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INDUSTRIAL NOISE SERIES

Part VII

ABSORPTIVE SILENCER

DESIGN

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ABSORPTIVE SILENCER DESIGN

1. INTRODUCTION

The basics of absorptive silencer design and the fundamental approach in design of parallel baffle silencers is presented. The reduction of sound energy using absorptive materials is achieved by transferring the acoustical pressure (wave motion) into material motion. This mechanical motion is converted into heat (energy loss) by material damping and friction. The more effective the sound wave penetrates the material the more effective the attenuation. Each baffle assembly consists of "compartments" and the basic theory is each compartment is *locally reacting* where the acoustic sound wave "pumps in and out" though the material as well as through the perforated facing sheet or pack material retainer. The perforation pattern adds damping and frictional losses to the aero-acoustic wave oscillating through the holes; the smaller the holes and more perforations, the more attenuation. The packing consists of absorptive material that is principally fibrous materials or open cell foams that allow the wave energy to penetrate, induce material motion, and be attenuated.

The wavelength of sound is a major factor in sound reduction as well since the absorptive material only starts to become effective when its thickness is at least one-tenth the wavelength so obviously, very low frequencies are not well attenuated. The effective depth of a baffle is one-half its thickness as acoustic energy is propagating into the baffle from both sides. Baffle geometry is described in the following figure and the descriptors vary between resources. Silencers were developed based on this geometry and the corresponding open area ratio (OAR) of the baffles in the silencer duct. The open area is the area available for gas or air flow through the silencer.

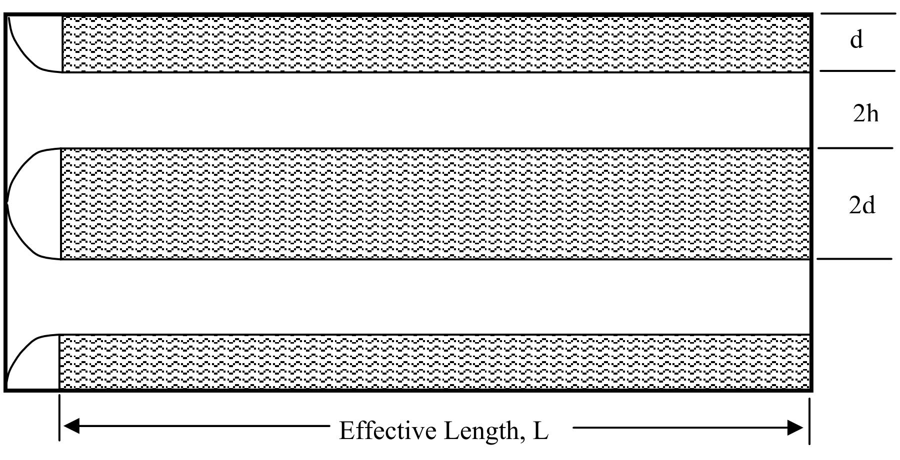

FIGURE 1. BAFFLE GEOMETRY

Silencer - Baffle OAR% = 100 x 2h/(2h + 2d) = 100 x h/(h + d)

Full air gap = 2h, Baffle thickness = 2d

Note that half gaps and half baffle thickness are h and d respectively and performance is based on these parameters permitting the use of half baffles and half gaps in silencer applications. Each configuration in the following series of figures has the identical acoustical performance and OAR to that of Figure 1 assuming the dimensions d, h and L are identical.

