13 NOISE IN MECHANICAL SYSTEMS

13.1 MECHANICAL SYSTEMS

Occupied spaces need the continuous delivery of the requirements for the human habitat—air, water, power, a controlled thermal environment—and the return of the waste products back to the surroundings in the form of carbon dioxide, waste water, sewage, refuse, and heat. To carry out these functions specialized machines are included in every building. The delivery of air and environmental control is provided by a heating, ventilating, and air conditioning (HVAC) system, water is circulated by pumps, and waste is removed through piping.

Fresh air is delivered by electric fans, most often centrifugal, but occasionally a plug, vane axial, or propeller type. Since thermal requirements necessitate the movement of more air than the oxygen requirements, most of the air in a room is recirculated to add or remove heat, and a portion of it is replaced with fresh air from the outside. Since the building is slightly pressurized by this process, some air leaks out through openings and the rest is removed by the mechanical system.

Temperature is controlled by blowing room air over a heat exchanger, a series of tubes, like the radiator in a car, through which a heated or cooled liquid is circulated. Heat, which is produced by electric resistive elements or a gas-fired boiler, is much easier to generate than cooling. Cooling is created by forcing a pressurized liquid or gas through an orifice, where it expands and some of the liquid changes to a gas, thereby absorbing heat from the surroundings. In the heat exchanger the cooled refrigerant is evaporated by taking heat from the circulating air. Once this process is complete the gas is recycled back to the condenser, where it is pressurized and converted back to a liquid, thereby releasing heat to the atmosphere. Figure 13.1 illustrates the process in a packaged air handler, which contains a compressor, a fan to exhaust the heat given off by the compressor, a pump to circulate the cooling fluid, a heat exchanger coil, and another fan to circulate the air in the room. When the compressor is physically separated from the fan coil unit as in Fig. 13.2, it is called a split system and the refrigerant is circulated through pipes connecting the two components.

Noise and vibration are often the byproducts of these mechanical processes. Figure 13.3 shows an example of an HVAC unit and several of the most common structural and airborne noise transmission paths. Each of the paths must be treated to assure overall noise control.

FIGURE 13.1 Heat Flow in a Package Air Handler

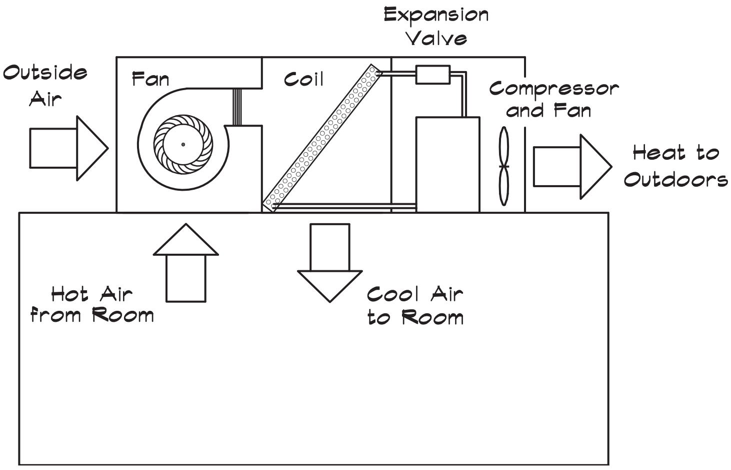


FIGURE 13.2 Air and Heat Flow in a Split System Air Conditioner

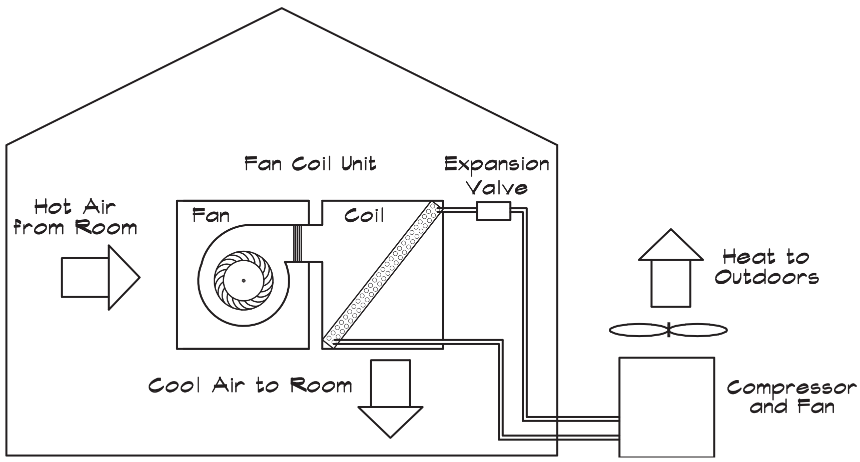
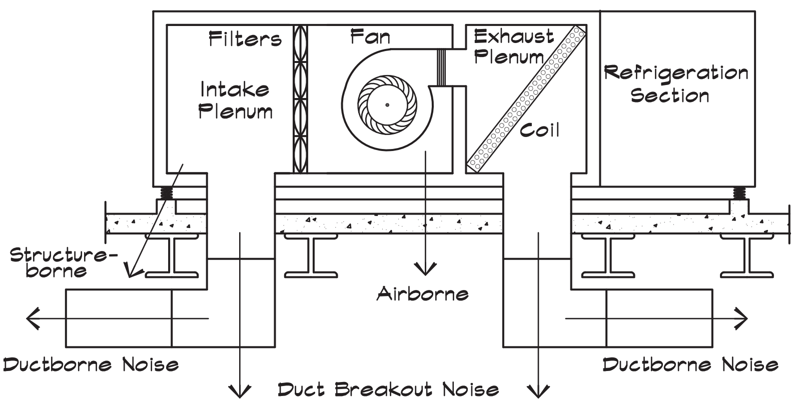


FIGURE 13.3 Rooftop Air Handling Unit Showing Noise Transmission Paths



If a generalization can be made, first it is important to insure that the structure-borne path is isolated. This may be accomplished by locating equipment away from sensitive receivers, and by vibration isolating it, along with all solid connections to it. Once the vibrational path is controlled, then the airborne path can be treated.

Manufacturer Supplied Data

An airborne sound transmission calculation begins with the sound power level generated by a piece of equipment. Often the manufacturer can provide measured sound power level data in octave bands or sound pressure levels at a known distance. The fact that data are available, however, should not lead to the suspension of disbelief about their accuracy. It is important to ferret out the origin of the measurements, and then to compare manufacturer-supplied data with data calculated from standard equations to see if the two are in general agreement. If they are not, further inquiries may be necessary to clarify the reason for the difference.

It seems logical that manufacturers would simply measure the noise their equipment makes and publish the data. Logic does not always prevail and companies may rely instead on calculations or other methods to determine noise level data. Although calculated data are better than nothing, the user does not necessarily know if the data are measured or calculated, and if calculated, which equations were used. In some instances, manufacturers will measure sound data on one unit and will publish data for other models, sizes, or speeds based on scaling relationships. This is standard practice among silencer manufacturers. The precise methodology and measurement techniques are important to learn, to confirm the appropriateness and applicability of the data. Occasionally manufacturers publish data with substantial errors. Here again a comparison to generic formulas can help uncover inconsistencies. Even when data have been measured directly on one unit, there can be some variation in levels due to the production process and details of the installation. The actual sound power level, based on carefully measured data, can still vary by a few dB in a given band from unit to unit, even under ideal conditions.

Airborne Calculations

If we have the sound power level of a source we can calculate the sound pressure level at the location of interest. When we are inside a room and sufficiently far from the source the reverberant field will predominate and we can proceed using Eq. 8.83 to predict the airborne sound pressure level in a space. If there is a significant contribution from a direct field component, such as when equipment is located outdoors, a separate calculation should be carried out using Eq. 2.74. If both types of field contribute, the levels from each should be combined.

When the sound wave cannot expand, such as when it is contained within a duct, there is no attenuation due to geometrical spreading. Instead the attenuation due to the duct lining or other elements in the ductwork is subtracted from the overall power, in each band, before the sound is introduced into a room. There it is analyzed as any other source would be.

13.2 NOISE GENERATED BY HVAC EQUIPMENT

Compilations of sound power level data radiated by mechanical equipment have been published and are sometimes available from equipment manufacturers. One of the best was developed in the 1960s by Laymon Miller (1968) for the U.S. Army. He continued this work through the 1980s and his studies are excellent references. The American Society of

TABLE 13.1 Sound Pressure Levels at 3 ft from Packaged Chillers, dB (Miller, 1980)

	Rotary Screw	Reciprocating		Centrifugal	
Freq	Compressor	10-50 Tons	51-200 Tons	< 500 Tons	> 500 Tons
31 Hz	70	79	81	92	92
63 Hz	76	83	86	93	93
125 Hz	80	84	87	94	94
250 Hz	92	85	90	95	95
500 Hz	89	86	91	91	93
1 k Hz	85	84	90	91	98
2 k Hz	80	82	87	91	98
4 k Hz	75	78	83	87	93
8 k Hz	73	72	78	80	87
A-Weighted	90	89	94	97	103

Heating Refrigeration and Air Conditioning Engineers, ASHRAE, also publishes data on fans, pumps, and air handlers, and is another good source. Other manufacturers and trade associations make available data on specific pieces of equipment.

