4000	76
8000	70
A-weighted, dB(A)	88

Table 5
Sound pressure levels (in db at 3-ft. distance from the front) for boilers

Octave Frequency (Hz)	Sound Pressure Level (dB)
31	70
63	70
125	70
250	70
500	75
1000	80
2000	85
4000	90
8000	90
A-weighted, dB(A)	94

Table 6

Sound pressure levels (in db at 3-ft. distance) for high-pressure thermally insulated steam valves and nearby piping

Octave Frequency Band (Hz)	Value to be Subtracted From Overall PWL (dB)
31	8
63	5
125	5
250	8
500	11
1000	15
2000	18
4000	21
8000	29
A-weighted, dB(A)	9

Table 7

Frequency adjustments (in db) for propeller-type cooling towers

Octave Frequency Band (Hz)	Value to be Subtracted From Overall PWL (dB)
31	6
63	6
125	8
250	10
500	11
1000	13
2000	12
4000	18
8000	25
A-weighted, dB(A)	7

Table 8

Frequency Adjustments (in dB) for Centrifugal-Fan Cooling Towers

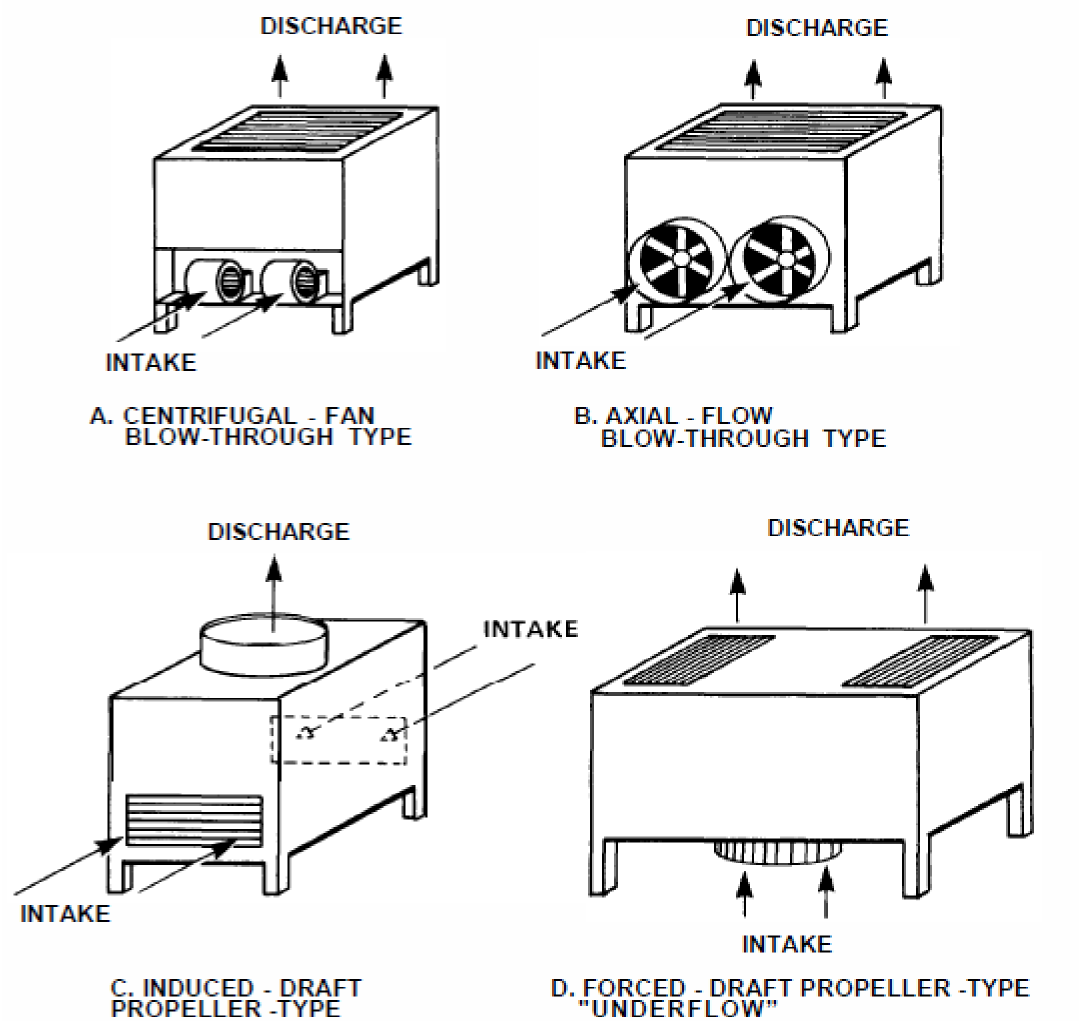

Figure 3
Principal types of cooling towers

When more than one fan or cooling tower is used, "fan hp" should be the total motor-drive hp of all fans or towers. Octave band PWLs can be obtained by subtracting the values of table C-8 from the overall PWL.

10.2 SPLS AT A DISTANCE. Cooling towers usually radiate different amounts of sound in different directions, and the directional corrections of Table 9 should be made to the average SPL. These corrections apply to the five principal directions from a cooling tower, i.e., in a direction perpendicular to each of the four sides and to the top of the tower. If it is necessary to estimate the SPL at some direction other than the principal directions, it is common practice to interpolate between the values given for the principal directions.

(Add these decibel corrections to the *average* SPL calculated for a given distance from the tower. Do not apply these corrections for close-in positions, such as less than 10 ft. Also, these corrections apply when there are no reflecting or obstructing surfaces that would modify the normal radiation of sound from the tower.)

Octave Band (Hz)	31	63	125	250	500	1000	2000	4000	8000
CENTRIFUGAL-FAN BLOW-THROUGH TYPE									
Front (Fan inlet)	+3	+3	+2	+3	+4	+3	+2	+2	+2
Side (Enclosed)	0	0	0	-2	-3	-4	-5	-5	-5
Rear (Enclosed)	0	0	-1	-2	-3	-4	-5	-6	-6
Top (Discharge)	-3	-3	-2	0	+1	+2	+3	+4	+5
AXIAL-FLOW BLOW-THROUGH TYPE									
Front (Fan inlet)	+2	+2	+4	+6	+6	+5	+5	+5	+5
Side (Enclosed)	+1	+1	+1	-2	-5	-5	-5	-5	-4
Rear (Enclosed)	-3	-3	-4	-7	-7	-7	-8	-11	-8
Top (Discharge)	-5	-5	-5	-5	-2	0	0	+2	+1
INDUCED-DRAFT PROPELLER-TYPE									
Front (Air inlet)	0	0	0	+1	+2	+2	+2	+3	+3
Side (Enclosed)	-3	-3	-3	-3	-3	-3	-4	-5	-6
Top (Discharge)	+3	+3	+3	+3	+3	+4	+4	+3	+3
"UNDERFLOW" FORCED-DRAFT PROPELLER-TYPE									
Any side	-1	-1	-1	-2	-2	-3	-3	-4	-4
Top	+2	+2	+2	+3	+3	+4	+4	+5	+5

Table 9

Correction to average SPLs for directional effects of cooling towers

10.3 CLOSE-IN SPLS. Sound power level data usually will not give accurate calculated SPLs at very close distances to large-size sources, such as cooling towers. The data may be used where it is required to estimate close-in SPLs at nearby walls, floors, or in closely confined spaces (at 3- to 5 foot distances from inlet and discharge openings).

