Design Advisory #9: CAS-DA9-2004

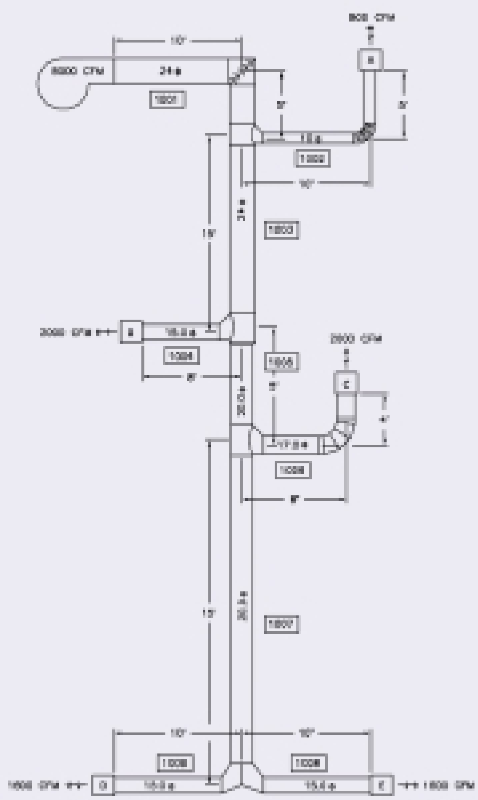
Designing Quietly

Hopefully by now, you've been able to review and familiarize yourself with the basic acoustical terminology that is presented in Chapter 7 of McGill AirFlow's *Duct System Design Guide*. Having a working knowledge of this terminology will help in understanding the acoustical concepts and design considerations involved with duct system design, which is covered in the next few chapters, starting with this release of "Chapter 8: Duct System Acoustics".

As mentioned earlier, many times the acoustics of a duct system is not adequately addressed if even considered at all. Just put in twenty feet of lined duct after the air handler and everything will be okay. Right? Not likely. Twenty feet of lined or double-wall duct may or may not be sufficient for attenuating the fan noise. With many of the noisy units being installed these days, it is probably not enough. Also, restrictions on the permissible noise levels, such as codes modeled after ANSI/ASA S12.60, *Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools,* require that an acoustical analysis be done for a duct system. Analysis should always precede the specification of the amount and type of silencers and acoustically treated duct.

To perform a proper analysis of a duct system, one must be familiar with the concepts of insertion loss, generated or self-noise, end reflections, break-out and break-in noise, and sound power splits. Chapter 8 shows how to account for the acoustical aspects of various components making up the duct system. A simple, single-wall duct system is analyzed to determine the outlet sound power levels. Chapters 9 and 10 then address the effects of acoustical treatments applied to this simple system.

Figure 8.2
Sample Problem 8-1
Chapter 8



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CHAPTER 8: Duct System Acoustics

There are two propagation paths that should concern the design professional. One is the noise which propagates through the duct system, and the other is the airborne noise radiated away from the duct into the surrounding spaces.

There is usually a tradeoff between these two propagation paths. A rigid and well-constructed duct system will contain much of the noise within the system. Usually this is desirable since there are several attenuation mechanisms available for noise propagating down a duct path. Nonmetallic duct systems, or those made of lightweight or poorly constructed materials, will radiate much of the in-duct noise to the surrounding spaces. While this can produce significant in-duct attenuation, it also can be the source for annoying radiated noise problems in the areas through which the duct passes.

In this chapter, we will first examine the primary noise source in HVAC systems: the air handler or fan. After that, we will discuss the methods of natural attenuation in duct systems, followed by a discussion of airflow generated noise. Finally, there is a presentation of noise radiation into and out of duct systems.

8.1 Fan Noise

The most reliable source of fan noise data is the manufacturer of the fan. A fan manufacturer should be able to provide test results from a laboratory testing their fan to Air-Movement and Control Association (AMCA) Standard 300*, Reverberant Room Method for Sound Testing of Fa*ns. AMCA Standard 300 specifies test setup requirements and calculation methods for rating the noise output of fans. AMCA is a trade organization currently comprised of fan, damper, and silencer manufacturers. One of AMCA's goals is to standardize the test methods and requirements for member manufacturers so that all members are employing the same guidelines in rating their product.