Refrigeration Equipment

Miller (1980) has collected and studied noise data on nearly 40 packaged chillers and reciprocating compressors. These units ranged in size from 15 tons to more than 500 tons of cooling capacity. A ton of refrigeration capacity is defined as the amount of heat removal required to produce one ton of ice from water at 32° F (0° C), 288,000 Btu (84.5 kW), in 24 hours or 12,000 Btuh (3.52 kWh). In air handling systems, fans generally are sized to provide about 400 cfm/ton of refrigeration. Sound data are given in terms of the sound pressure level at 3 ft. (1 m) from the equipment. No information on the physical size of the equipment is available. Several types of packaged chillers were investigated, differing primarily in the type of compressor. Table 13.1 shows the sound pressure levels at 3 ft. (1 m) for each type.

Cooling Towers and Evaporative Condensers

Cooling towers serve to cool water by using the latent heat absorbed during the process of evaporation. Water is introduced at the top of a cooling tower and falls to the bottom. Simultaneously air is blown or drawn upward through the falling water to aid in the mixing and increase evaporation. Noise is generated primarily by the fans; however, in certain cases the water itself can also contribute. Figure 13.4 shows examples of various types of cooling towers.

Overall sound power levels for each type are listed in Table 13.2 along with corrections to be subtracted from the overall level to obtain the level for each octave band. Cooling towers have a definite directivity, which depends on the type of fan, its location, and the side in question. Table 13.3 gives the approximate directional corrections to be added to the sound pressure levels calculated from the sound power levels in Table 13.2.

Note that these sound power level data have been calculated from sound pressure level measurements that are taken sufficiently far away from the unit that consideration of the size

FIGURE 13.4 Principal Types of Cooling Towers (Miller, 1980)

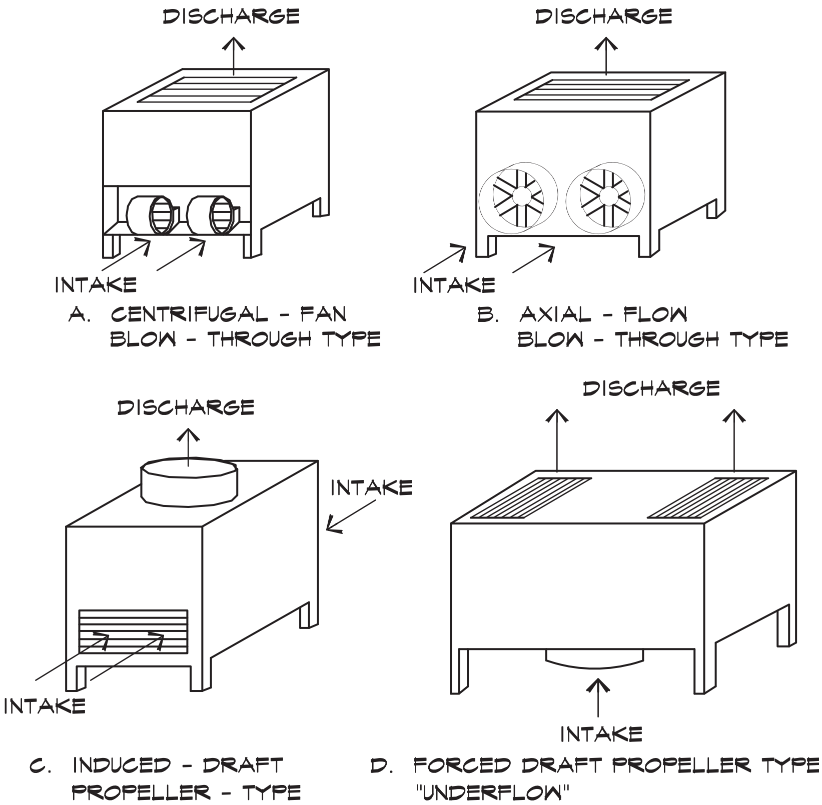


TABLE 13.2 Sound Power Levels of Cooling Towers, dB (Miller,1980)

	Propeller Type	Centrifugal Type
	$L_W = 95 + 10 \log(\text{fan hp}) - \text{Corr}$	$L_W = 85 + 10 \log(\text{fan hp}) - \text{Corr}$
31 Hz	8	6
63 Hz	5	6
125 Hz	5	8
250 Hz	8	10
500 Hz	11	11
1 k Hz	15	13
2 k Hz	18	12
4 k Hz	21	18
8 k Hz	29	25

TABLE 13.3 **Corrections to Average Sound Pressure Levels for the Directivity of Cooling Towers, dB (Miller, 1980)**

Octave Band (Hz)	31	63	125	250	500	1k	2k	4k	8k
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

was unnecessary for the pressure to power conversion. If near field sound pressure levels are needed, then the physical size of the source must be taken into account by using Eq. 2.91.

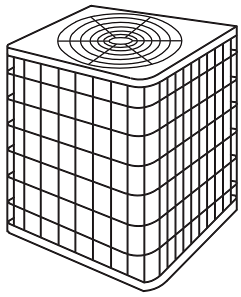
Air Cooled Condensers

In single or multifamily residences, air cooled condensers are used in place of the larger cooling towers or evaporative condensers. The noise from these units is due to the fan, usually a propeller type, with a small contribution from the air flow through the condenser coil decks. Figure 13.5 shows a sketch and measured data. These data are for a 3–5 ton residential unit based on sound pressure levels measured at 6 ft from the center of the fan and at 90° to the direction of airflow, which is out the top of the unit.

Pumps

Pumps are found in virtually every building. When they are located above grade they are best mounted on an inertial base and housekeeping pad, such as that in Fig. 13.6. Pipe elbows that are connected to the pump should be supported from the isolation base. Flexible couplings are used to compensate for pipe misalignment and to provide structural decoupling. Piping should be resiliently supported in accordance with the recommendations given in Chapt. 11.

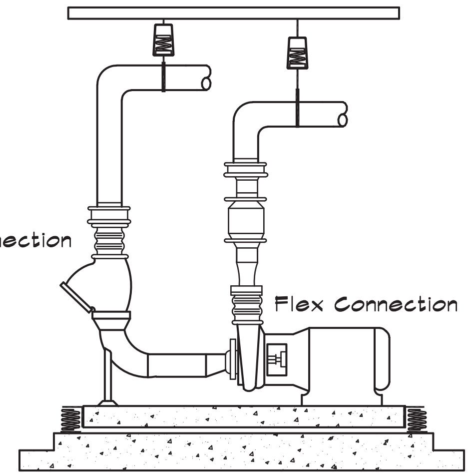
FIGURE 13.5 Sound Power Levels from a 3–5 Ton Air Cooled Compressor



Octave Band (Hz)	63	125	250	500	1k	2k	4k	8k
Level	87	84	75	75	74	71	64	56

FIGURE 13.6 Water Pump Installation with Inertial Base

Note that the piping is supported off the inertial base.



Sound pressure level data at a distance of 3 ft have been published by Miller (1980) and are reproduced in Table 13.4. Also shown in the table are the corrections to be subtracted from the overall level to obtain the octave band values. A-weighted levels are 2 dB lower than the overall levels.

13.3 NOISE GENERATION IN FANS

Noise in HVAC systems is created both actively by mechanical equipment, primarily fans, and passively by static components in the air stream, which can create flow-generated noise. Nearly every component in an HVAC system, no matter how benign, can contribute to noise creation. It is critical to be aware of the noise generating mechanisms and their relative impact on the overall system.

TABLE 13.4 Overall Sound Pressure Levels at 3 ft for Pumps (Miller, 1980)

Speed Range (rpm)	Drive Motor Nameplate Power Under 100 hp	Above 100 hp
	Overall Sound Pressure 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)

Corrections to Overall SPL for Pumps, dB

Octave Band (Hz)	31	63	125	250	500	1k	2k	4k	8k
Level Subtracted	13	12	11	9	9	6	9	13	19

Fans

All buildings have fans of one sort or another for air circulation. Fans are typed according to the mechanism used to propel the air in Fig. 13.7, and further subdivided according to the type of blade in Fig. 13.8. The basic types are axial and centrifugal. Axial fans are the simplest to understand; they have a fixed-pitch multiple-bladed rotor. Propeller fans are unhoused, whereas vane axial and tube axial fans include a shroud or housing around the impeller.