10.4 HALF-SPEED OPERATION. When it is practical to do so, the cooling tower fan can be reduced to half speed in order to reduce noise; such a reduction also reduces cooling capacity. Half-speed produces approximately two-thirds cooling capacity and approximately 8- to 10-dB noise reduction in the octave bands that contain most of the fan-induced noise. For half-speed operation, the octave band SPLs or PWLs of full-speed cooling tower noise may be reduced by the following amounts, where fB is the blade passage frequency and is calculated from the relation fB = No. of fan blades ***x*** shaft RPM/GO.

Octave band that contains:	Noise reduction due to half-speed:
1/8 fB	3 dB
1/4 fB	6 dB
1/2 fB	9 dB
fB	9 dB
2 fB	9 dB
4 fB	6 dB
8 fB	3 dB

If the blade passage frequency is not known, it may be assumed to fall in the 63-Hz band for propeller type cooling towers and in the 250-Hz band for centrifugal cooling towers. Waterfall noise usually dominates in the upper octave bands, and it would not change significantly with reduced fan speed.

10.5 LIMITATIONS.

10.5.1 DESIGN VARIATIONS. The data given here represent a fairly complete summary of cooling tower noise, but it must still be expected that noise levels may vary from manufacturer to manufacturer, and from model to model as specific design changes take place. Whenever possible, request the manufacturer to supply the specific noise levels for the specific needs.

10.5.2 ENCLOSED LOCATIONS. Most of the preceding discussion assumes that cooling towers will be used in outdoor locations. If they are located inside enclosed mechanical equipment rooms or within courts formed by several solid walls, the sound patterns will be distorted. In such instances, the PWL of the tower (or appropriate portions of the total PWL) can be placed in that setting, and the enclosed or partially enclosed space can be likened to a room having certain estimated amounts of reflecting and absorbing surfaces. Because of the limitless number of possible arrangements, this is not handled in a general way, so the problem of partially enclosed cooling towers is not treated here. In the absence of a detailed analysis of cooling tower noise levels inside enclosed spaces, it is suggested that the close-in noise levels of Table 10 be used as approximations.

10.6 EVAPORATIVE CONDENSERS. Evaporative condensers are somewhat similar to cooling towers in terms of noise generation. A few evaporative condensers have been included with the cooling towers, but not enough units have been measured to justify a separate study of evaporative condensers alone. In the absence of noise data on specific evaporative condensers, it is suggested that noise data be used for the most nearly similar type and size of cooling tower.

10.7 AIR-COOLED CONDENSERS. For some installations, an outdoor air-cooled condenser may serve as a substitute for a cooling tower or evaporative condenser. The noise of an air-cooled condenser is made up almost entirely of fan noise and possibly air-flow noise through the condenser coil decks. In general, the low-frequency fan noise dominates. Since most of the low-frequency noise of a typical cooling tower is due to the fan system, in the absence of specific data on air-cooled condensers, it is suggested that noise data be used for the most nearly similar type and size of cooling tower.

10.8 EJECTOR-TYPE COOLING TOWER. This is a fanless type cooling tower that induces air flow through the use of nozzles of high-pressure water spray. Noise levels are generally lower for the ejector cooling tower than for cooling towers using fans to produce air flow. Adequate vibration isolation of the drive pump, piping, and tower are necessary, although the elimination of the fan reduces the severity of tower vibration.

11. PUMPS. The overall and A-weighted 3 foot SPLs given in Table 10. The pump power rating is taken as the nameplate power of the drive motor. This is easily determined in field measurements, whereas actual hydraulic power would be unknown in a field situation. For pump ratings under 100 hp, the radiated noise increases with the function (10 log hp), but about 100 hp the noise changes more slowly with increasing power, hence, the function (3 log hp). Octave band SPLs are obtained by subtracting the values of Table 11 from the overall SPLs of Table 10. Pumps intended for high-pressure operation have smaller clearances between the blade tips and the cutoff edge and, as a result, may have higher noise peaks than shown in Tables 11 and 12 (by 5 dB, sometimes 10dB) in the octave bands containing the impeller blade passage frequency and its first harmonic. These would usually fall in the 1,000 and 2,000 Hz octave bands. The data of Tables 11 and 12 are summarized in Figure 4.

Speed Range rpm	Drive Motor Nameplate Power	
	Under 100 hp	Above 100 hp
Overall sound messure level, dB:		
3000-3600	71+10 log hp	85+3 log hp
1600-1800	74+10 log hp	88+3 log hp
1000-1500	69+10 log hp	83+3 log hp
450- 900	67+10 log hp	81+3 log hp
A-weighted sound level, dB(A):		
3000-3600	69+10 log hp	82+3 log hp
1600-1800	72+10 log hp	86+3 log hp
1000-1500	67+10 log hp	81+3 log hp
450- 900	65+10 log hp	79+3 log hp

Table 10

Overall and A-Weighted Sound Pressure Levels (in dB and dB(A)

at 3-ft. Distance) for Pumps

Octave Frequency Band (Hz)	Value to be Subtracted From Overall SPL (dB)
31	13
63	12
125	11
250	9
500	9
1000	6
2000	9
4000	13
8000	19
A-weighted, dB(A)	2

Table 11

Frequency Adjustments fin dB) for Pumps

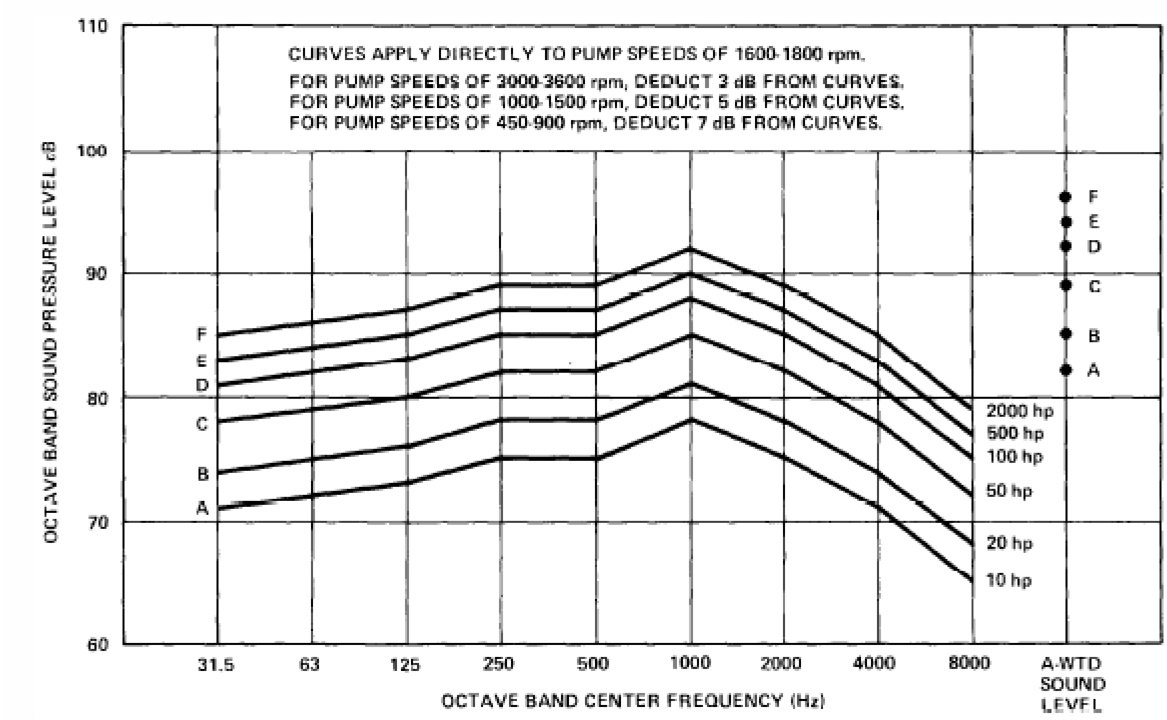

Figure 4

Sound Pressure Levels of Pumps at 3-ft. Distance

12. FANS

12.1 IN-DUCT NOISE. Recent issues of ASHRAE publications provide updated methods for estimating the in-duct noise of ventilating fans. Manufacturers also furnish in-duct PWL data of their fans on request. A current ASHRAE estimation is given by Equation 5:

$$Lw = Kw + 10 \log Q + 20 \log P + BFI + C \qquad \text{(Eq. 5)}$$

where Lw the in-duct sound power level of the fan at either the inlet or discharge end of the fan, Kw the specific sound power level for the particular fan design, Q is the volume flow rate in cfm (ft.3/min.), and P is the static pressure produced by the fan (inches of water gage). Values of Kw for the octave bands and for various basic fan blade designs are given in Part A of Table 12. The blade passage frequency of the fan is obtained from:

fan RPM x no. of blades/60

and the "blade frequency increment" BFI (in dB) is added to the octave band sound power level in the octave in which the blade passage frequency occurs. It is best to obtain the number of blades and the fan rotational speed from the manufacturer to calculate the blade passage frequency. In the event this information is not available, Part B of Table 12 provides the usual blade passage frequency. The estimates given by this method assume ideal inlet and outlet flow conditions and operation of the fan at its design condition. The noise is quite critical to these conditions and increases significantly for deviations from ideal conditions. Part C of Table 12 provides a correction factor for off-peak fan operation.

12.2 NOISE REDUCTION FROM FAN HOUSING. The fan housing and its nearby connected ductwork radiate fan noise into the fan room. The amount of noise is dependent on both internal and external dimensions of the housing and ductwork, the TL of the sheet metal, and the amount of sound absorption material inside the ductwork. Because of so many variables, there is no simple analysis procedure for estimating the PWL of the noise radiated by the housing and ductwork. However, Table 13 offers a rough estimate of this type of noise. These are simply deductions, in dB, from the induct fan noise. At low frequency, the housing appears acoustically transparent to the fan noise, but as frequency increases, the TL of the sheet metal becomes increasingly effective.

Part A. Specific Sound Power Levels for Inlet or Outlets and Blade Frequency Increments (BFI) for Fans.

Fan & Blade Type	Size & Operation	Octave Band Center Frequency – Hz 63	125	250	500	1000	2000	4000	8000	BFI
CENTRIFUGAL – Airfoil, Backward Curved and Inclined	Over 30 in. dia.	37	37	36	31	27	20	16	14	3
	Under 30 in. dia.	42	42	40	36	31	25	21	16	3
CENTRIFUGAL – Forward Curved	All sizes	50	50	40	33	33	28	23	16	2
CENTRIFUGAL – Radial	Low Pressure, SP 4" to 10"									
	Over 40 in. dia.	53	44	40	36	34	29	28	23	7
	Under 40 in. dia.	64	58	50	40	39	36	31	28	7
	Mid Pressure, SP 10" to 20"									
	Over 40 in. dia.	55	51	43	39	36	30	26	23	8
	Under 40 in. dia.	65	60	48	45	43	38	34	31	8
	High Pressure, SP 20" to 60"									
	Over 40 in. dia.	58	55	50	45	43	41	38	35	8
	Under 40 in. dia.	68	64	58	51	51	49	46	43	8
VANEAXIAL	Hub Ratio 0.3 to 0.4	48	40	40	45	44	42	35	31	6
	Hub Ratio 0.4 to 0.6	48	40	43	40	38	33	27	25	6
	Hub Ratio 0.6 to 0.8	58	49	46	48	46	44	40	37	6
TUBEAXIAL	Over 40 in. dia.	46	43	44	48	44	43	36	34	7
	Under 40 in. dia.	45	44	46	50	49	46	40	37	7
PROPELLER	All sizes	46	48	55	53	52	48	43	38	5

Note – Kw for inlet or outlet level, add 3 dB for total level.

Part B. Octave band in which blade frequency increment (BFI) occurs.

Fan Type	Octave Band in Which BFI occurs
Centrifugal	
Airfoil, backward curved & backward inclined	250 Hz
Forward curved	500 Hz
Radial blade	125 Hz
Vaneaxial	125 Hz
Tubeaxial	63 Hz
Propeller	63 Hz

Note – Use for estimating purposes only. Where actual fan is known, use manufacturers' data.

Part C. Correction factor, C, for off-peak operation.

Static Efficiency % of Peak	Correction Factor dB
90 to 100	0
85 to 89	3
75 to 84	6
65 to 74	9
55 to 64	12
50 to 54	15

Table 12

Specific Sound Power Levels Kw (in dB), Blade Frequency Increments (in dB) and Off Peak Correction for Fans of Various Types, for Use in Equation 5

Octave Frequency Band (Hz)	Value to be Subtracted From In-Duct Fan Noise (dB)
31	—
63	0
125	0
250	5
500	10
1000	15
2000	20
4000	22
8000	25

Table 13

Approximate Octave-Band Adjustments for Estimating the PWL of Noise Radiated by a Fan Housing and its Nearby Connected Duct Work

13. AIR COMPRESSORS. Two types of air compressors are frequently found in buildings: one is a relatively small compressor (usually under 5 hp) used to provide a high pressure air supply for operating the controls of the ventilation system, and the other is a medium size compressor (possibly up to 100 hp) used to provide “shop air” to maintenance shops, machine shops, and laboratory spaces, or to provide ventilation system control pressure for large buildings. Larger compressors are used for special industrial processes or special facilities, but these are not considered within the scope of this publication. The 3 foot SPLs are given in Figure 5 and Table 14.