Note that a fan manufacturer can not possibly test all of the various combinations of fan size and flow rates, so AMCA has produced AMCA Standard 301, *Methods for Calculating Fan Sound Ratings from Laboratory Test Data*. This standard allows a fan manufacturer to predict sound power levels of a fan at various speeds and for fans of a different size, but geometrically proportional. Therefore, it is important to make sure the test data is for a product identical to that being considered. It is possible that the data for a fan you are considering has had its sound power levels estimated using AMCA 301. Avoid assumptions if possible. Paragraph 5.1, *Setup Categories*, of AMCA 300 allows various test configurations to acquire the appropriate data.

When using manufacturer's information, it is also important to note whether the data is for total or ducted sound power levels. Total sound power includes contributions from the inlet and the discharge ports, as well as noise radiated from the motor, drive train, and equipment casings. We are concerned with the noise propagated into the duct system, and the total sound power levels will probably be at least 3 *dB* higher than these ducted levels.

Like that of stand-alone fans, sound power levels for packaged air-conditioning equipment also should be obtained from the manufacturer. The manufacturer's data should be from laboratory tests done in accordance with Air-Conditioning and Refrigeration Institute (ARI) Standard 260. ARI is a manufacturer's trade organization including manufacturers of central air-conditioning and commercial refrigeration equipment. ARI Standard 260 uses AMCA Standard 300 as the primary method of obtaining sound power levels, and incorporates some items specific to packaged units. ARI 260 also allows the estimation of sound power levels for untested units.

8.2 Natural Attenuation

Even with no specific provisions for airborne noise control, such as lined duct or silencers, duct systems provide natural attenuation of noise via several mechanisms: duct wall losses, elbow reflections, sound power splits, and terminal end reflections. Together, these natural attenuations can provide significant broad-band noise reduction and may actually eliminate the need for expensive attenuators.

8.2.1 Duct Wall Losses

Whenever sound waves or any pressure fluctuations travel through a confined space such as a duct system, some component of the pressure will be transmitted to the surrounding surface. This will cause the surface to vibrate slightly and thereby dissipate a fraction of the energy from the incident pressure wave. In the case of air flowing through a duct, the energy transmitted to the surface is a function of the shape and size of the duct and the frequency of the duct-borne sound.

These losses are not the same as direct radiation of sound from inside the duct to the surrounding spaces, which is discussed in **Section 8.4.1**. The natural attenuation mechanism of duct wall loss assumes that the duct walls are massive enough to contain most of the duct-borne noise and that the energy transfer is accomplished by transforming incident sound waves into duct surface vibrations, which are then radiated to the surrounding spaces as acoustical energy at a much lower level.

Natural attenuation for round and rectangular duct is shown in **Tables 8.1** and **8.2**. These losses are expressed in terms of decibels per foot. They would appear to be significant in long lengths of duct. However, duct's natural attenuation can never reduce the noise level below the generated noise level of air inside the duct.

Table 8.1
Sound Attenuation in Straight Circular Ducts

Diameter (*inches*)	**Attenuation** (*dB/ft*)						
	Octave Band Center Frequency (*Hz*)						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
D <= 7	0.03	0.03	0.05	0.05	0.10	0.10	0.10
7 < D <= 15	0.03	0.03	0.03	0.05	0.07	0.07	0.07
15 < D <= 30	0.02	0.02	0.02	0.03	0.05	0.05	0.05
30 < D <= 60	0.01	0.01	0.01	0.02	0.02	0.02	0.02

Table 8.2
Sound Attenuation in Unlined Rectangular
Sheet Metal Ducts

Duct Size *(in.x in.)*	**Perimeter/ Area** *(ft/ft^2)*	**Attenuation** *(dB/ft)*			
		Octave Band Center Frequency *(Hz)*			
		63	**125**	**250**	**>250**
6x6	8.0	0.30	0.20	0.10	0.10
12x12	4.0	0.35	0.20	0.10	0.06
12x24	3.0	0.40	0.20	0.10	0.05
24x24	2.0	0.25	0.20	0.10	0.03
48x48	1.0	0.15	0.10	0.07	0.02
72x72	0.7	0.10	0.10	0.05	0.02

The attenuation values shown in Table 8.2 apply only to rectangular sheet metal ducts that have gages selected according to SMACNA duct construction standards.
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Natural attenuation for flat oval ducts has not been investigated thoroughly, but McGill AirFlow has sponsored enough testing to warrant the following rule of thumb: use the natural attenuation for a round duct having a diameter equal to the flat oval duct's minor axis. For example, to estimate the natural attenuation experienced by a 12x36 flat oval duct, use the attenuation values for a 12 inch diameter duct.