FIGURE 13.7 Types of Fans

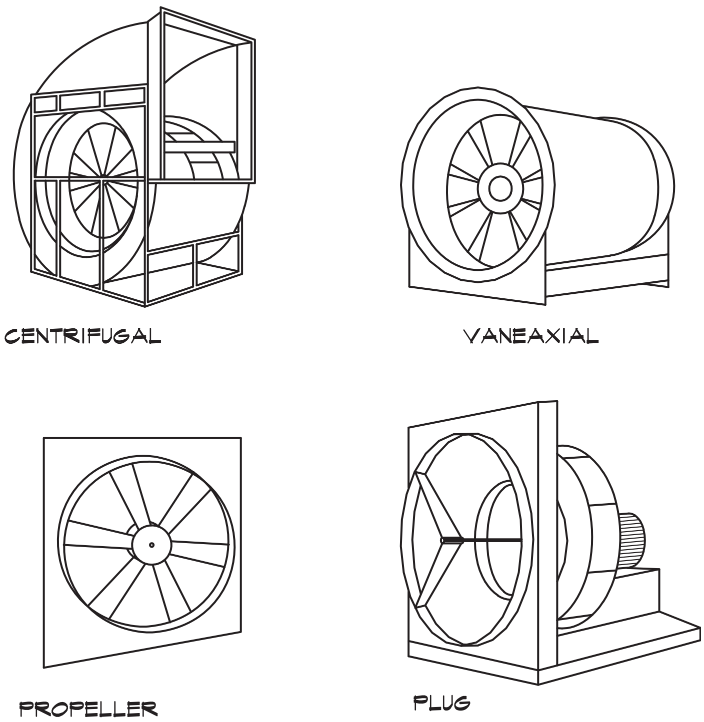
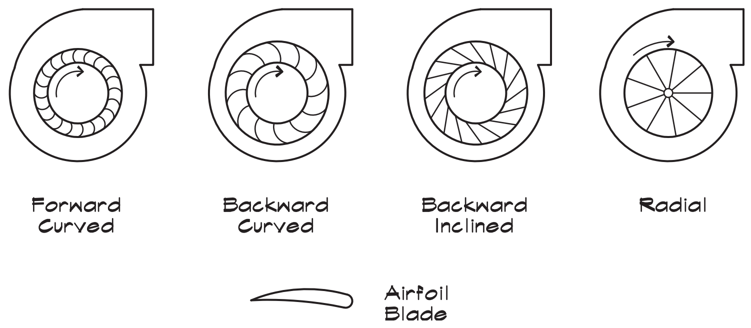


FIGURE 13.8 Types of Centrifugal Fans



Vane axial fans have fixed stator blades to straighten the flow after it passes through the rotor blades; tube axial fans do not.

Centrifugal fans consist of a series of blades, arranged at even intervals around a circle like a waterwheel, that throw the air from the inside to the outside of the circle as they rotate. Forward curved blades push the air out much like a jai alai racket. The air leaves the fan blade at a velocity higher than that of the blade tip. In backward-curved or backward-inclined blades the air velocity is lower than the tip velocity, so a lower noise level is generated. The forward-curved blades can generate the same air volume at a lower rotational speed, which means that the peak in their spectrum occurs at a lower frequency.

Fan noise is generated by several mechanisms, including the surge of the air pressure and velocity each time a blade passes, turbulent airflow in the air stream, and physical movement of the fan casing or enclosure. The noise emitted by each fan type follows a series of generalized laws called scaling laws, originally developed by Beranek, which have the general form (Graham, 1975 as given in ASHRAE, 1987)

$$L_w = K_F + 10 \log Q_F / Q_{REF} + 10 \log P_F / P_{REF} + C_{EFF} + C_{BFI} \tag{13.1}$$

where L_W = sound power level (dB re 10^{-12} Watts)
K_F = spectral constant which depends on the type of fan (dB) shown in Fig. 13.8
Q_F = volume of air per time passing through the fan (cfm or L/s)
Q_{REF} = reference volume (1 for cfm or 0.472 for L/s)

P_F = static pressure produced by the fan (in of water or Pa, gage)
P_{REF} = reference pressure (1 for in of water or 249 for Pa)

C_{EFF} = efficiency correction factor (dB)
C_{BFI} = blade frequency increment correction (dB)

Table 13.6 gives the off-peak the efficiency correction factor for fans running at less than peak efficiency. Equation 13.2 gives the method for calculating the fan's efficiency in FP units

$$\eta = \frac{100\, Q_F\, P_F}{6356\, W_{hp}} \tag{13.2}$$

TABLE 13.5 Level Correction K_F for Total Sound Power of Fans, dB (ASHRAE, 1987)

Fan Type	Octave Band Center Frequency (Hz)							
Centrifugal	63	125	250	500	1k	2k	4k	8k
Airfoil, Backwards Curved, Backward Inclined Wheel Diameter (inches)								
> 36 in	40	40	39	34	30	23	19	17
< 36 in	45	45	43	39	34	28	24	19
Forward Curved								
All	53	53	43	36	36	31	26	21
Radial Total Pressure (in. w.g.)								
Low 4–10	56	47	43	39	37	32	29	26
Med 6–15	58	54	45	42	38	33	29	26
High 15–60	61	58	53	48	46	44	41	38
Vaneaxial Hub Ratio								
0.3–0.4	49	43	43	48	47	45	38	34
0.4–0.6	49	43	46	43	41	36	30	28
0.6–0.8	53	52	51	51	49	47	43	40
Tubeaxial								
Wheel Diameter (inches)								
> 40 in	51	46	47	49	47	46	39	37
< 40 in	48	47	49	53	52	51	43	40
Propeller								
General ventilation and Cooling towers								
All	48	51	58	56	55	52	46	42

TABLE 13.6 Efficiency Corrections, C_{EFF} (ASHRAE, 1987)

Static Efficiency (% of Peak)	Correction Factor (dB)
90–100	0
85–89	3
75–85	6
65–74	9
55–64	12
50–54	15
below 50	16

TABLE 13.7 Blade Frequency Increment Correction, C_{BFI} (ASHRAE, 1987)

Fan Type	Blade Passing Octave, f_{bp}	C_{BFI}
Centrifugal		
Airfoil, backward curved, backward inclined	**250 Hz**	**3**
Forward curved	**500 Hz**	**2**
Radial blade pressure blower	**125 Hz**	**8**
Vaneaxial	**125 Hz**	**6**
Tubeaxial	**63 Hz**	**7**
Propeller		
Cooling Tower	**63 Hz**	**5**

where W_{hp} = power rating of the fan in horse power. The relative fan efficiency expressed as a percentage is

$$\eta_{rel} = 100 \left[\frac{\eta}{\eta_{peak}} \right] \tag{13.3}$$

If the peak efficiency is not known, it is normal to assume a relative efficiency of about 80% , a value that adds about 6 dB to the data. If the peak efficiency is available from the manufacturer usually the actual sound power levels are as well.

The additional factor known as the blade frequency increment correction C_{BFI}, shown in Table 13.7, is a number to be added to the overall level in the octave band containing the blade passing frequency, f_{bp}

$$f_{bp} = \frac{\text{fan rpm} \times \text{number of blades}}{60} \tag{13.4}$$

The sound is radiated from both the fan intake and discharge. The formula assumes ideal inlet and outlet flow conditions and operation of the fan at a given efficiency.

Fans can also radiate noise through their enclosures and into the surrounding space. This is referred to as casing radiation and may be calculated by subtracting a factor for the insertion loss of the casing. Insertion losses are very dependent on the gauge and construction of the fan housing and those cited in Table 13.8 are only approximate.

Housing attenuation values are subtracted from the fan sound power level data to obtain a rough estimate of the power levels radiated by the fan through the housing when the fan is attached to ductwork. At low frequencies the power is unaffected by the casing since the enclosure vibration radiates as much noise as the unhoused fan would. The attenuations reflect the assumption that there is no separate enclosure around the fan housing and no absorption inside the housing, but that there is a silencer or lining in the ductwork close to the fan.

TABLE 13.8 Adjustments for the Attenuation of the Fan Housing, dB (Miller, 1980)

	Octave Band Center Frequency (Hz)								
	31	63	125	250	500	1k	2k	4k	8k
Attenuation	0	0	0	5	10	15	20	22	25

Fan Coil Units and Heat Pumps

In small offices and residential installations split HVAC systems are often used. These consist of an air-cooled condenser outdoors and a fan coil indoors as illustrated in Fig. 13.2. A refrigerant is circulated between the two devices moving under pressure in mostly liquid form between the condenser and the fan coil, and returning as a gas. The high-pressure liquid is forced through an expansion valve and thence into a cooling coil, where heat is removed from the room air.

The main source of noise in a fan coil is the small fan that circulates air through the coil and into the conditioned space. Figure 13.9 gives measured sound power levels at several rates of flow. When a fan coil unit is located above a T-bar ceiling the noise it generates is difficult to control even with lined ductwork or silencers. Wherever possible fan coils should be installed in closets or above drywall ceilings, which provide the necessary transmission loss. Lined ducts or silencers are usually required.

The compression expansion cycle can be used to heat as well as cool a space. The process is reversible in a device known as a heat pump, which literally can carry heat into or out of a building. Figure 13.10 illustrates this process. In the winter or heating mode, at the top of the figure, refrigerant is circulated through a refrigerant-to-water heat exchanger, where it absorbs heat from water that is colder than the exterior environment but warmer than the refrigerant. The heat absorbed warms the refrigerant and converts it to a gas. It then

FIGURE 13.9 Discharge Noise Levels of Fan Coil Units (Fry, 1988)