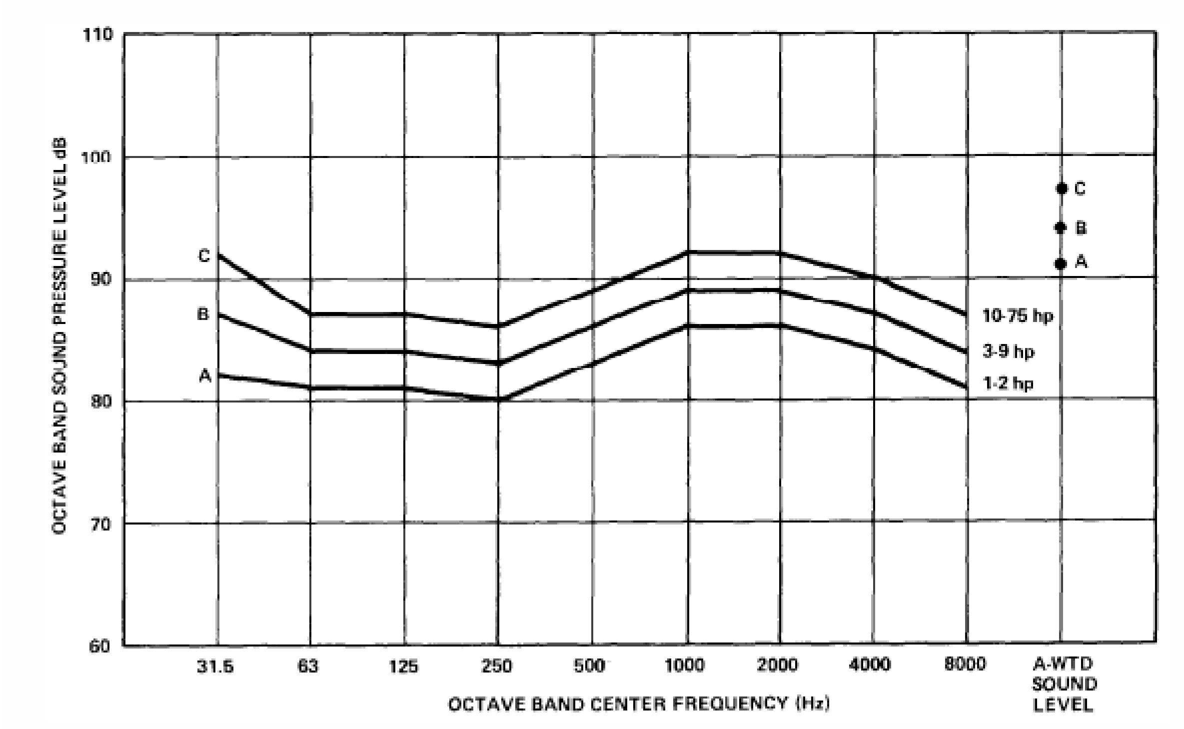

Figure 5

Sound Pressure Levels of Air Compressors at 3-ft. Distance

Octave Frequency Band (Hz)	Air Compressor Power Range		
	1-2 hp (dB)	3-9 hp (dB)	10-75 hp (bB)
31	82	87	92
63	81	84	87
125	81	84	87
250	80	83	86
500	83	86	89
1000	86	89	92
2000	86	89	92
4000	84	87	90
8000	81	84	87
A-weighted, dB(A)	91	94	97

Table 14

Sound Pressure Levels (in dB at 3-ft. Distance) for

Air Compressors

14. RECIPROCATING ENGINES. A comprehensive study has been made of the noise characteristics of reciprocating and turbine engines fueled by natural gas and liquid fuel. The noise levels of the engines as sound sources are summarized here, because these engines may be used as power sources in buildings, and their noise should be taken into account. Typically, each engine type has three sound sources of interest; the engine casing, the air inlet into the engine, and the exhaust from the engine.

14.1 ENGINE CASING. The PWL of the noise radiated by the casing of a natural-gas or diesel reciprocating engine is given by Equation 6:

$$Lw = 93 + 10 \log (\text{rated hp}) + A + B + C + D , \qquad \text{(Eq. 6)}$$

where Lw is the overall sound power level (in dB),“rated hp” is the engine manufacturer’s continuous full-load rating for the engine (in horsepower), and A, B, C, and D are correction terms (in dB), given in Table 15. Octave band PWLs can be obtained by subtracting the Table 16 values from the overall PWL given by Equation 6. The octave band corrections are different for the different engine speed groups. For small engines (under about 450 hp), the air intake noise is usually radiated close to the engine casing, so it is not easy or necessary to separate these two sources; and the engine casing noise may be considered as including air intake noise (from both naturally aspirated and turbocharged engines).

	dB
Speed correction term "A"	
Under 600 rpm	-5
600-1500 rpm	-2
Above 1500 rpm	0
Fuel correction term "B"	
Diesel fuel only	0
Diesel and/or natural gas	0
Natural gas only (may have small amount of "pilot oil")	-3
Cylinder arrangement term "C"	
In-line	0
V-type	-1
Radial	-1
Air intake correction term "D"	
Unducted air inlet to unmuffled Roots Blower	+3
Ducted air from outside the room or into muffled Roots Blower	0
All other inlets to engine (with or without turbochargers)	0

Table 15

Correction Terms (in dB) to be Applied to Equation 6 for Estimating the Overall PWL of the Casing Noise of a Reciprocating Engine

Octave Frequency Band (Hz)	Value to be Subtracted From Overall PWL, in dB			
	Engine Speed Under 600 rpm	Engine Speed 600-1500 rpm		Engin Speed Over 1500 rp
		Without Roots Blower	With Roots Blower	
31	12	14	22	22
63	12	9	16	14
125	6	7	18	7
250	5	8	14	7
500	7	7	3	8
1000	9	7	4	6
2000	12	9	10	7
4000	18	13	15	13
8000	28	19	26	20
A-weighted, dB(A)	4	3	1	2

Table 16

Frequency Adjustments (in dB) for Casing Noise of Reciprocating Engines

14.2 TURBOCHARGED AIR INLET. Most large engines have turbochargers at their inlet to provide pressurized air into the engine for increased performance. The turbocharger is a turbine driven by the released exhaust gas of the engine. The turbine is a high-frequency sound source. Turbine configuration and noise output can vary appreciably, but an approximation of the PWL, of the turbocharger noise is given by Equation 7:

$$Lw = 94 + 5 \log (\text{rated hp}) - L/6 \qquad \text{(Eq. 7)}$$

where Lw and “rated hp” are already defined and L is the length, in ft., of a ducted inlet to the turbocharger. For many large engines, the air inlet may be ducted to the engine from a fresh air supply or a location outside the room or building. The term L/6, in dB, suggests that each 6 ft. of inlet ductwork, whether or not lined with sound absorption material, will provide about 1 dB of reduction of the turbocharger noise radiated from the open end of the duct. This is not an accurate figure for ductwork in general; it merely represents a simple token value for this estimate. The octave band values given in Table 17 are subtracted from the overall PWL of Equation 7 to obtain the octave band PWLs of turbocharged inlet noise.

Octave Frequency Band (Hz)	Value to be Subtracted From Overall PWL (dB)
31	4
63	11
125	13
250	13
500	12
1000	9
2000	8
4000	9
8000	17
A-weighted, dB(A)	3

Table 17

Frequency Adjustments fin dB) for Turbocharger

Air Inlet Noise

14.3 ENGINE EXHAUST. The PWL of the noise radiated from the unmuffled exhaust of an engine is given by Equation 8:

$$Lw = 119 + 10 \log (\text{rated hp}) - T - L/4 \qquad \text{(Eq. 8)}$$

where T is the turbocharger correction term (T = 0 dB for an engine without a turbocharger and T = 6 dB for an engine with a turbocharger) and L is the length, in ft., of the exhaust pipe. A turbocharger takes energy out of the discharge gases and results in an approximately 6-dB reduction in noise. The octave band PWLs of unmuffled exhaust noise are obtained by subtracting the values of Table 18 from the overall PWL derived from Equation 8. If the engine is equipped with an exhaust muffler, the final noise radiated from the end of the tailpipe is the PWL of the unmuffled exhaust minus the insertion loss, in octave bands, of the muffler.

Octave Frequency Band (Hz)	Value to be Subtracted From Overall PWL (dB)
31	5
63	9
125	3
250	7
500	15
1000	19
2000	25
4000	35
8000	43
A-weighted, dB (A)	12

Table 18

Frequency Adjustments (in dB) for Unmuffled

Engine Exhaust Noise

15. GAS TURBINE ENGINES

15.1 PWL OF THREE SOURCES. As with reciprocating engines, the three principal sound sources of turbine engines are: the engine casing, the air inlet, and the exhaust. Most gas turbine manufactures will provide sound power estimates of these sources. However when these are unavailable the overall PWLs of these three sources, with no noise reduction treatments, are given in the following equations: for engine casing noise,

$$Lw = 120 + 5 \log (\text{rated MW}) \qquad \text{(Eq. 9)}$$

for air inlet noise,

$$Lw = 127 + 15 \log (\text{rated MW}) \qquad \text{(Eq. 10)}$$

for exhaust noise

$$Lw = 133 + 10 \log (\text{rated MW}) \qquad \text{(Eq. 11)}$$

where "rated MW" is the maximum continuous full-load rating of the engine in megawatts. If the manufacturer lists the rating in "effective shaft horsepower" (eshp), the MW rating may be approximated by

$$MW = eshp/1400$$

Overall PWLs, obtained from Equations 9 through 11, are tabulated in Table 19 for a useful range of MW ratings.