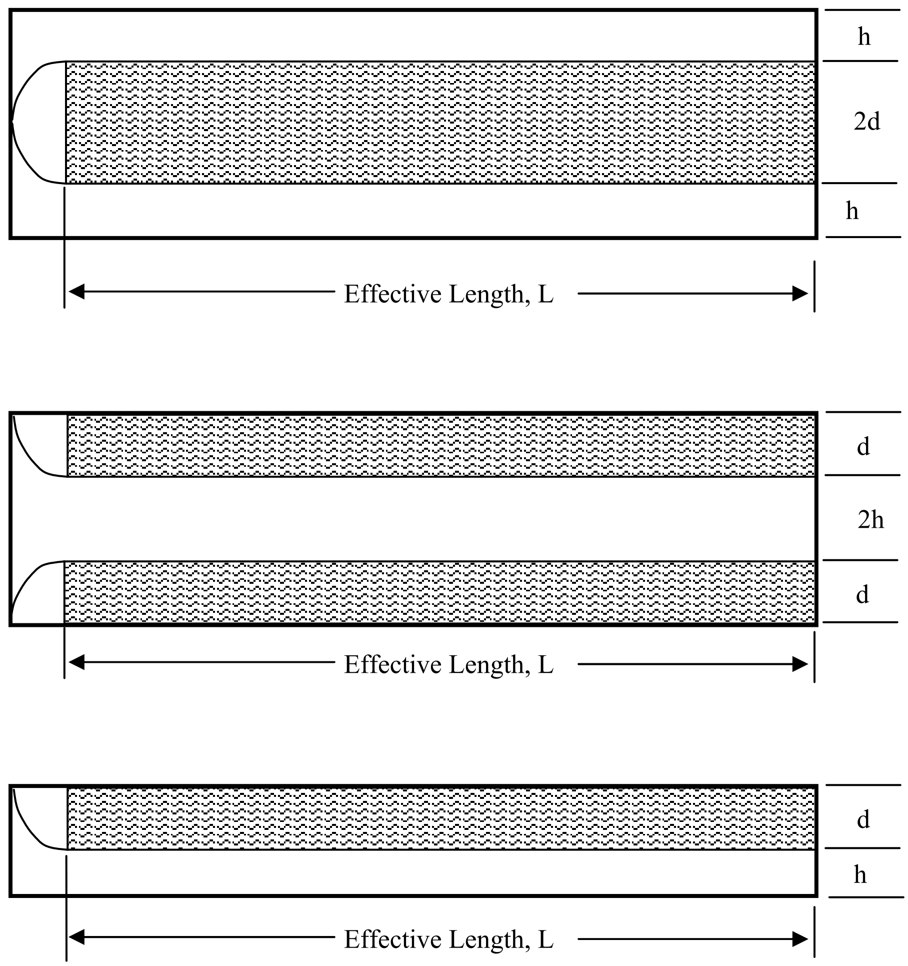

FIGURE 2. BAFFLE GEOMETRIES HAVING IDENTICAL ACOUSTICAL PERFORMANCE

The number of baffles and air gaps only affects flow area as does the height or span of the baffle; more baffles, more flow area, lower air gap velocity, lower pressure loss. The number of parallel baffles does not affect the acoustical performance whatsoever except in the case of self flow noise; a function of the gas or air velocity between the baffles and face area of the baffles. The face area is the total cross sectional area of the silencer occupied by baffles.

There are two basic parallel baffle arrangements. The familiar flat, rectangular parallel baffles and the annular ring baffle used in cylindrical silencers. In general, the method used for calculating flat panel baffles can be used for annular silencers but there is some performance departure about the 2,000-3,000 Hz region that may cause a slight drop in performance and in other geometries the performance improves. The reason for using flat parallel baffle approach is cylindrical databases are not widely published and those that are only provide data in a limited range. A safety margin will cover these issues.

Key parameters that are used for baffle design are the dimensions cited above, the airflow resistivity of the pack material (rayls/m), the airflow resistance (rayls) of any thin layer retainer materials. More sophisticated analyses include the thickness and areal density of the perforated face sheet, perforation diameter and oar, the density of the pack material, and its material properties. These latter parameters are frequently missing from standard "lookup curves."

2. SILENCER PERFORMANCE

The challenge in baffle design is determining the attenuation in a grazing flow environment; that is, the sound travels parallel to the surface (grazing) thus interaction of the sound wave with the baffle is not optimal at all. The direction of flow with respect to the direction of the sound wave also affects performance in that for an identical baffle and sound energy, the performance will be different for a positive traveling wave versus a negative traveling wave. The following figure illustrates the flow regimes; the curves represent the sound energy.