8.2.2 Elbow Reflections

When sound waves traveling in a duct encounter a hard metal elbow, a portion of the sound wave is reflected back in the direction of propagation. **Tables 8.3**, **8.4** and **8.5** provide attenuation values for elbows as a function of elbow diameter (minor axis for flat oval) and frequency. See **Figure 8.1** for a description of each elbow type.

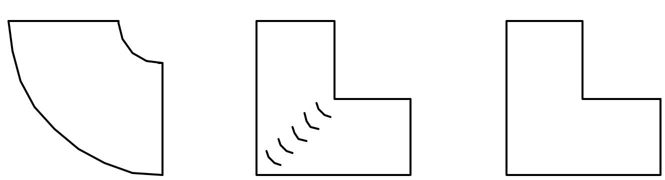

Figure 8.1
Rectangular Elbows
Radiused, mitered with vanes, mitered without vanes (from left to right)

Elbows with bend angles less than 90 have sound power level reductions proportional to the actual bend angle divided by 90. For example, a 45 degree elbow will have approximately one half (45/90) the attenuation of a 90 degree elbow.

Table 8.3
Insertion Loss of Radiused Rectangular Elbows

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** *(dB)*
fw < 1.9	0
1.9 <= fw < 3.8	1
3.8 <= fw < 7.5	2
fw > 7.5	3

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Table 8.4
Insertion Loss of Unlined and Lined Rectangular Mitered Elbows with Turning Vanes

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** *(dB)*	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 <= fw < 3.8	1	1
3.8 <= fw < 7.5	4	4
7.5 <= fw < 15	6	7
fw > 15	4	7

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Table 8.5
Insertion Loss of Unlined and Lined Rectangular Mitered Elbows Without Turning Vanes

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** (*dB*)	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 <= fw < 3.8	1	1
3.8 <= fw < 7.5	5	6
7.5 <= fw < 15	8	11
15 <= fw < 30	4	10
fw > 30	3	10

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8.2.3 Sound Power Splits

Perhaps the most significant natural attenuation mechanism is sound power splits. In the same way that air will split at a divided-flow fitting, the airborne sound power energy in watts (not sound power level in *dB*) will also be divided. This energy division will be proportional to the cross-sectional area of the straight-through (downstream) flow path of interest, divided by the total of the cross-sectional areas of all downstream flow paths at a particular junction.

For example, if a 12-*inch* common duct has a 5-*inch* branch and an 11-*inch* straight-through, then the sound energy will split as the following ratios:

$$Branch = \frac{A_b}{A_b + A_s} = \frac{\frac{p5^2}{4}}{\frac{p5^2}{4} + \frac{p11^2}{4}} = 0.17$$

$$Main = \frac{A_s}{A_b + A_s} = \frac{\frac{p11^2}{4}}{\frac{p5^2}{4} + \frac{p11^2}{4}} = 0.83$$

To convert this energy split to a sound power level reduction in decibels, use **Equations 8.1** and **8.2**:

$$\Delta Lw_{c-b} = 10 \log [A_b/(A_b + A_s)] \qquad \textbf{Equation 8.1}$$

$$\Delta Lw_{c-s} = 10 \log [A_s/(A_b + A_s)] \qquad \textbf{Equation 8.2}$$

where:

ΔLw_{c-b} = Sound power level reduction, common (upstream) to reference branch (*dB*)

ΔLw_{c-s} = Sound power level reduction, common (upstream) to straight-through (downstream) (*dB*)

A_b = Branch cross-sectional area (ft^2)

A_s = Straight-through cross-sectional area (downstream and nonreference branches) (ft^2)

For the above example, the sound power level reduction for the branch path would be 10 log 0.17 (or 8 *dB*). The straight-through path would have a sound power level reduction of 10 log 0.83 (or 1 *dB*). Sound level reductions due to power splits apply to all frequencies. **Table 8.6** provides a quick reference of sound power level reductions for various area ratios.

Table 8.6
Duct Branch Sound Power Reduction

$A_b / (A_b + A_s)$	ΔLw	$A_b / (A_b + A_s)$	ΔLw
1.00	0	0.100	10
0.80	1	0.080	11
0.63	2	0.063	12
0.50	3	0.050	13
0.40	4	0.040	14
0.32	5	0.032	15
0.25	6	0.025	16
0.20	7	0.020	17
0.16	8	0.016	18
0.12	9	0.012	19

8.2.4 End Reflections

When there is a significant change of area at the termination of a duct run, some of the low-frequency acoustical energy is reflected back into the duct, in the direction of propagation, due to the change in acoustical impedance of the air stream. That is, the multitude of air molecules in a much larger space (compared to a duct) cannot transfer energy as quickly as it is being delivered by a duct. This situation occurs when a open-ended duct discharges air directly into a large (large compared to the duct) room.