15.1.1 TONAL COMPONENTS. For casing and inlet noise, particularly strong high-frequency sounds may occur at several of the upper octave bands. However which bands contain the tones will depend on the specific design of the turbine and, as such, will differ between models and manufacturers. Therefore, the octave band adjustments of Table 20 allow for these peaks in several different bands, even though they probably will not occur in all bands. Because of this randomness of peak frequencies, the A-weighted levels may also vary from the values quoted.

15.1.2 ENGINE COVERS. The engine manufacturer sometimes provides the engine casing with a protective thermal wrapping or an enclosing cabinet, either of which can give some noise reduction. Table 19 suggests the approximate noise reduction for casing noise that can be assigned to different types of engine enclosures. Refer to the notes of the table for a broad description of the enclosures. The values of Table 19 may be subtracted from the octave band PWLs of casing noise to obtain the adjusted PWLs of the covered or enclosed casing. An enclosure specifically designed to control casing noise can give larger noise reduction values than those in the table. However it should be noted that the performance of enclosures that are supported on the same structure as the gas turbine, will be limited by structure borne sound. For this reason care should be used in applying laboratory data of enclosure performance to the estimation of sound reduction of gas turbine enclosures.

15.2 EXHAUST AND INTAKE STACK DIRECTIVITY. Frequently, the exhaust of a gas turbine engine is directed upward. The directivity of the stack provides a degree of noise control in the horizontal direction. Or, in some installations, it may be beneficial to point the intake or exhaust opening horizontally in a direction away from a sensitive receiver area. In either event, the directivity is a factor in noise radiation. Table 21 gives the approximate directivity effect of a large exhaust opening. This can be used for either a horizontal or vertical stack exhausting hot gases. Table 21 shows that from approximately 0 to 60 degrees from the axis of the stack, the stack will yield higher sound levels than if there was no stack and the sound were emitted by a non-directional point source. From about 60 to 135 degrees from the axis, there is less sound level than if there were no stack. In other words, directly ahead of the opening there is an increase in noise, and off to the side of the opening there is a decrease in noise. Table 21 values also apply for a large-area intake opening into a gas turbine for the 0 to 60 degree range; for the 90 to 135 degree range, subtract an addition 3 dB from the already negative-valued quantities. For horizontal stacks, sound-reflecting obstacles out in front of the stack opening can alter the directivity pattern. Even irregularities on the ground surface can cause some backscattering of sound into the 90 to 180 degree regions, for horizontal stacks serving either as intake or exhaust openings. For small openings in a wall, such as for ducted connections to a fan intake or discharge, use approximately one-half the directivity effect of Table 21 (as applied to intake openings) for the 0 to 90 degree region. For angles beyond 90 degrees, estimate the effect of the wall as a barrier.

16. ELECTRIC MOTORS. Motors cover a range of 1 to 4000 hp and 450 to 3600 RPM. The data include both “drip-proof’ (DRPR) (splash-proof or weather-protected) and “totally enclosed fan-cooled” (TEFC) motors. Noise levels increase with power and speed.

Rated MW	Casing PWL dB	Inlet PWL dB	Exhaust PWL dB
0.10	115	112	123
0.13	116	114	124
0.16	116	115	125
0.20	117	117	126
0.25	117	118	127
0.32	118	120	128
0.40	118	121	129
0.50	118	122	130
0.63	119	124	131
0.80	120	126	132
1.0	120	127	133
1.3	121	129	134
1.6	121	130	135
2.0	122	132	136
2.5	122	133	137
3.2	123	135	138
4.0	123	136	139
5.0	123	137	140
6.3	124	139	141
8.0	125	141	142
10	125	142	143
13	126	144	144
16	126	145	145
20	127	147	146
25	127	148	147
32	128	150	148
40	128	151	149
50	128	152	150
63	129	154	151
80	130	156	152

Table 19

Overall PWLs of the Principal Noise Components of Gas Turbine Engines Having No Noise Control Treatments

Octave Frequency Band (Hz)	Value To Be Subtracted From Overall PWL, in dB		
	Casing	Inlet	Exhaust
31	10	19	12
63	7	18	8
125	5	17	6
250	4	17	6
500	4	14	7
1000	4	8	9
2000	4	3	11
4000	4	3	15
8000	4	6	21
A-weighted, dB(A)	2	0	4

Table 20

Frequency Adjustments (in dB) for Gas Turbine Engine Noise Sources

Octave Frequency Band (Hz)	Relative Sound Level for Indicated Angle From Axis				
	0°	45°	60°	90°[a]	135° and larger[a]
31	8	5	2	-2	-3
63	8	5	2	-3	-4
125	8	5	2	-4	-6
250	8	6	2	-6	-8
500	9	6	2	-8	-10
1000	9	6	1	-10	-13
2000	10	7	0	-12	-16
4000	10	7	-1	-14	-18
8000	10	7	-2	-16	-20

[a]For air intake openings subtract 3 dB from the values in the 90° and 135° columns, i.e., -2 -3 = -5 dB for 31 cps at 90°.

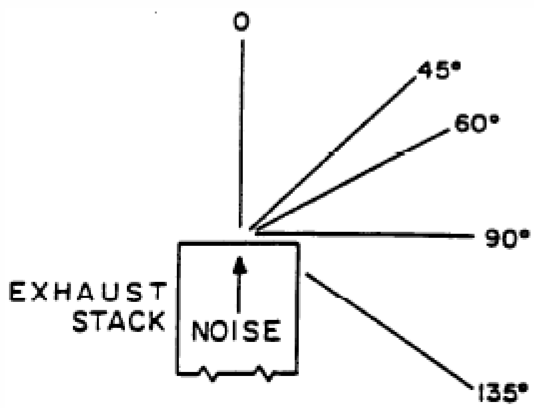

Table 21

Approximate Directivity Effect fin dB) of a Large Exhaust Stack Compared to a Nondirectional Source of the Same Power

16.1 TEFC MOTORS. The overall SPLs of TEFC motors, at the normalized 3 foot condition, follow approximately the relationships of Equations 12 and 13.

For power ratings under 50 hp,

$$Lp = 15 + 17 \log hp + 15 \log RPM \qquad \text{(Eq. 12)}$$

and for power ratings above 50 hp,

$$Lp = 27 + 10 \log hp + 15 \log RPM \qquad \text{(Eq. 13)}$$

where “hp” is the nameplate motor rating in horsepower and “RPM” is the motor shaft speed. For motors above 400 hp, the calculated noise value for a 400-hp motor should be used. These data are not applicable to large commercial motors in the power range of 1000 to 5000 hp. The octave band corrections for TEFC motors are given in Table 22. The data of Equations 12 and 13 and Table 22 are summarized in Figure 6, which gives the SPLs at 3 foot distance for TEFC motors for a working range of speeds and powers. Some motors produce strong tonal sounds in the 500, 1,000, or 2,000 Hz octave bands because of the cooling fan blade frequency. Table 22 and Figure 6 allow for a moderate amount of these tones, but a small percentage of motors may still exceed these calculated levels by as much as 5 to 8 dB. When specified, motors that are quieter than these calculated values by 5 to 10 dB can be purchased.

Octave Frequency Band (Hz)	Value to be Subtracted From Overall SPL (dB)
31	14
63	14
125	11
250	9
500	6
1000	6-
2000	7
4000	12
8000	20
A-weighted, dB(A)	1

Table 22

Frequency Adjustments (in dB) for TEFC Electric Motors.