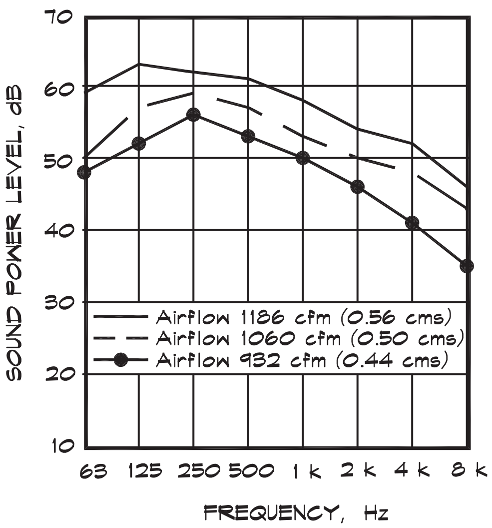
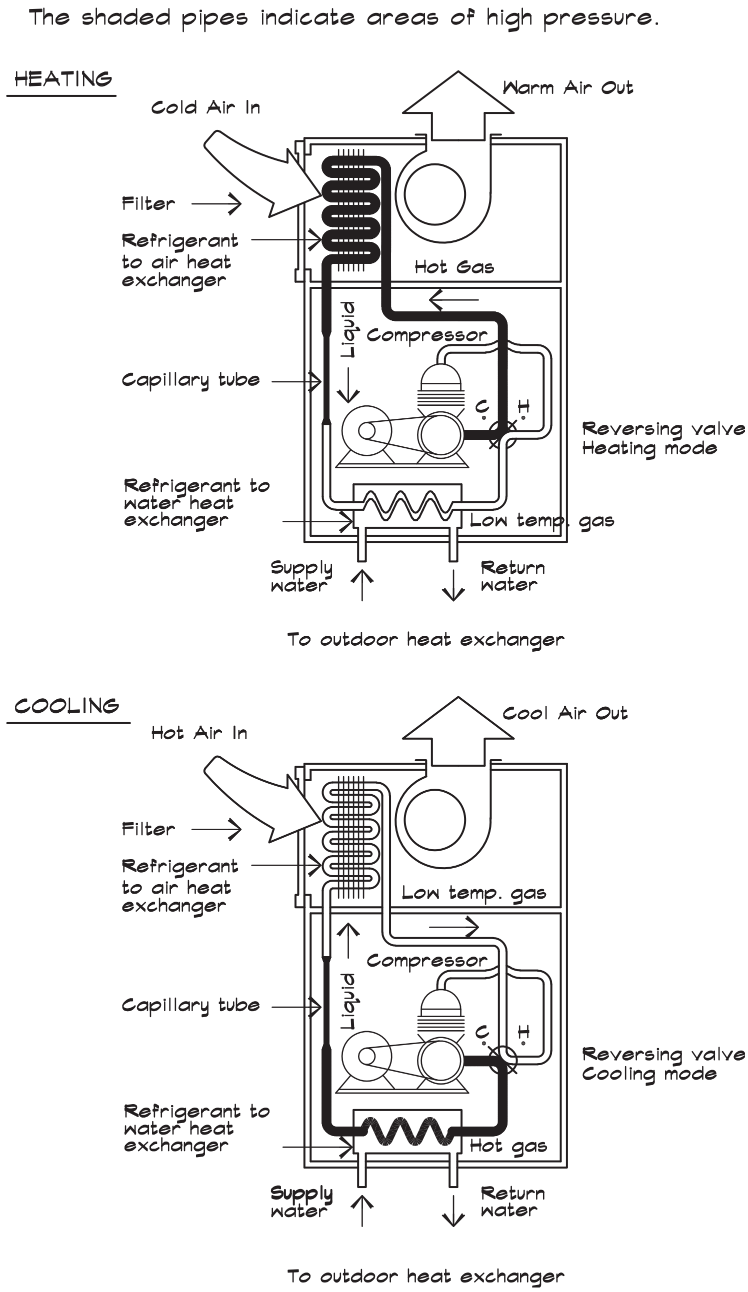


FIGURE 13.10 Air and Heat Flow in a Heat Pump System (California Heat Pump)



flows to the compressor, which further warms and pressurizes it by performing mechanical work on it. The hot gas then flows through a coil where a fan blows air over it and into the occupied space. The gas gives off heat in the exchange and condenses into a liquid. The liquid is then forced under pressure through the capillary tube where it expands and is returned to the heat exchanger. The cooling cycle is just the reverse of this process and is enabled by changing the direction of flow of the reversing valve. Noise generated by heat pump units can be greater than fan coils since the compressor is located in the same unit as the coil.

VAV Units and Mixing Boxes

In recent years, due to the emphasis being placed on energy conservation, the variable air volume or VAV system has become a commonly used design. A VAV system consists of a fan operating at a constant velocity that pressurizes a series of valves, which in turn feed a network of diffusers, distributed throughout the occupied space. The valves consist of remotely controlled dampers, which regulate the airflow to each space. A bypass duct is used to route the unused air back to the inlet side of the air handler in order to maintain a constant volume through the fan. A VAV unit, pictured in Fig. 13.11, must be capable of regulating the airflow from the full design capacity down to a very small flow, usually by means of butterfly dampers.

Blazier (1981) has published discharge (in Fig. 13.12) and radiated (in Fig. 13.13) sound power levels generated by VAV units for two rates of flow. The noise is generated by disturbed flow around the dampers.

Many manufacturers publish data on both discharge and casing radiated sound from VAV units. The most useful data are given in terms of sound power levels; however, some manufacturers list data in terms of NC levels, which are obtained by assuming a certain

FIGURE 13.11 A Variable Air Volume Unit

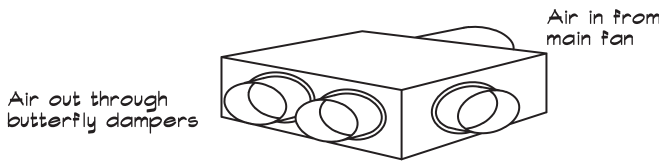


FIGURE 13.12 Range of Discharge VAV Noise Levels at Two Operating Points (Blazier, 1981)

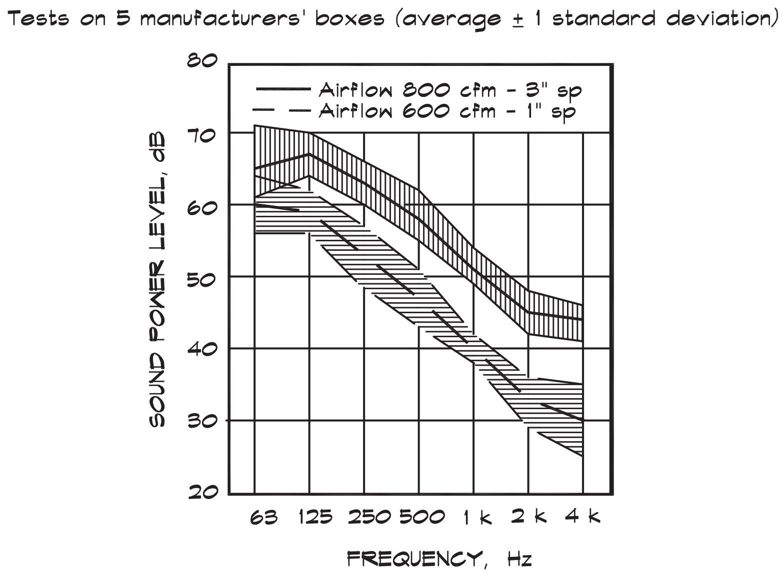


FIGURE 13.13 Range of Radiated VAV Noise Levels at Two Operating Points (Blazier, 1981)

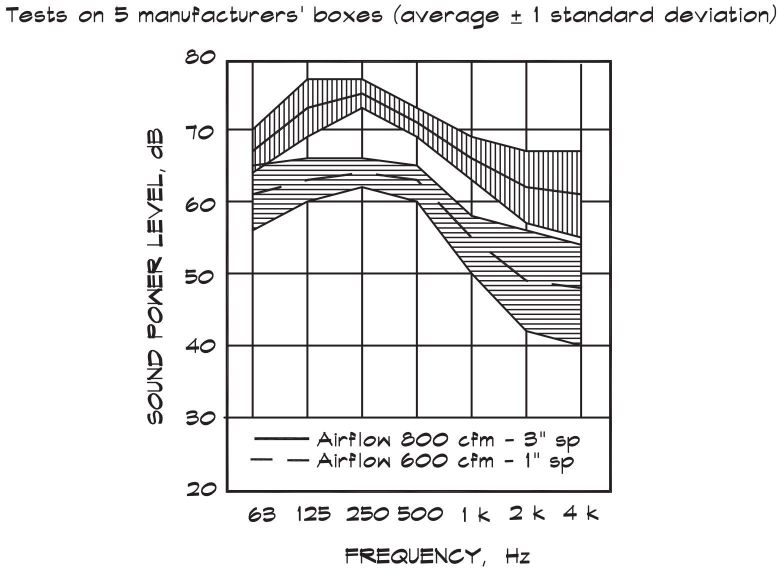
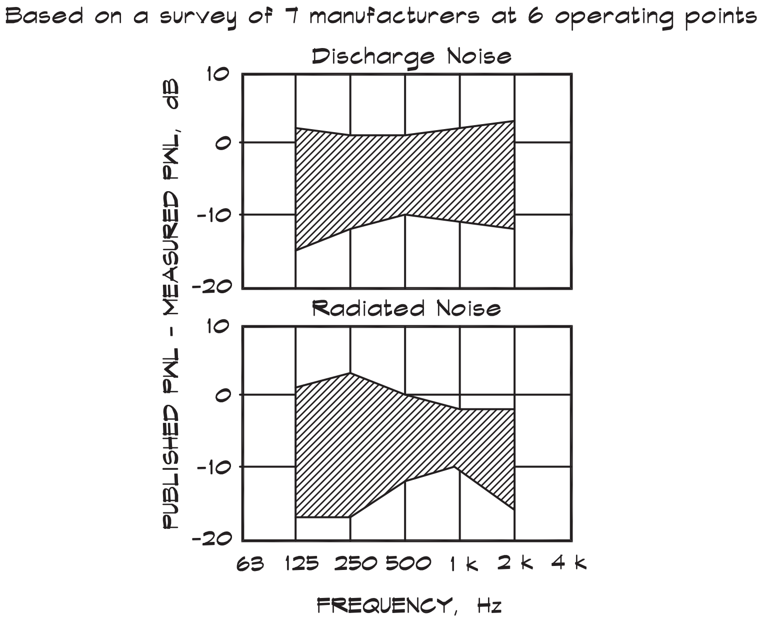
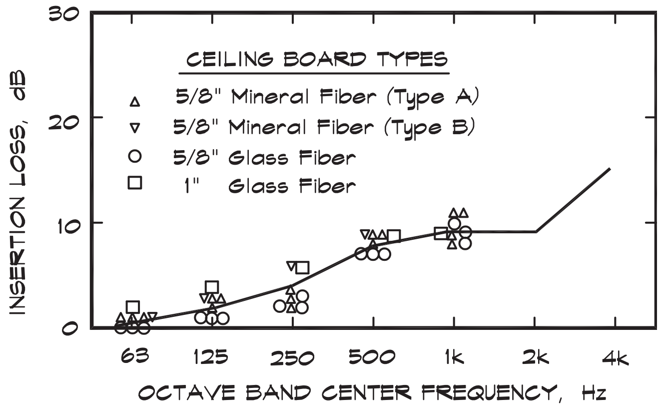