The end reflection effect is reduced or eliminated when a diffuser, register, or other terminal device is placed at the duct opening. When flexible duct is used as a final run to a terminal, the end reflection effect is essentially negated. Given current design practices, this mechanism will not provide significant attenuation for many systems. However, where low frequency noise is anticipated to be a problem, special designs which make use of the end reflection phenomenon can provide a cost-effective solution.

Tables 8.7 and **8.8** provide expected attenuation for end reflection as a function of frequency and duct diameter. These values assume the discharge is a open-ended duct (no diffuser), and that there are at least 3 to 5 diameters of straight rigid duct upstream of the discharge. If the duct is rectangular, use **Equation 8.3** to calculate the diameter to use in the tables.

$$D = \sqrt{4A/p}$$ **Equation 8.3**

Though some low aspect diffusers agree reasonably using **Equation 8.3** and **Tables 8.7** and **8.8**, it is recommended that when diffusers are placed at the duct termination, 6 dB be subtracted from the values shown. After the adjustment, any resulting negative values should be adjusted to zero.

Table 8.7
Duct End Reflection Loss -- Duct Terminated in Free Space

Diameter (*inches*)	**End Reflection Loss** (*dB*) — **Octave Band Center Frequency** (*Hz*)					
	63	**125**	**250**	**500**	**1000**	**2000**
6	20	14	9	5	2	1
8	18	12	7	3	1	0
10	16	11	6	2	1	0
12	14	9	5	2	1	0
16	12	7	3	1	0	0
20	10	6	2	1	0	0
24	9	5	2	0	0	0
28	8	4	1	0	0	0
32	7	3	1	0	0	0
36	6	3	1	0	0	0
48	5	2	1	0	0	0
72	3	1	0	0	0	0

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Table 8.8
Duct End Reflection Loss -- Duct Terminated Flush with Wall

Diameter (*inches*)	**End Reflection Loss** (*dB*) **Octave Band Center Frequency** (*Hz*)				
	63	**125**	**250**	**500**	**1000**
6	18	13	8	4	1
8	16	11	6	2	1
10	14	9	5	2	1
12	13	8	4	1	0
16	10	6	2	1	0
20	9	5	2	1	0
24	8	4	1	0	0
28	7	3	1	0	0
32	6	2	1	0	0
36	5	2	1	0	0
48	4	1	0	0	0
72	2	1	0	0	0

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8.3 Airflow-Generated Noise

Noise is generated by air flowing over duct surfaces. Although this noise may not always be audible, especially in areas where the fan noise is substantial, there are several conditions that will increase the levels of airflow-generated noise. The following situations should be avoided to minimize generated noise: (1) high velocities (>2,000 *fpm* for rectangular duct, >3,000 *fpm* for round and flat oval – except initial velocities as given in **Table 2.1**), (2) airflow turbulence, (3) obstructions in the airstream (tie rods, extractors, etc.), and (4) abrupt-turn fittings, especially those without turning vanes.

Duct and all types of fittings can create airflow-generated noise. **Appendix A.9.10** provides a summary of the generated noise properties of duct and concludes that it is directly proportional to both air velocity and duct diameter. In most systems, where both velocity and diameter have small magnitudes at discharge locations, the levels of duct self-noise will not contribute to the overall in-duct sound power level.

The reference in Appendix **A.9.10** contains a procedure for estimating the noise generation of several types of fittings. If the airflow-generated noise of a fitting is at least 10 *dB* below the residual section's sound power level, it will not contribute to the overall sound power level and can be ignored.

It is advisable to calculate the airflow-generated noise levels for all sections, regardless of their location relative to the fan. These levels should be compared to the residual fan sound power levels determined from an acoustical analysis. If the levels are within 10 *dB*, they will contribute to the overall noise level and should be added to the residual section's sound power levels, using simplified decibel addition as shown in **Table 7.3**. Sample Problem 8-1 shows the proper procedures.

Generally, as long as velocities are kept at reasonable levels and care is taken in the selection of optimum fittings, the airflow-generated noise of normal fitting components can be ignored. As we shall see later, this is not always the case for duct silencers.