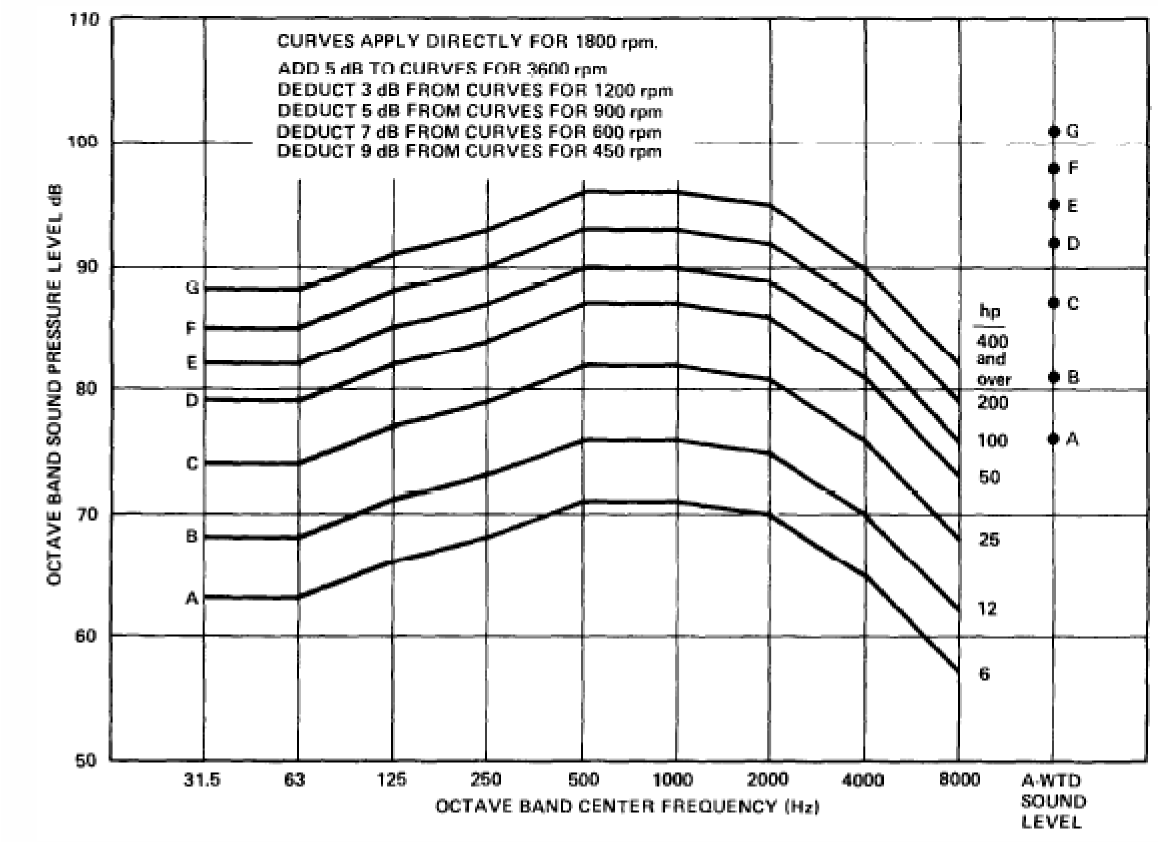

Figure 6

Sound pressure levels of TEFC motors at 3-ft. distance.

16.2 DRPR MOTORS. The overall SPLs of DRPR motors, at the normalized 3 foot condition, follow approximately the relationships of Equations 14 and 15.

For power ratings under 50 hp,

$$Lp = 10 + 17 \log hp + 15 \log RPM. \qquad \text{(Eq. 14)}$$

and for power ratings above 50 hp,

$$Lp = 22 + 10 \log hp + 15 \log RPM. \qquad \text{(Eq. 15)}$$

For motors above 400 hp, the calculated noise value for a 400 hp motor should be used. The octave band corrections for DRPR motors are given in Table 23. The data of Equations 14 and 15 and Table 23 are summarized in Figure 7, which gives the SPLs at 3 foot distance for DRPR motors over a range of speeds and powers.

Octave Frequency Band (Hz)	Value to be Subtracted From Overall SPL (dB)
31	9
63	9
125	7
250	7
500	6
1000	9
2000	12
4000	18
8000	27
A-weighted, dB(A)	4

Table 23

Frequency Adjustments (in dB) for DRPR Electric Motors.

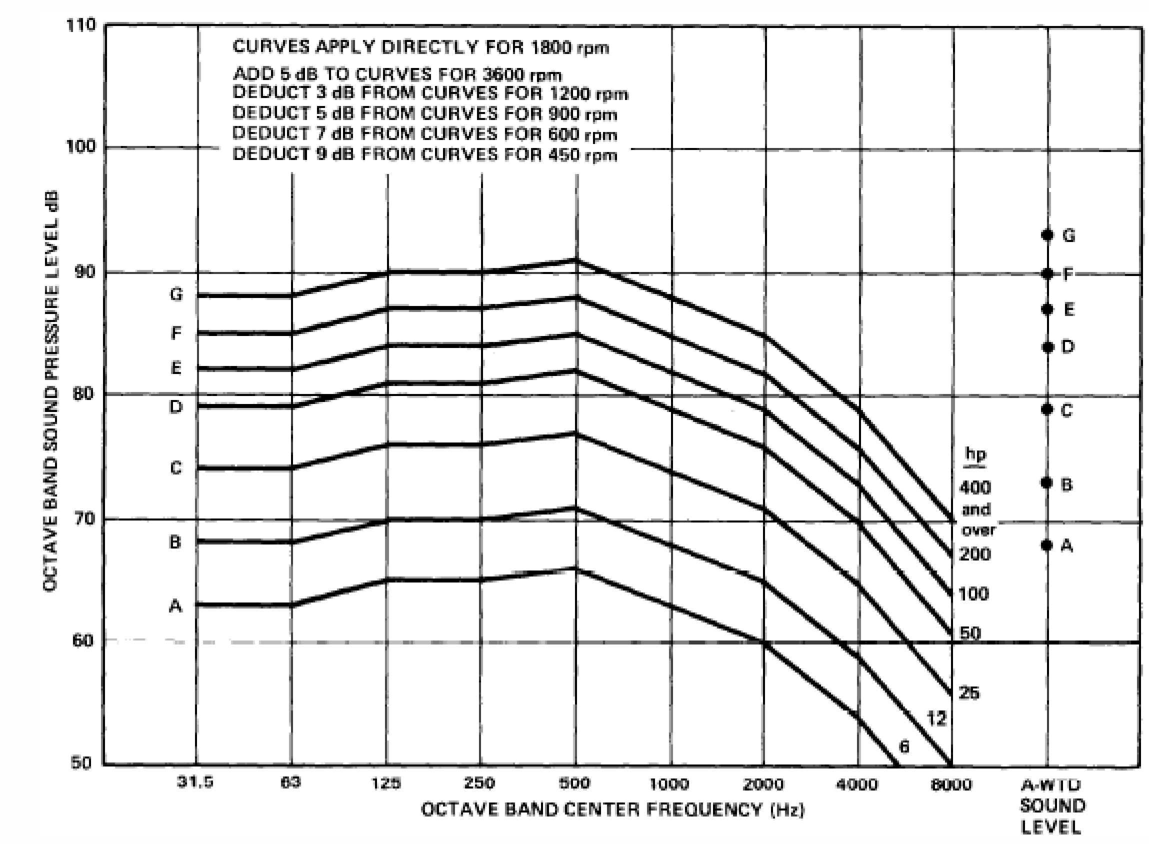

Figure 7

Sound Pressure Levels of DRPR Motors at 3 ft. Distance.