FIGURE 13.14 Comparison between Published and Measured Sound Power Levels (Blazier, 1981)



power-to-pressure conversion in the receiving room (usually −10 dB) and sometimes an additional noise reduction (also −10 dB) due to the ceiling tile. Blazier (1981) has measured the noise generated by VAV units as they compare to data published by the manufacturers. He lists in Fig. 13.14 the difference between published and measured sound power levels citing a 6 to 10 dB understatement of the noise furnished by manufacturers. Some of the

FIGURE 13.15 Measured Insertion Loss of Typical Lay-in Acoustical Ceiling Systems (Blazier, 1981)



discrepancy may be due to the difficulty in duplicating in the field the smooth entry and exit flow conditions under which the laboratory data are taken. Blazier (1981) has also measured the insertion loss due to acoustical tile ceilings given in Fig 13.15. Note that the loss approaches 10 dB only at high frequencies.

13.4 NOISE GENERATION IN DUCTS

Flow Noise in Straight Ducts

Once air has been set in motion it can generate noise by creating pressure fluctuations through turbulence, vortex shedding, mixing, and other mechanisms. Steady flow in a straight duct does not generate appreciable noise, when compared with other sources such as abrupt transitions in the air path, takeoffs, and elbows. Figure 13.16 (Fry, 1988) shows sound power

FIGURE 13.16 Sound Power Spectra of 600 mm (24 in) × 600 mm Straight Steel Duct for Various Air Velocities (Fry, 1988)

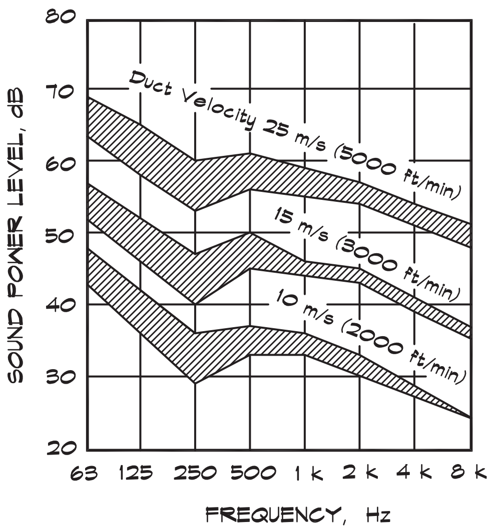
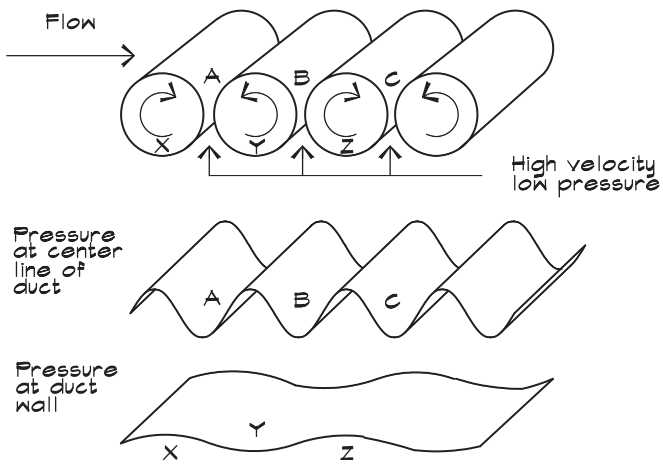


FIGURE 13.17 Buffeting in Rectangular Ducts (Fry, 1988)



level data on noise generated in straight duct runs for straightened flow. Levels generally follow an 18 dB per doubling of velocity scaling law.

The cited data show a significant rise in level when the cross sectional duct dimension is equal to a wavelength, which for this example is about 500 Hz. This bump coincides with the establishment of full cross duct turbulent eddies illustrated in Fig. 13.17. Eddies form downstream of disturbing elements such as rods or dampers. By themselves eddies are not particularly efficient sound radiators; however, they can generate noise when they impinge on a flat plate or other low-frequency radiator. In an open duct, eddies cause the flow to alternately speed up and slow down, producing pressure maxima at points X and Z and a pressure minimum at point Y.

In air distribution design it is prudent to control the duct velocity by increasing the cross sectional area of the duct, thus slowing the flow as it approaches the space served. A duct layout is pictured in Fig. 13.18.

FIGURE 13.18 Typical Duct Run in an HVAC System

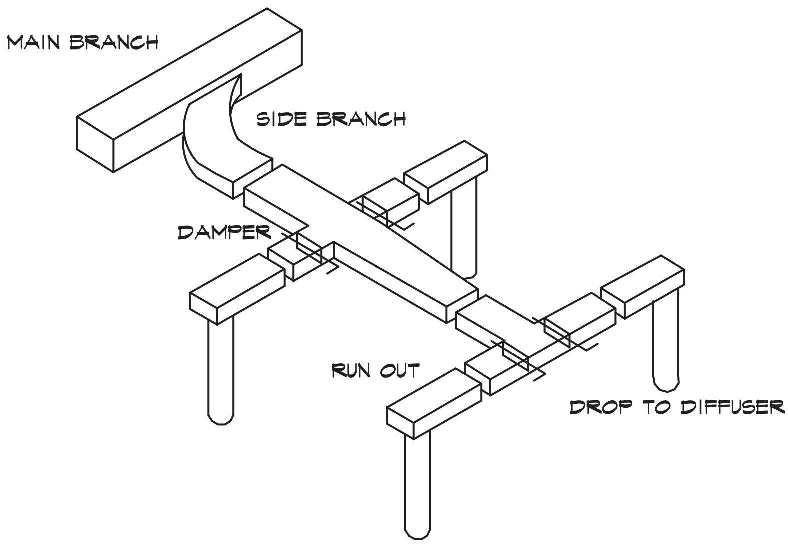


TABLE 13.9 Velocity Criteria for Air Distribution Systems

		Maximum Air Velocities (ft/min)					
Description		Distance from Termination					
	Slot Speed at Termination	< 10 ft (3 m) No Lining	> 5 ft (1.5 m)	> 10 ft (3 m)		> 20 ft (6 m)	
NC Criteria		< 5 ft (1.5 m) Lining*	Lining*	No Lining	Lining*	No Lining	Lining*
NC 15 supply	250	300	500	350	800	425	1000
NC 15 return	300	350	600	350	950	500	1200
NC 20 supply	300	350	600	425	950	550	1200
NC 20 return	350	425	725	500	1150	650	1450
NC 25 supply	350	425	725	500	1150	700	1450
NC 25 return	425	500	875	650	1375	800	1725
NC 30 supply	425	500	875	700	1375	850	1725
NC 30 return	500	600	1050	800	1650	950	2075
NC 35 supply	500	600	1050	800	1650	1000	2075
NC 35 return	600	700	1250	900	2000	1150	2500
NC 40 supply	600	700	1250	900	2000	1150	2500
NC 40 return	725	850	1500	1075	2400	1380	3000
NC 45 supply	725	850	1500	1075	2400	1375	3000
NC 45 return	875	1000	1800	1300	2875	1675	3575

* Duct must be lined with 1" fiberglass duct liner or flexible duct from this point to termination.

1 m/sec = 196.8 ft/min

FIGURE 13.19 Sound Power Levels of Abrupt and Gradual Area Transitions from 600 mm × 600 mm to 200 mm × 200 mm Duct Cross Sections (Fry, 1988)

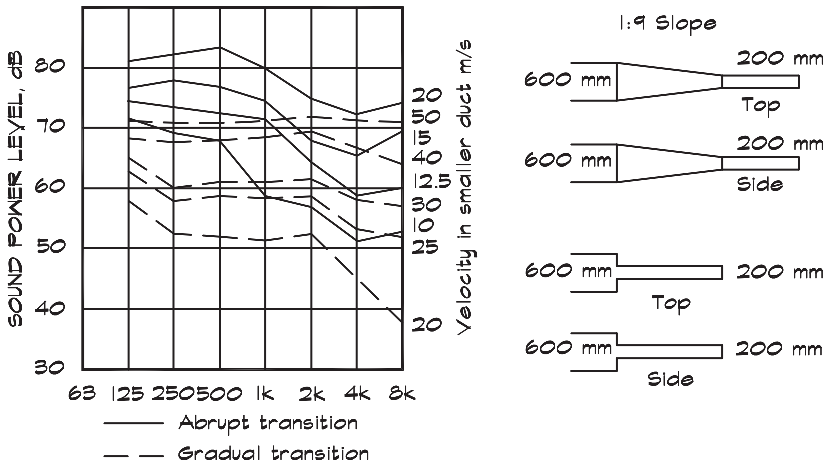


Table 13.9 gives velocity recommendations appropriate for various NC levels in the receiving space. These include consideration of the attenuation of downstream sections of lined duct. For the lined duct velocities, it is assumed that the remaining ductwork is covered on the inside with a 1" (25 mm) thick fiberglass duct liner or is flex duct.

