Topics in Acoustics

Introduction

A previously published Fan Engineering Letter (FE-300) addressed the basics of sound and sound ratings as used in the fan industry. This document points out that sound ratings are usually published as sound power ratings. Users are often interested in the fan's contribution to sound pressure in the vicinity of the fan. FE-300 supplies some simple calculations that can be used for certain circumstances to make sound pressure predictions from the fan sound ratings. It also points out that making these predictions can often be very complicated, and that this is usually referred to a sound consultant or acoustician to determine.

In this document, some of the more complicated issues of working with fan acoustics will be addressed. While additional methods of sound pressure prediction will be given, the need for consultants cannot be eliminated. These topics can be immensely complicated and the intent here is to make the user aware of the issues so that proper input can be provided to the experts.

Before delving into the various topics, a few facts will be offered to orient the reader:

1. Fan sound ratings usually tell the power exiting the fan inlet or the fan outlet or both.
2. While there is a test code for measuring the sound power radiating out through the fan housing, this value is usually calculated from other rating values.
3. In most cases, the fan motor sound is not included in the fan sound rating and the motor must be treated as a separate sound source.
4. The sound pressure measured at any location can be dominated by any of the following sources:
   a. Fan inlet sound
   b. Fan outlet sound
   c. Housing radiated sound
   d. Motor noise
   e. V-Belt drive noise
   f. Abnormal mechanical noises caused by bearings and vibration
   g. Background or external noise sources including duct connections
5. Reducing any one noise source will not necessarily have any significant impact on the pressure at a location.
   a. It is best to reduce the component that contributes the highest value
   b. A fan inlet silencer does not reduce the outlet or radiated sound
6. The data used in predicting casing radiated sound is often anecdotal and subject to very high error.
7. Most calculations in sound are frequency specific and each octave band must be calculated separately.
   a. Usually the last step in a sound calculation is combining the values for each octave band, as outlined in FE-300.

There will be three topics offered in this discussion. The first is "Duct End Reflection", the second is "Casing Radiated Noise" and the third is "Passive Fan Silencers".

Duct End Reflection

At the exit end of many wind type musical instruments is an expanding horn. Without the horn, the output of the instrument would be dramatically reduced in amplitude. When sound traveling in a duct meets an abrupt change in area or terminates abruptly, part of the sound is reflected. For an abrupt duct termination, the amount of reflection is frequency dependent. Low frequencies are reflected more than high frequencies. There is a frequency where this effect becomes insignificant for it and all higher frequencies.

There are two types of end reflection that are recognized in the literature. One is for unflanged and the other is for a duct with an infinite flange. Of course, in the real world we often have ducts with small flanges. There are also calculations for a variety of horn shapes, but these are not used in air handling systems since we want to have the sound stay in the duct and not escape into a room. The AMCA 300 sound test code addresses both the unflanged and infinite flanged methods while some of the ASHRAE literature only covers the unflanged methods.

For ducted fans, sound ratings usually reflect the sound power released into the duct. When using reverberant room testing, the reflected sound at the end of a test duct is added back by a calculation. There is also an in-duct test method that will measure the sound in the duct directly and this calculation is not required to establish the rating. Note that for fans that are not ducted, such as plenum fans and prop fans, this correction is not relevant.

For small ducts the end reflection in the first octave band (63 Hz center frequency) can be very large. Corrections over 20dB may have to be made. Some have suggested that these large corrections are inappropriate since the sound could escape through the duct walls and actually be measured during the test. Thus adding a larger number overly inflates the sound rating. The current test code makes no attempt to determine if this is a valid criticism.

Table 1 on page 2 shows the end reflection correction values to be subtracted from sound ratings established for ducted fans to predict the sound power that "escapes" the duct. These values are for unflanged ducts.

Table 1. End Reflection Correction Values for Open Ducts in an Infinite Space

Center Frequency	Duct Diameter (inches)						
	4	8	12	18	24	36	48
63	24.7	18.7	15.3	12.0	9.7	6.8	4.9
125	18.8	13.0	9.8	6.8	5.0	2.8	1.5
250	13.0	7.6	5.0	2.8	1.5	0.5	0.2
500	7.6	3.4	1.5	0.5	0.2	0.0	0.0
1000	3.4	0.7	0.2	0.0	0.0	0.0	0.0
2000	0.7	0.0	0.0	0.0	0.0	0.0	0.0
4000	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Casing Radiated Noise