17. STEAM TURBINES. Noise levels are found generally to increase with increasing power rating, but it has not been possible to attribute any specific noise characteristics with speed or turbine blade passage frequency (because these were not known on the units measured). The suggested normalized SPLs at 3 foot distance are given in Figure 8 and Table 24.

Octave Frequency Band (Hz)	Steam Turbine Power Range		
	500-1500 hp (dB)	1501-5000 hp (dB)	5001-15000 hp (dB)
31	86	88	90
63	91	93	95
125	91	93	95
250	88	90	92
500	85	87	89
1000	85	88	91
2000	87	91	95
4000	84	88	92
8000	76	81	86
A-weighted, dB(A)	92	95	99

Table 24

Sound Pressure Levels (in dB at 3 ft distance) for steam turbines.

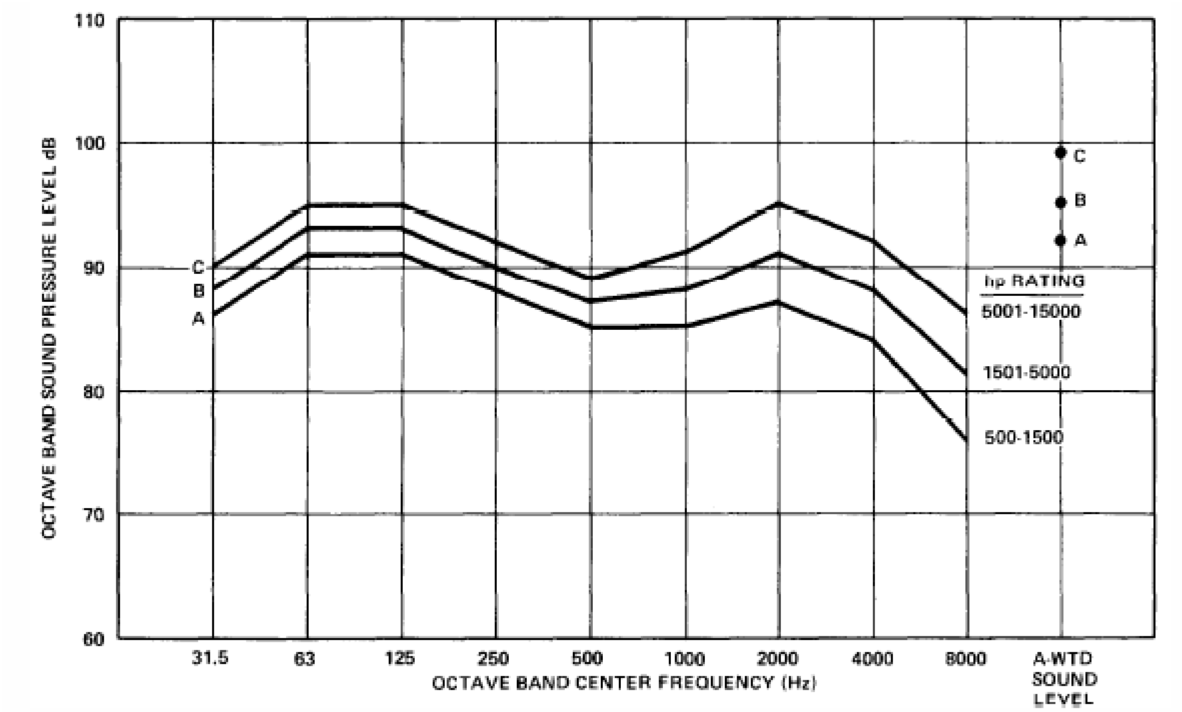

Figure 8

Sound Pressure Levels of Steam Turbines at 3 ft. Distance.

18. GEARS. It is generally true that the noise output increases with increasing speed and power but it is not possible to predict in which frequency band the gear tooth contact rate or the "ringing frequencies" will occur for any unknown gear. The possibility that these frequency components may occur in any of the upper octave bands is covered by, Equation 16, which gives the octave band SPL estimate (at the 3 feet normalized condition) for all bands at and above 125 Hz:

$$L_p = 78 + 3 \log(\text{RPM}) + 4 \log(\text{hp}) \qquad \text{(Eq. 16)}$$

where "RPM" is the speed of the slower gear shaft and "hp" is the horsepower rating of the gear or the power transmitted through the gear. For the 63 Hz band, 3 dB is deducted; and for the 31 Hz band, 6 dB is deducted from the Equation 16 value. This estimate may not be highly accurate, but it will provide a reasonable engineering evaluation of the gear noise. Table 25 gives the estimated SPL in the 125 through 8,000 Hz bands for a variety of speeds and powers, based on Equation 16.

Deduct 6 dB for 31-Hz band
Deduct 3 dB for 63-Hz band

Speed of Slower Gear Shaft (rpm)	Power Rating of Gear					
	50 hp (dB)	100 hp (dB)	200 hp (dB)	500 hp (dB)	1000 hp (dB)	2000 hp (dB)
600	93	94	95	97	98	99
1200	94	95	96	98	99	100
1800	95	96	97	99	100	101
2400	95	96	97	99	100	101
3600	95	97	98	100	101	102
4800	96	97	98	100	101	102

Table 25

Approximate Sound Pressure Levels (in dB at 3-ft. Distance) for Gears, in the 125- through 8000-Hz Octave Bands, from Equation 16.

19. GENERATORS. The noise of generators, in general, can be quite variable, depending on speed, the presence or absence of air cooling vanes, clearances of various rotor parts, etc., but, most of all, on the driver mechanism. When driven by gas or diesel reciprocating engines, the generator is usually so much quieter than the engine that it can hardly be measured, much less heard. For gas turbine engines the high-speed generator may be coupled to the engine through a large gear, and the gear and the generator may together produce somewhat indistinguishable noise in their compartment, which frequently is separated by a bulk head from the engine compartment. Table 26 gives an approximation of the overall PWL of several generators. It is not claimed that this is an accurate estimate, but it should give reasonable working values of PWL. It is to be noted that the PWL of the generator is usually less than that of the drive gear and less than that of the untreated engine casing. Octave band corrections to the overall PWL are given in Table 27.

Generator Speed, (rpm)	Electrical Power Rating of Generator							
	0.2 MW (dB)	0.5 MW (dB)	1 MW (dB)	2 MW (dB)	5 MW (dB)	10 MW (dB)	20 MW (dB)	50 MW (dB)
600	95	99	102	105	109	112	115	119
1200	97	101	104	107	111	114	117	121
1800	98	102	105	108	112	115	118	122
2400	99	103	106	109	113	116	119	123
3600	100	104	107	110	114	117	120	124
4800	101	105	108	111	115	118	121	125

Table 26

Approximate Overall PWL (in dB) of Generators, Excluding the Noise of the Driver Unit.

Octave Frequency Band (Hz)	Value to be Subtracted From Overall SPL (dB)
31	11
63	8
125	7
250	7
500	7
1000	9
2000	11
4000	14
8000	19
A-weighted, dB(A)	4

Table 27

Frequency Adjustments (in dB) for Generators, Without Drive Unit.