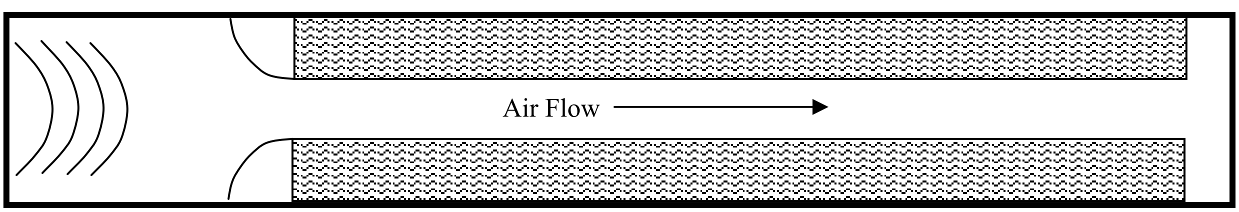

FIGURE 3A. POSITIVE FLOW CASE – SOUND AND FLOW ARE TRAVELING DOWNSTREAM; THIS IS COMMONLY KNOWN AS EXHAUST OR SUPPLY FLOW.

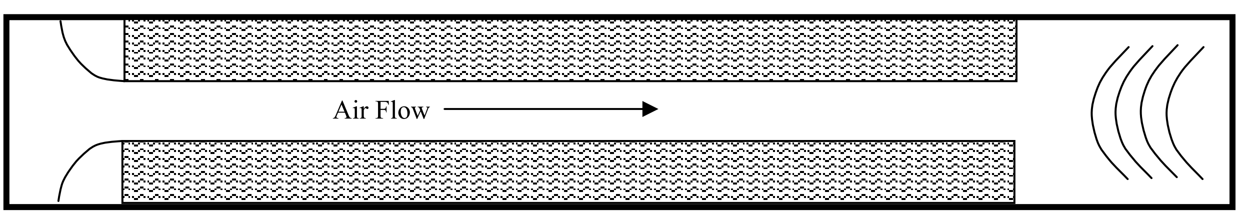

FIGURE 3B. NEGATIVE FLOW CASE – SOUND TRAVELS UPSTREAM AND FLOW TRAVELS DOWNSTREAM; THIS IS COMMONLY KNOWN AS INLET OR RETURN FLOW.

Calculating silencer performance was not developed when ventilation equipment started being installed in buildings and the only method was to build and test silencers. Later, the HVAC[1] industry developed standardized testing protocols to measure silencer performance. The national standard is ASTM E 477, *Standard Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers*. The ISO standard is, ISO 7235 *Acoustics – Laboratory measurement procedures for ducted silencers and air-terminal units – Insertion loss, flow noise and total pressure loss*. Also, there is a standard for measuring silencer performance in-situ or for very large silencers that cannot be measured in a laboratory; it is ISO 11820 *Acoustics – Measurements on silencers in situ*. It provides guidelines for measuring both insertion loss (IL) and transmission loss (TL).

Although computational capabilities now exist, the complexity of the silencer required a significant amount of analysis that naturally had some uncertainty regarding performance thus most all HVAC silencer manufacturers just tested their products and cataloged them for easy selection. With over 50 years of testing, databases (performance curves based on OAR) have been developed. Figure 4 (Fig. 10.20 from Beranek and Ver)[2] shows normalized attenuation performance for a 66% open area silencer based on four acoustical materials under no-flow (static) conditions. The four materials have normalized flow resistances of 1, 2, 5 and 10.