When fans are ducted both on the inlet and outlet, the area near the fan housing may be dominated by casing radiated noise. In concept, we have the total sound inside of the fan housing, which is reduced somewhat by passing through the housing. The remaining sound then radiates out into the free space surrounding the fan where it can be sensed. There are a number of traps that one can fall into when making predictions of these sound values:

1. The total sound in the fan housing is not the inlet sound or the outlet sound.
    a. One estimate is the logarithmic sum of the inlet and outlet ratings.
    b. When both inlet and outlet ratings are not available, add 3 to the values given.
2. Sound can radiate from the housing, attached ductwork and flex connectors. It can also leak through the hole in the housing that the shaft passes through. The highest of these will dominate in determining the final sound value,
3. Sound from other sources such a motors, drives and bearings may dominate over the casing radiated noise. These put a limit on the amount of reduction that is possible by reducing the casing radiated values.
4. Calculated casing radiated sound is normally a gross simplification of a very complex phenomenon. Expect relatively large errors in the calculated values.

For a bare fan housing, it is often assumed that the sound transmission loss obeys the mass law for random incidence. The loss can be calculated for common construction materials using only the weight per square foot of area. Table 2 gives the values from this calculation. Table 3 shows the weight per area for

Table 2. Sound Transmission Loss Values for Random Incidence Sound and Frequencies at Less than Half the Fundamental Natural Frequency of the Panel

Pounds per Ft^2	Center Frequency							
	63	125	250	500	1000	2000	4000	8000
0.5	4	5	9	13	18	23	28	34
1	5	9	13	18	23	28	34	39
2	9	13	18	23	28	34	39	45
3	11	16	21	26	31	37	42	48
5	15	19	25	30	35	41	46	52
7	17	22	27	33	38	43	49	55
10	20	25	30	35	41	46	52	58
15	23	28	33	39	44	50	55	61
20	25	30	35	41	46	52	58	63

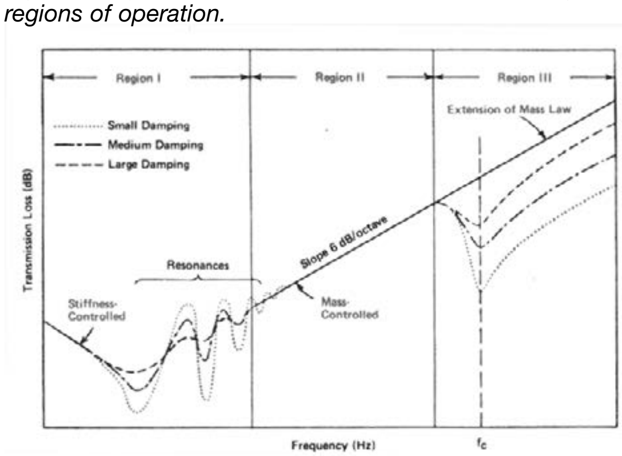
Table 3. Weight (in pounds) per square foot for steel and aluminum

Material	Lb./Sq. Ft.	Material	Lb./Sq. Ft.
Steel 22 Ga.	1.22	Aluminum 0.063"	0.90
Steel 20 Ga.	1.46	Aluminum 0.080"	1.14
Steel 18 Ga.	1.95	Aluminum 0.100"	1.43
Steel 16 Ga.	2.44	Aluminum 0.125"	1.78
Steel 14 Ga.	3.04	Aluminum 0.190"	2.71
Steel 12 Ga.	4.26	Aluminum 0.250"	3.56
Steel 10 Ga.	5.48	Aluminum 0.375"	5.35
Steel 7 Ga.	7.31		
Steel 0.25 in. thick	10.19		

a number of materials. The transmission loss values are subtracted from the ratings to estimate the levels radiating through the housing.

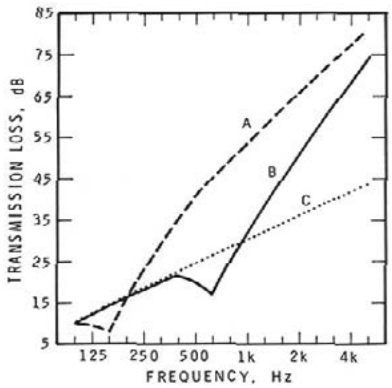
In real structures, the mass law is only valid within a range of frequencies. In typical fan housings, there are resonant frequencies in panels of the housing that affect low frequency results. In this range, panel stiffness and damping control the transmission loss. At a resonant frequency, the sound can pass through the housing at almost full amplitude, where damping is low. Figure 1 shows that there are three frequency ranges of operation. Region I is the resonant dominated region. Region II is the mass law dominated region. Region II has a slope of about 6 dB per octave. Many partitions exhibit a dip in the transmission loss curve at higher frequencies and this is where Region III begins. The dip is caused by acoustic coupling, which is beyond the scope of this discussion. The frequency fc is called the "critical" or "coincident" frequency in different sources. At higher frequencies, the slope is greater than 6 dB per octave and is reported as 9-10 dB/octave.