Sample Problem 8-1

---

*For the system shown in Figure 8.1, calculate the sound power levels at Terminals A, B, C, D, and E, using the fan noise levels from Section 1001 shown in **Table 8.9**. Take into account natural attenuation and fitting generated noise.*

Answer: **Tables 8.9** to **8.12** show the resultant natural attenuation for each section up to the diffuser. **Tables 8.9** to **8.12** also account for the airflow-generated noise levels determined in accordance with **Appendix A.9.2**. The resulting sound power levels for each outlet are determined in the final step (row) of each table.

Fan sound power levels (Lw) are given in **Table 8.9** and are used for section 1001, which is located immediately downstream of the fan. There is no power split attenuation for section 1001; however, this section does have natural attenuation in the duct and elbow.

The attenuation of each section is subtracted from the residual sound power level of the previous (upstream) section, then the generated noise level is added using the method shown in **Table 7.3**. In this way, the sound power can be determined at any location in the system. Note that the residual sound power entering section 1003 is the attenuated level from section 1001. Also, the end reflection correction values for each terminal is 6 *dB* less than those found in **Table 8.8**, due to the diffusers.

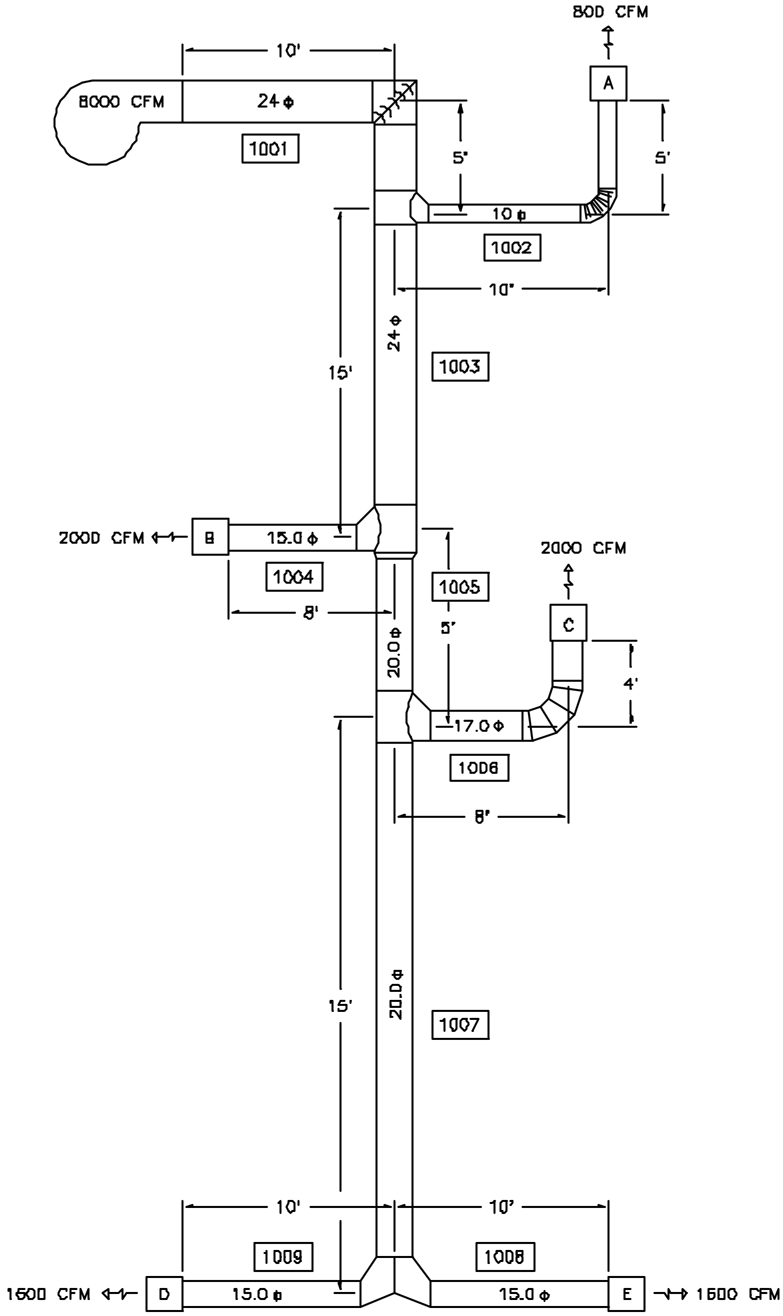

Figure 8.2
Sample Problem 8-1

Table 8.9
Single-Wall Natural Attenuation Acoustical Analysis, Outlet A

Outlet A	**Octave Band/Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1001								
1. Fan *Lw* values	89	89	90	83	78	72	68	63
2. Duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
3. Mitered 90E elbow with Vanes, 24 in. diameter	0	1	2	3	3	3	3	3
Remaining	89	88	88	80	74	68	64	59
4. Duct *GNL* at 2546 *fpm*	67	62	51	47	46	43	35	35
5. Elbow *GNL* at 2546 *fpm*	63	64	63	59	53	43	30	13
Resultant (Section 1001)	89	88	88	80	74	68	64	59
Section 1002								
6. LoLoss™ tee, branch 10 *inches* (sound power split)	8	8	8	8	8	8	8	8
Remaining	81	80	80	72	66	60	56	51
7. LoLoss™ tee, Branch *GNL*	40	39	37	35	32	28	23	18
Remaining	81	80	80	72	66	60	56	51
8. Duct, 10 *inches* *L*=15 *ft*	0	0	0	1	1	1	1	1