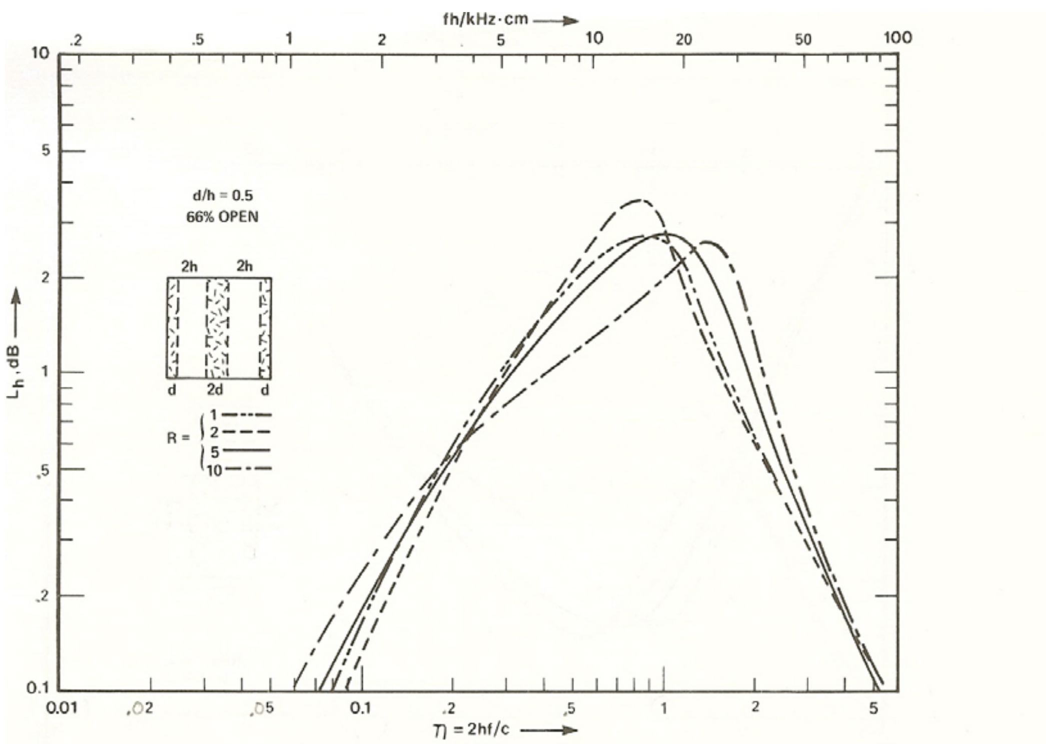

FIGURE 4. NORMALIZED ATTENUATION VERSUS FREQUENCY FOR PARALLEL BAFFLES

[1] Heating, Ventilation and Air Conditioning (HVAC)

[2] A.G. Galaitsis and I.L. Ver, "Passive Silencers and Lined Ducts," Ch 10 in *Noise and Vibration Control Engineering Principles and Applications*, edited by Beranek, L.L. & Ver, I.L. (John Wiley & Sons Inc. NY 1992)

The abscissa is the full air gap (2h) divided by the wavelength (c/f). The ordinate shows the attenuation rate, L_h in decibels per half gap (h); the baffle attenuation is L_h multiplied by (L/h). Some resources show the attenuation rate per full gap (2h) so the ordinate values would be half that shown; so be careful when comparing databases. And also note, terminology and symbols may vary between resources.

Parallel baffle performance is contingent upon the open area of the baffle arrangement (OAR) and a unique performance curve for each open area is required; thus, a family of curves covering 25%, 33%, 40%, 50%, and 66% are common in any design and analysis library and each family of curves will include performances based on the pack material's flow resistivity. If a baffle arrangement has some other OAR then the result is obtained by interpolation.

What is not shown in these normalized graphs is the perforated face sheet of the baffle. The open area of the perforated face sheet also affects performance; particularly above 1,000 Hz. Most all HVAC silencers use 40% open area thus it is seldom an issue in terms of application but for industrial applications where low percentage perforated sheeting may be used it does become important. In general, for typical engine applications 23% is acceptable but in the case of turbine exhausts that have high frequency content, then 32-35% should be considered. For turbine inlets, high speed fans, and any devices that have significant tones above 1,000 Hz, use 40% as a minimum. The open area ratio (OAR) in percent of perforated sheeting may be calculated by,

Square perforation pattern: $$P\% = 78.5(d/b)^2 \tag{1}$$

Hex perforation pattern: $$P\% = 90.7(d/b)^2 \tag{2}$$

Where d is the diameter of the hole and b is the center to center spacing of the holes.