Noise Generated by Transitions

Aerodynamic noise is generated at both gradual and abrupt changes in duct area. Gradual transitions and low velocities generate less turbulence than abrupt transitions and high velocities. Beyond these generalizations there are few models to use for sound power level prediction. Fry (1988) has published measured data for several expansion ratios in rectangular ducts, which are reproduced in Figs. 13.19 and 13.20.

Air Generated Noise in Junctions and Turns

Noise generated in transition elements such as turns, elbows, junctions, and takeoffs can run 10 to 20 dB higher than the sound power levels generated in straight duct runs. Ducts having radiused bends, with an aspect ratio of 1:3 generated, no more noise than a straight duct (Fry, 1988). Elbows having a 90° bend are about 10 dB noisier than straight duct. One or more turning vanes can reduce the noise 8 to 10 dB at high and low frequencies while increasing it 3 to 4 dB in the mid frequencies.

Ver (1984) published an empirical equation for the sound power levels given off by various fittings, which was reproduced in the 1987 ASHRAE guide. For branches, turns (including elbows without turning vanes), and junctions such as those pictured in Fig. 13.21, it is

$$L_{W\,OCT}(f_0) = K_J + 10 \log (f_0/63) + 50 \log (U_B) + 10 \log (S_B) + 10 \log (D_B) + C_B + \Delta r + \Delta T \tag{13.5}$$

FIGURE 13.20 Sound Power Levels of Abrupt and Gradual Area Transitions from 600 mm × 600 mm to 600 mm × 200 mm Duct Cross Sections (Fry, 1988)

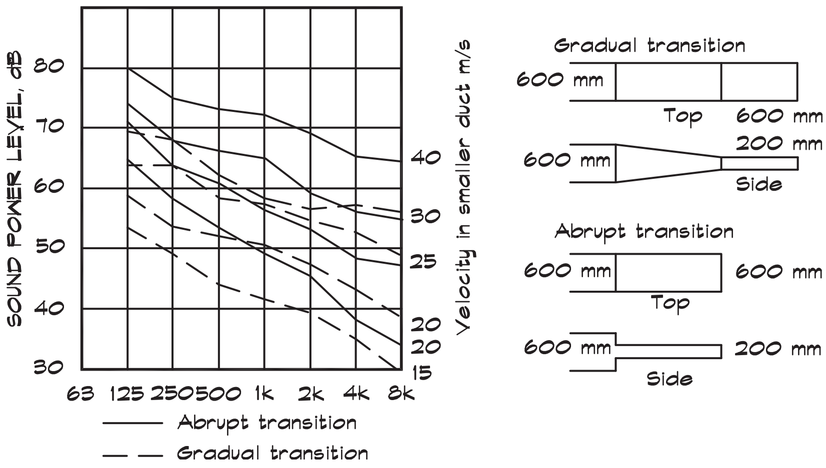
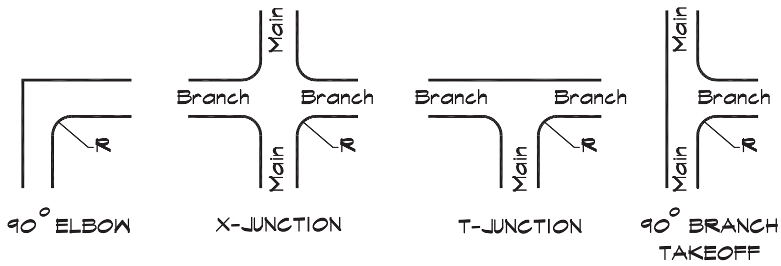


FIGURE 13.21 Elbows, Junctions, and Branch Takeoffs



where $L_{W\ OCT}$ = octave band sound power level (dB re 10^{-12} Watts)
f_0 = center frequency of the octave band (Hz)
K_J = characteristic spectrum of the junction or turn, based on the Strouhal number
U_B = velocity in the branch duct (ft/s)
S_B = cross sectional area of the branch duct (ft^2)
D_B = equivalent diameter of the branch duct or in the case of junctions, $D_B = \sqrt{4\ S_B/\pi}$ (ft)
C_B = a constant which depends on the type of branch or junction

The flow velocity in the branch is calculated using

$$U_B = Q_B/(60\ S_B)\ (\text{ft/s}) \tag{13.6}$$

where Q_B = flow volume in the branch, (cfm)

The term Δr is a correction for the roundness of the bend or elbow associated with the turn or junction (Reynolds, 1990)

$$\Delta r = \left\{ 1.0 - \frac{R_D}{0.15} \right\} \left[6.793 - 1.86 \log (S_t) \right] \tag{13.7}$$

where R_D is the rounding parameter $R_D = \dfrac{R}{12\, D_B}$ for the radius, R (inches), of the inside edge of the bend, and $S_t = f_0\, D_B / U_B$ is the Strouhal number.

The term ΔT is a correction for upstream turbulence, which is applied only when there are dampers, elbows, or branch takeoffs upstream, and within five main duct diameters, of the turn or junction under consideration.

$$\Delta T = -1.667 + 1.8\, m - 0.133\, m^2 \tag{13.8}$$

$$m = U_M / U_B \tag{13.9}$$

where U_M = velocity in the main duct (ft/min)
U_B = velocity in the branch duct (ft/min)

The characteristic spectrum K_J in Eq. 13.5 may be calculated (Reynolds, 1990) using

$$\begin{aligned} K_J = & -21.61 + 12.388\, m^{0.673} \\ & - 16.482\, m^{-0.303} \log (S_t) \\ & - 5.047\, m^{-0.254} \log (S_t)^2 \end{aligned} \tag{13.10}$$

Finally the correction term C_B depends on the type of junction in Fig. 13.21.

For X - junctions

$$C_B = 20 \log \left[\frac{D_M}{D_B} \right] + 3 \tag{13.11}$$

and for T junctions

$$C_B = 3 \tag{13.12}$$

For 90° elbows without turning vanes,

$$C_B = 0 \tag{13.13}$$

For a 90° branch takeoff,

$$C_B = 20 \log \left[\frac{D_M}{D_B} \right] \tag{13.14}$$

Air Generated Noise in Dampers

Damper noise follows the same general spectrum equation (Eq. 13.5) as was used for branches and turns, although the terms are defined somewhat differently

$$L_{W\ OCT}(f_0) = K_D + 10\log(f_0/63) + 50\log(U_C) + 10\log(S) + 10\log(D_H) + C_D \tag{13.15}$$

where
$L_{W\ OCT}$ = octave band sound power level (dB re 10^{-12} Watts)
f_0 = center frequency of the octave band (Hz)
K_D = characteristic spectrum of the damper, based on the pressure loss factor and the Strouhal number
U_C = flow velocity in the constricted part of the duct (ft/s)
S = cross sectional area of the branch duct (ft^2)
D_H = duct height normal to the damper axis (ft)

Before solving Eq. 13.15, several preliminary calculations must be undertaken. The characteristic spectrum is determined from the Strouhal number, which depends on the velocity, the blockage factor, and the pressure loss coefficient, C.

The pressure loss coefficient in FP units is

$$C = 15.9 \cdot 10^6 \frac{\Delta P}{(Q/S)^2} \tag{13.16}$$

where
ΔP = pressure drop across the fitting (in. w.g.)
Q = flow volume, (cfm)

The blockage factor, B, for multiblade dampers and elbows with turning vanes is

$$B = \frac{\sqrt{C} - 1}{C - 1} \quad \text{for } C \neq 1 \tag{13.17}$$

or

$$B = 0.5 \quad \text{if } C = 1 \tag{13.18}$$

For single blade dampers, it is

$$B = \frac{\sqrt{C} - 1}{C - 1} \quad \text{if } C < 4 \tag{13.19}$$

or

$$B = 0.68\, C^{-0.15} - 0.22 \quad \text{if } C > 4 \tag{13.20}$$

Next the constricted velocity is calculated using

$$U_c = Q/(60\ SB) \quad (\text{ft/sec}) \tag{13.21}$$

which gives the Strouhal number

$$S_t = f_0\, D / U_c \tag{13.22}$$

Having calculated these numbers for the particular fitting we can find the characteristic spectrum for dampers (Reynolds, 1990)

$$\begin{aligned} K_D &= -36.6 - 10.7 \log (S_t) \quad \text{for } S_t \le 25 \\ K_D &= -1.1 - 35.9 \log (S_t) \quad \text{for } S_t > 25 \end{aligned} \tag{13.23}$$

Air Noise Generated by Elbows with Turning Vanes

For elbows with turning vanes we use Ver's equation with a slightly different definition of the terms

$$\begin{aligned} L_{W\,OCT}(f_0) = K_T &+ 10 \log (f_0/63) + 50 \log (U_C) \\ &+ 10 \log (S) + 10 \log (D_C) + 10 \log n \end{aligned} \tag{13.24}$$

where $L_{W\,OCT}$ = octave band sound power level (dB re 10^{-12} Watts)
f_0 = center frequency of the octave band (Hz)
K_T = characteristic spectrum of an elbow with turning vanes
U_C = flow velocity in the constricted part of the flow field (ft/s)
S = cross sectional area of the elbow (ft^2)
D_C = chord length of a typical vane (in)
n = number of turning vanes

Figure 13.22 shows the definition of the chord length.