9. Pleated 1.5 CLR, 90E elbow, 10 *inches* diameter	0	0	1	2	3	3	3	3
Remaining	81	80	79	69	62	56	52	47
10. Duct *GNL* at 1467 *fpm*	35	30	28	27	26	24	20	20
11. Elbow *GNL* at 1467 *fpm*	0	0	0	0	0	0	0	0
Resultant 2 (Section 1002) Outlet A, *Lw* (*dB*)	81	80	79	69	62	56	52	47

Table 8.10
Single-Wall Natural Attenuation Acoustical Analysis, Outlet B

Outlet B	**Octave Band Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1003 (resultant of 1001)								
1. Entering Sound Power	89	88	88	80	74	68	64	59
2. Duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
Mitered 90E Elbow w/Vanes 24 *inch*-diameter	0	1	2	3	3	3	3	3
3. LoLoss™ Tee, Main, 24 *inches* (sound power split)	1	1	1	1	1	1	1	1
Remaining	88	86	85	76	69	63	59	54
4. LoLoss™ Tee Main *GNL*	48	39	37	35	32	28	23	18
5. Duct *GNL* at 2292 *fpm*	64	59	49	45	44	41	32	32
Resultant 3 (Section 1003)	88	86	85	76	69	63	59	54
Section 1004								
1 Duct, 15 *inches*, *L*=8 *ft*	0	0	0	0	1	1	1	1
2 LoLoss™ Tee Branch, 15 *inches* (sound power split)	4	4	4	4	4	4	4	4
Remaining	84	82	81	72	64	58	54	49
3 LoLoss™ Tee, Branch *GNL*	40	39	37	34	30	25	20	14
4 Duct *GNL* at 1629 *fpm*	43	39	37	33	35	33	23	23
Resultant 4 (Section 1004) Outlet B, *Lw* (*dB*)	84	82	81	72	64	58	54	49

Table 8.11
Single-Wall Natural Attenuation Acoustical Analysis, Outlet C

Outlet C	Octave Band Center Frequency (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1005								
1. Entering Sound Power resultant 3 (Section 1003)	88	86	85	76	69	63	59	54
2. Duct, 20 *inches*, *L* = 5 *ft*	0	0	0	0	0	0	0	0
3. LoLoss™ Tee, Main (sound power split)	2	2	2	2	2	2	2	2
Remaining	86	84	83	74	67	61	57	52
4. LoLoss™ Tee, Main *GNL*	44	43	41	38	34	29	24	18
5. Duct *GNL* at 2384 *fpm*	58	55	50	45	48	46	33	33
Resultant 5 (Section 1005)	86	84	83	74	67	61	57	52
Section 1006								
1. Duct, 17 *inches*, *L* = 12 *ft*	0	0	0	0	1	1	1	1
2. 5-gore 90E elbow 1.5 CLR	0	1	2	3	3	3	3	3
3. LoLoss™ Tee, Branch 17 *inches* (sound power split)	4	4	4	4	4	4	4	4
Remaining	82	79	77	67	59	53	49	44
4. LoLoss™ Tee, Branch GNL	42	40	38	35	31	27	21	15
5. Duct *GNL* at 1269 *fpm*	34	31	30	26	29	28	18	18
6. Elbow *GNL* at 1269 *fpm*	0	0	0	0	0	0	0	0
Resultant 6 (Section 1006) Outlet C, *Lw* (*dB*)	82	79	77	67	59	53	49	44