Figure 1. Typical transmission loss curves showing three regions of operation.



Where higher housing transmission loss is required, the housing can be layered with insulation on the outside along with an outer skin. While there are methods to calculate the effectiveness of layered housings, most estimates are based on testing of samples. Figure 2 on page 3 shows that changing the gap between layers can dramatically affect the response. It also shows that the three regions of performance are relative and Region III dominates the Curve A performance through most of the important frequencies.

Figure 2. Transmission loss for two layers of 0.02" thick steel walls separated by a sound absorbing material of various thicknesses.



Curve A has a gap of 4" and has a natural frequency at 135 Hz.

Curve B has a gap of 0.2" and has a natural frequency at 630 Hz.

Curve C is the mass law prediction for 0.04" steel plate.

Data is published for measured sound transmission loss for a variety of materials. Note that data is not available for all frequencies and published data for the same material may vary significantly due to testing variations. Table 3 below shows typical results for some materials.

In summary, casing radiated noise can be estimated and controlled to manageable levels when necessary. However, use caution when using layered construction, which can have very large sound transmission losses (over 40 dB). At some point, another source of noise will become dominant and the expected noise reduction will not actually be fulfilled.

Passive Fan Silencers

A common way of blocking sound from progressing down a duct or of preventing the sound from exiting the end of a duct is to use a silencer. Certain special applications can make use of an active silencer, which uses speakers to inject sound out of phase with the existing sound and provide cancellation. Active silencers are primarily used for tonal reduction. However, by far the most common types of silencers are passive silencers. There are reactive silencers and dissipative silencers, but all passive silencers exhibit properties of both types. A reactive silencer generally uses changes of direction, changes of area and branches with chambers to provide reflections of the sound waves. They are commonly tuned to specific frequencies where they become most effective. For example, a tuned reactive silencer can be configured to be most effective in reducing the amplitude at the blade pass frequency.

To many in the air moving industry, the term “silencer” or “sound trap” refers to silencers, which contains a packing (commonly fiberglass) that dissipates the sound. The packing helps convert the sound energy to heat. Generally packing must be covered over by woven glass cloth, mylar, a thick coating or perforated plate to prevent erosion in the air stream. Often combinations of these coverings are used.

These silencers are offered in a variety of rectangular and round configurations. They do add some flow resistance to the air stream and this must be compensated for by adding to the static pressure requirement of the fan. A common resistance is 1" wg, but the resistance may vary from about 0.05" wg. to 6.0" wg. Some designs are integrated with an expanding area évasé to both reduce the sound and increase the static pressure capability. Also, when round silencers are used on axial flow fans, a cylindrical center body can be included, which also improves silencer effectiveness. Many designs can be used to improve the sound reduction effectiveness in a given space, however in general these will increase the static pressure requirement.

While the rectangular and round dimensions of the silencer are generally matched to the fan, the length can be varied. For round silencers, the length is often given as a multiple of the diameter. A 2D silencer is two diameters long. Longer silencers perform better, and to be effective in low frequencies, very long lengths may be required. The effectiveness is better for upstream versus downstream locations. This can be expected by considering an example. At a flow velocity of 4000 FPM and using a 4 ft. long silencer a 500 Hz sound wave will go through 1.68 cycles moving downstream. When moving upstream, it will go through 1.89 cycles. So, the 500 Hz wave “sees” the silencer as about 12% longer in the upstream condition. The flow can also focus the sound wave, which creates additional changes in its performance.

Table 3. Sound Transmission Loss for Various Tested Materials

		Octave Band							
Material	**Lb/Ft³**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
FRP (Reinforced Plastic) 1/8" thick	**1.13**		15	18	25	26	29	36	
FRP (Reinforced Plastic) 1/4" thick	**2.08**		19	22	28	31	32	25	
FRP (Reinforced Plastic) 1/2" thick	**4.20**		21	27	29	34	27	36	
Steel Sheet 3/16" thick	**7.64**		29	31	30	32	33	31	
Steel Sheet 16 Ga	**2.38**		18	22	28	31	35	41	
Steel Sheet 20 Ga	**1.50**		16	19	25	27	32	39	
Steel Sheet 24 Ga	**1.02**		13	16	23	24	29	36	
Lead Vinyl	**1.25**		17	19	28	30	34	39	
Lead Vinyl + 2.5" FG Insul + Lead Vinyl			25	34	38	43	47	58	
16 Ga Stl + 2.5" FG Insul + 16 Ga Stl			26	33	36	38	41	51	
16 Ga Stl + 4.5" Insul + 4" ribbed .04 Alum			25	43	48	56	63	61	
3/16" Stl + 4" FG Insul + 16 Ga Stl			42	44	43	50	54	60	