The characteristic spectrum (Reynolds, 1990) is

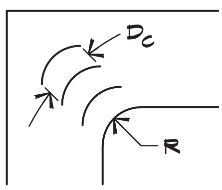
$$K_T = -47.5 - 7.69 \,[\log (S_t)]^{2.5} \tag{13.25}$$

where the Strouhal number in Eq. 13.22 is calculated from the pressure loss coefficient in Eq. 13.16, the blockage factor,

$$B = \frac{\sqrt{C} - 1}{C - 1} \tag{13.26}$$

and the constricted velocity in Eq. 13.21.

FIGURE 13.22 Ninety Degree Elbow with Turning Vanes



Grilles, Diffusers, and Integral Dampers

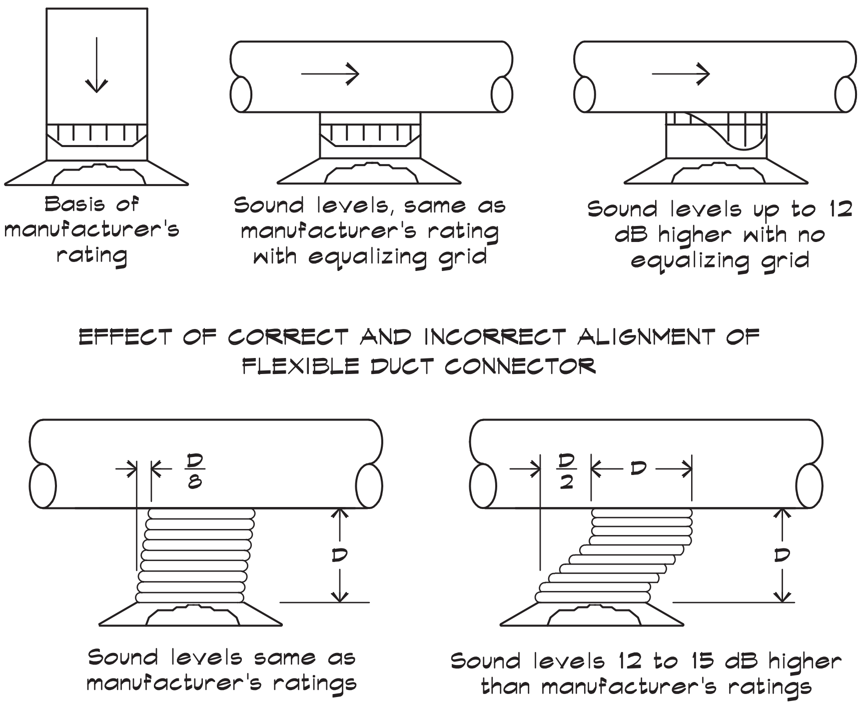
Diffuser generated noise is of paramount importance in HVAC noise control since it cannot be attenuated by the addition of downstream devices. Since diffuser noise is primarily dependent on the air velocity through the device, the only method for attenuating it is to reduce it, either by adding additional diffusers, or by increasing the size of the existing diffusers. Often diffuser noise is influenced by the upstream flow conditions that can be modified. Pressure equalizing grilles at the entry to the diffuser can help reduce the contribution due to turbulence.

Sound data on diffuser noise is published by manufacturers in terms of NC levels; however, these are only valid for ideal flow conditions. To achieve ideal conditions, flexible ducts must be straight for at least one duct diameter before the connection to the diffuser and must not be pinched or constricted. Figure 13.23 shows examples of correct and incorrect flexible duct connections.

It should be noted that manufacturer published data are given for one diffuser or, in the case of linear diffusers, for one four-foot long segment, with a power-to-pressure conversion of 10 dB, which corresponds to a room absorption of about 400 sabins and a distance of about 12 ft. The actual power-to-pressure conversion factor should be calculated for the specific room in question. Where there are multiple diffusers in a space, a factor of 10 log n, where n is the number of diffusers or the total number of four-foot segments of linear diffuser, must be added to the published noise levels.

When sound levels from diffusers are not available, they can be approximated using a general equation (Reynolds, 1990). It relates the diffuser noise to the sixth power of the flow

FIGURE 13.23 Correct and Incorrect Diffuser Installation (Fry, 1988)



velocity and the third power of the pressure drop.

$$L_w = 10 \log S_G + 30 \log \xi + 60 \log U_G - 31.3 \tag{13.27}$$

where L_W = overall sound power level (dB re 10^{-12} Watts)
S_G = cross sectional face area of the grille or diffuser (ft^2)
U_G = flow velocity prior to the diffuser (ft/s)
ξ = normalized pressure drop coefficient

It is clear that the noise emitted by diffusers is very dependent on the flow velocity and the formula yields an 18 dB per doubling of velocity relationship. For a given flow volume a doubling of grill area will reduce noise by 15 dB.

The normalized pressure drop is

$$\xi = 334.9 \frac{\Delta P}{\rho_0 U_G^2} \tag{13.28}$$

where ΔP = pressure drop across the diffuser (in w.g.)
ρ_0 = density of air (0.075 lb/ft^3)
U_G = flow velocity prior to the diffuser (ft/min)
$= \dfrac{Q}{60 S_G}$ (for Q in cfm)

The octave band sound power levels can be calculated from the overall level by adding a correction factor to Eq. 13.27

$$L_{W\,OCT} = L_W + C_D \tag{13.29}$$

The correction term for round diffusers is

$$C_D = -5.82 - 0.15A - 1.13 A^2 \tag{13.30}$$

and for rectangular (including slot) diffusers,

$$C_D = -11.82 - 0.15A - 1.13 A^2 \tag{13.31}$$

and is normalized to a peak frequency

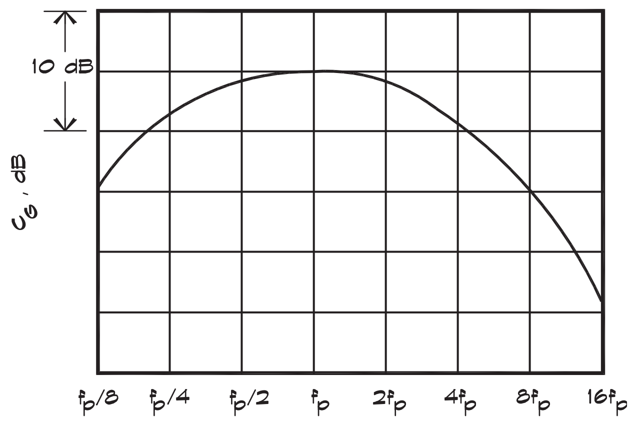
$$f_P = 48.8 U_G \tag{13.32}$$

The term A is

$$A = N_B (f_P) - N_B (f) \tag{13.33}$$

where $N_B(f)$ is the band number of the octave of interest and $N_B(f_P)$ is the band number of the octave where the peak frequency occurs. Octave band numbers are 0 for 32 Hz, 1 for 63 Hz, 2 for 125 Hz, and so forth. So, for example, if the peak frequency falls within the 125 Hz octave band, the band number is 2. The value of A for the 63 Hz octave band would be $A = 2 - 1 = 1$ and would decrease by 1 for each octave above that. The shape of the diffuser spectrum curve is given in Fig. 13.24.

FIGURE 13.24 Generalized Shape of the Diffuser Spectrum (ASHRAE, 1995)



Dampers located close to an outlet diffuser can add appreciably to the noise generated by the termination. First the damper generates vortex shedding in its wake, which is a source of noise. Second, downstream turbulence increases the noise generated by the grille. Manufacturers of these devices can provide sound power levels for a given flow volume. If these are not available it can be assumed that levels will increase 5 dB in all bands with the dampers in the fully open position. As the dampers are closed, there is an increase in the pressure drop across the damper, which restricts the flow. The overall sound power level increases approximately as (Fry, 1988)

$$\Delta L_w = 33 \log (\Delta P / \Delta P_0) \tag{13.34}$$

where ΔL_W = increase in sound power level radiated by the diffuser (dB)
ΔP = new static pressure drop across the unit (in. w.g.)
ΔP_0 = initial static pressure drop with the damper in place but with the vanes fully open with the same flow volume (in. w.g.)

Figure 13.25 shows the effect of integral dampers on noise radiated by ceiling diffusers for various settings. For dampers and grills to act as separate sources they should be located at least 4 duct diameters apart.

13.5 NOISE FROM OTHER MECHANICAL EQUIPMENT

Air Compressors

Air compressors in buildings generally fall into two categories: small units, under 5 hp, used to provide high pressure air for pneumatically controlled HVAC systems; and large units of up to 100 hp, which provide shop air to machine shops, laboratories, and maintenance areas. Miller (1980) has published sound pressure levels at 3 ft based on measurements of nine machines. Seven of these were reciprocating with motors ranging from 1 to 75 hp, and two were centrifugal, one of 10 hp and another 20 hp. Figure 13.26 reproduces his results. The principal source of air compressor noise is the air intake, which can be treated using a muffler between the filter and the intake manifold.