Table 8.12 Single-Wall Natural Attenuation Acoustical Analysis, Outlet D/E								
Outlet D/E	**Octave Band Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1007								
1. Entering Sound Power, Resultant 5 (Section 1005)	86	84	83	74	67	61	57	52
2. Duct 20 *inches*, *L* = 15 *ft*	0	0	0	0	1	1	1	1
3. LoLoss™ Tee, Main (sound power split)	2	2	2	2	2	2	2	2
Remaining	84	82	81	72	64	58	54	49
4. LoLoss™ Tee, Main *GNL*	45	43	41	38	34	30	24	18
5. Duct *GNL* at 1467 *fpm*	41	39	35	31	33	31	21	21
Resultant 7 (Section 1007)	84	82	81	72	64	58	54	49
Sections 1008 and 1009								
1. Duct 15 *inches*, *L* = 10 *ft*	0	0	0	1	1	1	1	1
2. Vee fitting branches, 15 *inches*	3	3	3	3	3	3	3	3
Remaining	81	79	78	68	60	54	50	45
3. Vee Fitting Branches, *GNL*	56	52	46	40	33	25	16	6
4. Duct *GNL* at 1304 *fpm*	33	31	29	27	28	26	18	18
Resultant 7 (Sections 1008 and 1009) Outlet D/E, *Lw* (*dB*)	81	79	78	68	60	54	50	45

As you can see, the natural attenuation in a duct system can be significant, and it is due to the natural attenuation of the duct, elbows, power splits (branch fittings), and end reflections. Sometimes, a potential noise problem can be alleviated by re-routing the duct.

Chapter 10 will take the analysis of **Sample Problem 8.1** one step further by comparing the natural attenuated noise level to the desired noise level criteria for each outlet to determine the amount of additional attenuation required via lined duct and fittings and silencers.

8.4 Radiated Duct Noise

8.4.1 Break-Out Noise

In addition to the fan noise and airflow-generated noise propagated within a duct system, another concern is the noise that radiates from HVAC duct to the surrounding spaces. This is often referred to as break-out noise, and it can be a critical design parameter whenever duct passes through or over an acoustically sensitive area. A radiated noise problem is likely to exist if the local in-duct sound power level at any frequency, minus the duct transmission loss, exceeds or is within 3 to 5 *dB* of the noise criteria (*NC*) level of the critical space. Noise criteria will be discussed in a subsequent section. Use **Tables 8.13** to **8.16** to determine the transmission loss of ducts when noise is from inside the duct and radiating outward.

Table 8.13
Break-Out Transmission Loss of Single-Wall Rectangular Duct

Duct Size *(in x in)*	**Gage**	**Transmission Loss** *(dB)*							
		Octave Band Center Frequency *(Hz)*							
		63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
12 x 12	24	21	24	27	30	33	36	41	45
12 x 24	24	19	22	25	28	31	35	41	45
12 x 48	22	19	22	25	28	31	37	43	45
24 x 24	22	20	23	26	29	32	37	43	45
24 x 48	20	20	23	26	29	31	39	45	45
48 x 48	18	21	24	27	30	35	41	45	45
48 x 96	18	19	22	25	29	35	41	45	45

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Table 8.14
Break-Out Transmission Loss of Double-Wall Rectangular Duct[1,2,3]

Duct Size *(in x in)*	**Gage**	**Transmission Loss** *(dB)*							
		Octave Band Center Frequency *(Hz)*							
		63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
24 x 24	22	20	23	26	29	35	42	44	45

1.Data based on McGill AirFlow Corporation testing and ASHRAE tables.
2.Double-wall duct is Rectangular-k27® with 1-1/2 inch insulation sandwiched between a 24 gage perforated inner liner and a 22 gage outer solid shell (total compressed thickness is about 1-3/8 inches); perforated inner liner is 23 percent open area.
3. data from 5 ft length duct

Table 8.15
Break-Out Transmission Loss of Single-Wall Round Duct

Diameter *(inches)*	Length *(feet)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000
Long Seam Ducts									
8	15	26	>45	(53)	55	52	44	35	34
14	15	24	>50	60	54	36	34	31	25
22	15	22	>47	53	37	33	33	27	25
32	15	22	(51)	46	26	26	24	22	38
Spiral Wound Ducts									
8	10	26	>48	>64	>75	72	56	56	46
14	10	26	>43	>53	55	33	34	35	25
26	10	24	>45	50	26	26	25	22	36
26	10	16	>48	53	36	32	32	28	41
32	10	22	>43	42	28	25	26	24	40