There are two main environmental effects to be considered. Over time, a silencer may lose effectiveness due to the accumulation of dirt. This tends to plug the holes in the perforated plate. Cleaning may restore it to its full effectiveness. Another effect is for high temperature performance. At high temperatures, there is a shift in the effectiveness of the silencer to higher frequencies. Thus, it is more effective at higher frequencies and less effective at low frequencies.

The air velocity blowing through the silencer will also generate sound. So the process in determining the final sound is to subtract the attenuation (insertion loss) of the silencer from the fan rating, then to combine (logarithmically) with the self noise. This puts a lower limit on the amount of reduction that can be achieved with a silencer. Table 4 shows typical silencer performance ratings. Note that these ratings include the pressure drop, insertion loss and self noise for various face velocities. The negative face velocities refer to ratings with flow going into the silencer and the positive values are for flow out of the silencer.

Where space permits, the ducts can be lined with a sound absorbing material such as fiberglass. In this case there may be rating data published for both the sound transmission loss through the duct and liner, as well as ratings for the sound reduction of the sound passing down the duct. The latter is usually published as dB per unit of length for each octave band. This can be very effective in reducing noise if a substantial length of duct is lined.

Table 4. Typical Catalog Data for Silencers

Length Inches *(mm)*	**Pressure Drop** *(in. wg.)*	**Face Velocity** *fpm (m/s)*	**Insertion Loss (dB)** *Octave Band Center Frequency (Hz)*								**Generated Noise L_w (dB; 10^{-12} Watts)** *Octave Band Center Frequency (Hz)*							
			1 63	**2 125**	**3 250**	**4 500**	**5 1K**	**6 2K**	**7 4K**	**8 8K**	**1 63**	**2 125**	**3 250**	**4 500**	**5 1K**	**6 2K**	**7 4K**	**8 8K**
36 (914)	**0.80 (199)**	**-2000 (-10.2)**	6	13	18	28	37	35	20	16	54	54	55	59	60	63	65	61
	0.20 (50)	**-1000 (-5.1)**	4	9	16	27	36	35	20	17	41	40	39	42	51	55	45	35
	0.20 (50)	**+1000 (+5.1)**	4	8	15	26	35	35	22	16	41	35	30	31	32	32	25	21
	0.80 (199)	**+2000 (+10.2)**	3	6	13	23	31	35	23	18	59	58	52	48	48	51	51	51
60 (1,524)	**0.96 (238)**	**-2000 (-10.2)**	11	19	25	40	53	49	29	18	54	54	55	59	60	63	65	61
	0.24 (50)	**-1000 (-5.1)**	11	18	23	40	50	49	34	21	41	40	39	42	51	55	45	35
	0.24 (50)	**+1000 (+5.1)**	8	14	22	39	50	50	37	26	41	35	30	31	32	32	25	21
	0.96 (238)	**+2000 (+10.2)**	7	13	20	35	48	49	36	25	59	58	52	48	48	51	51	51
84 (2,134)	**1.48 (368)**	**-2000 (-10.2)**	10	20	38	52	55	56	39	25	54	54	55	59	60	63	65	61
	0.37 (92)	**-1000 (-5.1)**	9	20	37	49	55	55	40	29	41	40	39	42	51	55	45	35
	0.37 (92)	**+1000 (+5.1)**	7	18	34	49	55	55	45	30	41	35	30	31	32	32	25	21
	1.48 (368)	**+2000 (+10.2)**	6	15	32	47	53	53	47	34	59	58	52	48	48	51	51	51
120 (3,048)	**1.30 (323)**	**-2000 (-10.2)**	11	31	42	57	59	55	48	30	54	54	55	59	60	63	65	61
	0.48 (119)	**-1000 (-5.1)**	11	30	41	53	55	55	50	33	41	40	39	42	51	55	45	35
	0.48 (119)	**+1000 (+5.1)**	9	26	41	56	55	57	50	36	41	35	30	31	32	32	25	21
	1.30 (323)	**+2000 (+10.2)**	8	22	39	56	53	59	51	41	59	58	52	48	48	51	51	51