FIGURE 13.25 Ceiling Diffuser Damper—Sound Power Spectra (Fry, 1988)

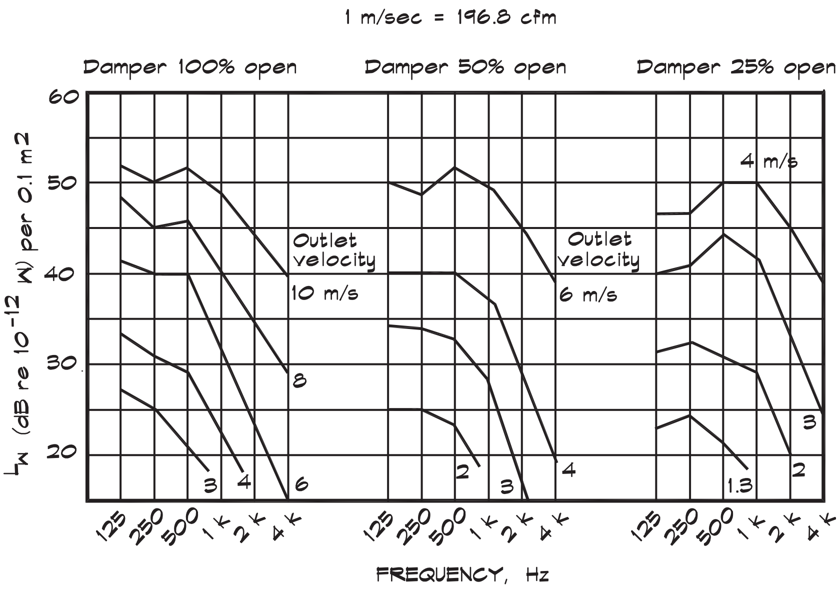
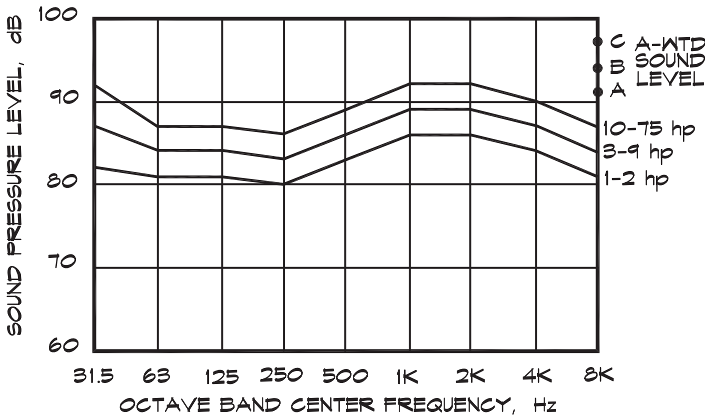


FIGURE 13.26 Sound Pressure Levels of Air Compressors at a Distance of 3 Feet (Miller, 1980)



Transformers

Transformers are usually located in an electrical equipment room, where they are sequestered from sensitive receivers. Ideally these rooms do not have common walls or floor-ceiling separations. Transformer noise is created through a process of magnetostriction, an expansion and contraction due to a magnetic field, caused by current within the coils. For a sinusoidal input voltage this phenomenon occurs twice every cycle, at 120 Hz, in single phase units and at harmonics of this frequency. Miller (1980) has published a conversion equation to obtain the sound power spectrum from the manufacturer's NEMA rating (the average of A-weighted sound pressure levels taken at a distance of 1 ft from an imaginary vertical surface passing

TABLE 13.10 Level Adjustments for the NEMA Rating of a Transformer, dB (Miller, 1980)

	Octave Band Center Frequency, Hz								
	31	**63**	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
C_T	-1	5	7	2	2	-4	-9	-14	-21

through a string tied around the unit), and the correction term, which includes the 10.5 dB adjustment for the power to pressure conversion. Table 13.10 gives the correction term for an unenclosed transformer.

$$L_w = \text{NEMA rating} + 10 \log S_T + C_T \tag{13.35}$$

where $L_{w\ OCT}$ = octave band sound pressure level (dB)
S_T = surface area of the transformer (ft^2)
C_T = octave band correction (dB)

Over time, transformers can grow noisier as their laminations and tie bolts become loose. Miller cites increases as large as 5 dB at the fundamental and 10 dB in the second and third harmonic frequencies. When transformers are enclosed in small vaults they can induce standing wave patterns in the room, which have the effect of increasing the transmitted power by 6 dB in the same bands.

Transformers that are directly tied to a wall can induce structure-borne noise. Jones (1984) recommends isolation techniques shown in Fig. 13.27 to prevent this. In areas of seismic activity, one or more sway braces may be necessary to provide stability at the top of the unit.

Reciprocating Engines and Emergency Generators

Most large buildings have emergency generators to provide power when the normal sources fail. It is often argued that noise control of emergency generators is unnecessary since they would be used only in an emergency, when noise is a secondary concern. Although this is probably true, generators must be tested periodically, an hour a month, and during these test periods the building functions normally and noise is still a concern. During power outages generators can be needed over longer periods of time.

Generator sets are powered by a diesel, methane, or propane fuel reciprocating engine and radiate sound from their casing, intake, and exhaust. Miller (1980) measured the casing radiated power levels, which followed the relationship

$$L_w = 93 + 10 \log (\text{rated hp}) + A + B + C + D \tag{13.36}$$

where L_w = overall sound power level (dB)
rated hp = engine manufacturer's continuous full load rating for the engine, (horse power)
A, B, C, D = correction terms given in Table 13.11 (dB)

Octave-band casing-radiated noise can be obtained from the overall sound power level spectrum by subtracting the levels given in Table 13.12.

FIGURE 13.27 Vibration Isolation of Floor-Mounted Transformers (Jones, 1984)

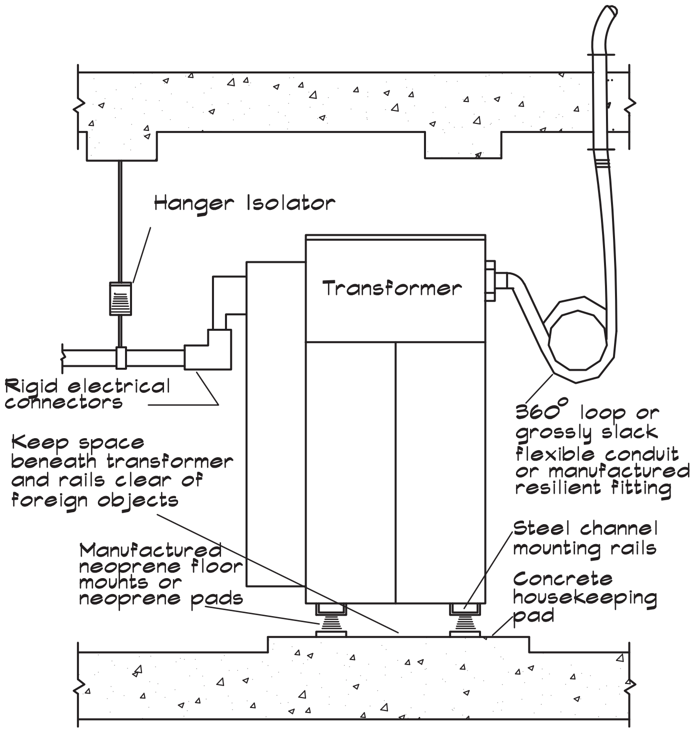


TABLE 13.11 Level Adjustments for Engine Casing Radiated Noise, dB (Miller, 1980)

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 amounts 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**

Noise radiated from the inlet is usually the same as the casing radiation unless there is a separate ducted inlet to a turbocharger. In these cases the inlet noise is given by

$$L_w = 94 + 5 \log (\text{rated hp}) \tag{13.37}$$

TABLE 13.12 Frequency Adjustments for Casing Radiated Noise of Reciprocating Engines (Miller, 1980)

Octave Frequency Band (Hz)	Value to be Subtracted from Sound Power Level, dB			
	Engine Speed Under 600 rpm	Engine Speed 600-1500 rpm: Without Roots Blower	Engine Speed 600-1500 rpm: With Roots Blower	Engine Speed Over 1500 rpm
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	4	3	1	2

TABLE 13.13 Level Adjustments for Turbocharger Air Inlet, dB (Miller, 1980)

	Octave Band Center Frequency (Hz)									
	31	63	125	250	500	1k	2k	4k	8k	A
Correction	4	11	13	13	12	9	8	9	17	3

Any losses due to inlet ductwork or silencers must be subtracted from the octave band sound power levels. The corrections for each octave band are given in Table 13.13 and are subtracted from the overall sound power level.

The exhaust is the loudest source. The overall sound power level for noise radiated from an unmuffled engine exhaust is

$$L_w = 119 + 10 \log (\text{rated hp}) - T \tag{13.38}$$

where the factor T is the turbocharger correction term (T = 0 dB for no turbocharger and T = 6 dB for an engine with a turbocharger). The effects of any downstream exhaust piping or mufflers must be subtracted from the sound power level in each band. Octave-band adjustments to be subtracted form the overall sound power level are shown in Table 13.14.

TABLE 13.14 Level Adjustments for Engine Exhaust, dB (Miller, 1980)

	Octave Band Center Frequency (Hz)									
	31	63	125	250	500	1k	2k	4k	8k	A
Correction	5	9	3	7	15	19	25	35	43	12