20. TRANSFORMERS. The National Electrical Manufacturers Association (NEMA) provides a means of rating the noise output of transformers. The NEMA "audible sound level," as it is called in the standard, is the average of several A-weighted sound levels measured at certain specified positions. The NEMA sound level for a transformer can be provided by the manufacturer. On the basis of field studies of many transformer installations, the PWL in octave bands has been related to the NEMA rating and the area of the four side walls of the unit. This relationship is expressed by Equation 17:

$$Lw = \text{NEMA rating} + 10 \log A + C, \qquad \text{(Eq. 17)}$$

where "NEMA rating" is the A-weighted sound level of the transformer provided by the manufacturer, obtained in accordance with current NEMA Standards, A is the total surface area of the four side walls of the transformer in ft^2, and C is an octave band correction that has different values for different uses, as shown in Table 28. If the exact dimensions of the transformer are not known, an approximation will suffice. If in doubt, the area should be estimated on the high side. An error of 25 percent in area will produce a change of 1 dB in the PWL. The most nearly applicable C value from Table 28 should be used. The Cl value assumes normal radiation of sound. The C2 value should be used in regular-shaped confined spaces where standing waves will very likely occur, which typically may produce 6 dB higher sound levels at the transformer harmonic frequencies of 120, 240, 360, 480, and 600 Hz (for 60-Hz line frequency; or other sound frequencies for other line frequencies). Actually, the sound power level of the transformer does not increase in this location, but the sound analysis procedure is more readily handled by presuming that the sound power is increased. The C3 value is an approximation of the noise of a transformer that has grown noisier (by about 10 dB) during its lifetime. This happens occasionally when the laminations or tie-bolts become loose, and the transformer begins to buzz or rattle. In a highly critical location, it would be wise to use this value. All of the Table 28 values assume that the transformer initially meets it quoted NEMA sound level rating. Field measurements have shown that transformers may actually have A-weighted sound levels that range from a few decibels

(2 or 3 dB) above to as much as 5 or 6 dB below the quoted NEMA value. Quieted transformers that contain various forms of noise control treatments can be purchased at as much as 15 to 20 dB below normal NEMA ratings. If a quieter transformer is purchased and used, the lowered sound level rating should be used in place of the regular NEMA rating in Equation 17, and the appropriate corrections from Table 28 selected.

Octave Frequency Band (Hz)	Octave Band Corrections, in dB		
	C_1, see Note 1	C_2, see Note 2	C_3, see Note 3
31	-1	-1	-1
63	+5	+8	+8
125	+7	+13	+13
250	+2	+8	+12
500	+2	+8	+12
1000	-4	-1	+6
2000	-9	-9	+1
4000	-14	-14	-4
8000	-21	-21	-11

Note 1. Use C_1 for outdoor location or for indoor location in a large mechanical equipment room (over about 5000 ft.^{3}) containing many other pieces of mechanical equipment that serve as obstacles to diffuse sound and breakup standing waves.

Note 2. Use C_2 for indoor locations in transformer vaults or small rooms (under about 5000 ft.^{3}) with parallel walls and relatively few other large-size obstacles that can diffuse sound and breakup standing waves.

Note 3. Use C_3 for any location where a serious noise problem would result if the transformer should become noisy above its NEMA rating, following its installation and initial period of use.

Table 28

Octave-Band Corrections (in dB) to be Used in Equation 17 for Obtaining PWL of Transformers in Different Installation Conditions.

21. OPENING IN A WALL. An opening, such as a door, window, or louvered vent, in an exterior wall of a noisy room will allow noise to escape from that room and perhaps be disturbing to neighbors. The PWL of the sound that passes through the opening can be estimated from equation 18:

$$Lw = Lp + 10 \log A - 10 \qquad \text{(Eq. 18)}$$

where Lp is the SPL in the room at the location of the opening and A is the area, in ft^2, of the opening. (Note, the factor of -10 is due to the use of ft^2 for A, if m^2 had been used then this factor would be 0). Once the PWL is estimated, the SPL at any neighbor distance can be estimated. For normal openings (windows or vents) without ducted connections to the noise source, it may be assumed that the sound radiates freely in all directions in front of the opening, but to the rear of the wall containing the opening, the barrier effect of the wall should be taken into account. For ducted connections from a sound source to an opening in the wall, the sound is somewhat "beamed" out of the opening and may be assumed to have a directivity effect of above one-half the amount given in Table 21 for air intake openings of large stacks.

22. GLOSSARY

Absorption
Conversion of acoustic energy to heat energy or another form of energy within the medium of sound-absorbing materials.

Absorption Coefficient
The ratio of sound energy absorbed by the acoustical material to that absorbed by a perfect absorptive material. It is expressed as a decimal fraction.

Average Sound Level and Average SPL
The arithmetic average of several related sound levels (or SPL in a specified frequency band) measured at different positions or different times, or both.

A-Weighting (dBA)
A frequency response characteristic incorporated in sound-level meters and similar instrumentation. The A-weighted scale response de-emphasizes the lower frequencies and is therefore similar to the human hearing.

Background Noise
The total noise produced by all other sources associated with a given environment in the vicinity of a specific sound source of interest, and includes any Residual Noise.

Decibel (dB)
A unit for expressing the relative power level difference between acoustical or electrical signals. It is ten times the common logarithm of the ratio of two related quantities that are proportional to power.

Field Sound Transmission Class (FSTC)
A single-number rating derived from measured values of field sound transmission loss in accordance with ASTM E-413, "Rating Sound Insulation", and ASTM E-336, "Measurement of Airborne Sound Insulation in Buildings". It provides an estimate of the performance of actual partitions in place and takes into account acoustical room effects.

Field Sound Transmission (FSTL)
The sound loss through a partition installed in a building, in a Loss specified frequency band. It is the ratio of the airborne sound power incident on the partition to the sound power transmitted by the partition and radiated on the other side, expressed in decibels.

Frequency (Hz)
The number of cycles occurring per second. (Hertz is a unit of frequency, defined as one cycle per second).

Noise
Any unwanted sound that can produce undesirable effects or reactions in humans.

Noise Criteria (NC)
Octave band curves used to define acceptable levels of mechanical equipment noise in occupied spaces. Superseded by the Room Criteria (RC).

Noise Isolation Class (NIC)
A single-number rating derived from measured values of noise reduction, as though they were values of transmission loss, in accordance with E-413. It provides an estimate of the sound isolation between two enclosed spaces that are acoustically connected by one or more paths.

Octave Band
A range of frequencies whose upper band limit frequency is nominally twice the lower band limit frequency.

Octave-Band Sound Level
The integrated sound pressure level of only those sin-wave Pressure components in a specified octave band, for a noise or sound having a wide spectrum.

Residual Noise
The measured sound level which represents the summation of the sound from all the discrete sources affecting a given site at a given time, exclusive of the Background Noise or the sound from a Specific Sound Source of interest. In acoustics, residual noise often is defined as the sound level exceeding 90% of a noise monitoring period.

Room Criteria (RC)
Octave band criteria used to evaluate acceptable levels of mechanical equipment noise in occupied spaces.

Sound Power level (Lw or PWL)
Ten times the common logarithm of the ratio of the total acoustic power radiated by a sound source to a reference power. A reference power of a picowatt or 10-12 watt is conventionally used.

Sound Pressure Level (Lp or SPL)
Ten times the common logarithm to the base 10 of the ratio of the mean square sound pressure to

the square of a reference pressure. Therefore, the sound pressure level is equal to 20 times the common logarithm of the ratio of the sound pressure to a reference pressure (20 micropascals or 0.0002 microbar).

Sound Transmission Class (STC)

A single-number rating derived from measured values of transmission loss in accordance with ASTM E-413, "Classification for Rating Sound Insulation" and ASTM E-90, "Test Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions". It is designed to give an estimate of the sound insulation properties of a partition or a rank ordering of a series of partitions.

Sound Transmission Loss (TL)

A measure of sound insulation provided by a structural configuration. Expressed in decibels, it is ten times the common logarithm of the sound energy transmitted through a partition, to the total energy incident upon the opposite surface.