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Table 8.16
Break-Out Transmission Loss of Single-Wall Flat Oval Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000	8000
6 x 12	24	31	34	37	40	43	--	--	--
6 x 24	24	24	27	30	33	36	--	--	--
12 x 24	24	28	31	34	37	--	--	--	--
12 x 48	22	23	26	29	32	--	--	--	--
24 x 48	22	27	30	33	--	--	--	--	--
24 x 96	20	22	25	28	--	--	--	--	--
48 x 96	18	28	31	--	--	--	--	--	--

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Note that in **Tables 8.13** and **8.16**, the data are for duct lengths of 20 feet, but the values may be used for the cross-section shown regardless of length. In **Table 8.15**, if the transmission loss listed is preceded by the ">" sign, the actual transmission loss may be higher than shown. This is due to a limitation in the laboratory testing facilities that acquired the data. Data in parenthesis has a greater uncertainty than usual. The references in **Appendix A.9.2** and **A.9.10** give additional information for estimating break-out noise.

8.4.2 Break-In Noise

Just as noise from inside the duct transmits outward, ambient noise can be transmitted into a duct. This is known as break-in noise. When the in-duct noise is 10 dB or more than the break-in noise, the break-in noise can be ignored. However, at locations where fan noise and aerodynamically generated noise are minimal, significant levels of break-in noise can propagate inside the duct and radiate to surrounding spaces.

The ability of ducts to resist break-in noise is quantified as break-in transmission loss. Break-in transmission loss data is located in **Tables 8.17** to **8.19**.

Note that in **Tables 8.17** and **8.19**, the data are for duct lengths of 20 feet, but the values may be used for the cross-section shown regardless of length. In **Table 8.18**, if the transmission loss listed is preceded by the ">" sign, the actual transmission loss may be higher than shown. This is due to a limitation in the laboratory testing facilities that acquired the data. Data in parenthesis has a greater uncertainty than usual. The references in **Appendix A.9.2** and **A.9.10** give additional information for estimating break-in noise.

Table 8.17
Break-In Transmission Loss of Single-Wall Rectangular Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* — Octave Band Center Frequency *(Hz)*							
		63	125	250	500	1000	2000	4000	8000
12 x 12	24	16	16	16	25	30	33	38	42
12 x 24	24	15	15	17	25	28	32	38	42
12 x 48	22	14	14	22	25	28	34	40	42
24 x 24	22	13	13	21	26	29	34	40	42
24 x 48	20	12	15	23	26	28	36	42	42
48 x 48	18	10	19	24	27	32	38	42	42
48 x 96	18	11	19	22	26	32	38	42	42

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Table 8.18
Break-in Transmission Loss of Single-Wall Round Duct

Diameter *(inches)*	Length *(feet)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000
Long Seam Ducts									
8	15	26	>17	(31)	39	42	41	32	31
14	15	24	>27	43	43	31	31	28	22
22	15	22	>28	40	30	30	30	24	22
32	15	22	(35)	36	23	23	21	19	35
Spiral Wound Ducts									
8	10	26	>20	>42	>59	>62	53	43	26
14	10	26	>20	>36	44	28	31	32	22
26	10	24	>27	38	20	23	22	19	33
26	10	16	>30	>41	30	29	29	25	38
32	10	22	>27	32	25	22	23	21	37

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Table 8.19
Break-in Transmission Loss of Single-Wall Flat Oval Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000	8000
12 x 6	24	18	18	22	31	40	--	--	--
24 x 6	24	17	17	18	30	33	--	--	--
24 x 12	24	15	16	25	34	--	--	--	--
48 x 12	22	14	14	26	29	--	--	--	--
48 x 24	22	12	21	30	--	--	--	--	--
96 x 24	20	11	22	25	--	--	--	--	--
96 x 48	18	19	28	--	--	--	--	--	--

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8.4.3 Nonmetal Ducts

A special word of caution is in order when nonmetal ducts are being considered, especially in or near acoustically sensitive areas. These products, such as fiberglass duct , duct board, or flexible duct, have negligible mass and therefore will have very little transmission loss. A substantial amount of the noise inside a duct of this type will be immediately radiated to the surrounding spaces. Although manufacturers of these products claim high in-duct attenuation, due to the absorptive wall surfaces, very few publish data on break-in or break-out transmission loss.