3. Flow Resistance and Resistivity

Flow resistance or resistivity is a measure of the material's resistance to penetration by a sound wave; the lower the number the easier it is for penetration and better attenuation. Very low flow resistivity is good for high frequencies and very high flow resistivity is good for low frequencies so there are trade-offs. ASTM C 522, *Standard Test Method for Airflow Resistance of Acoustical Materials* is the method used for performing the measurements. Very thin materials such as fill retainers (screens, cloth and needle-mats) are measured for their total resistance in rayls and thicker packing materials are measured for their flow resistivity in rayls/meter. The rayl in this application is an aerodynamic measure of the pressure differential across the material at a known volumetric air velocity. Although not a true acoustical measurement, it is used for rating the packing material. Testing occurs in a duct with a known cross area so the parameters for rayls/m are composed of the following units, $Pa{\cdot}s/m^2$ but Pa is N/m^2 so you may encounter, $N{\cdot}s/m^4$.

In Figure 4 there are several curves showing flow resistivity. Note that the peak attenuation occurs where η is near unity when the air gap (2h) is equal to a wavelength. The curves shown on these types of figures are the normalized flow resistance of the material and dependent upon the thickness d. By selecting a particular or desired performance curve (R in Figure 4) and knowing the baffle thickness (d) the flow resistivity (R_l) of the needed material is then determined by employing the following equation,

$$R = R_l\, d/\rho c \qquad \text{dimensionless} \tag{3}$$

where ρc is the acoustical impedance of the air, 415 rayls. The units of R_l are rayls/meter or rayls/inch. It is very important to note that the flow resistance will change with the operating temperature (T) because of the change in the gas or air viscosity. It may be scaled by,

$$R(T) = R_0(T/T_0)^{1.2} \qquad \text{dimensionless} \tag{4}$$

$$R_0 = R_l d/\rho_0 c_0 \qquad \text{dimensionless} \tag{5}$$

T is the absolute temperature in Kelvin (273° + °C) and Equation (5) represents the measured laboratory flow resistance at ambient or laboratory temperature, T_0.

4. Flow Velocity

The flow through baffles also affects the attenuation and longevity of the packing:

- The flow velocity in any application should never exceed Mach 0.3 (0.3 x speed of sound) because the gas or air will begin to compress changing its properties.

- The velocity between the baffles produces self noise that can limit the attenuation of a silencer thus the velocity should not exceed Mach 0.1 as a general rule.

- The flow affects the sound energy interaction with the baffle by either making the acoustic interaction either shorter or longer based on the velocity and direction of noise relative to the flow. Downstream and upstream flow cases are shown in Figure 3. Flow is either positive or negative and is important in analyzing performance.

- If calculating baffle performance, by using published curves (at no flow), the attenuation rate is adjusted by (Bies & Hansen, Eq. (9.79)[3],

$$D_M = D_0[1 - 1.5M + M^2] \tag{6}$$

Where, D_0 is the attenuation rate (without flow) and $-0.3 < M < 0.3$ where M is the Mach number of the velocity between the baffles. Be sure to use the correct sign (+/-) of the Mach number depending on the flow case (Figure 3). Some

[3] Bies and Hansen, *Engineering Noise Control, Theory and Practice*, 2nd edition, © David A. Bies and Colin H. Hanson, 1996

resources use different approaches for adjusting for self noise but this is most conservative and easy to use.

5. Baffle Attenuation Example

An example calculation of baffle performance is given here, presenting only six middle bands. Assume a baffle configuration is 8 inches thick (2d) separated by 6 inches air gap (2h) that is 4 feet long (43% OAR). The air temperature is 68° F with a positive velocity of 100 fps through the baffles. For convenience of using Figure 5, the packing material's flow resistivity, R_l is 20,000 rayls/m; this has a normalized value close to "5" for d = 4 inches (0.1016 m). The results are given in the following table as explained below:

Table 1 – Example Calculation of Baffle Attenuation

Freq =	125	250	500	1000	2000	4000
η =	0.11	0.22	0.44	0.89	1.77	3.55
L_h =	0.2	0.6	1.5	2.8	1.4	0.2
D_M =	.175	.525	1.31	2.45	1.22	.175
4 ft =	2.8	8.4	21.0	39.0	19.5	2.8

Step 1 – Calculate η (2hf/c) for each band or frequency for the air gap of 6 inches.
Step 2 – Go to Figure 4 and at each value of η across the abscissa go up and intersect the R=5 curve (solid line).
Step 3 – Then traverse over to the ordinate and record the L_h value.
Step 4 – Calculate the gap velocity and its Mach number (note sign for flow case), M = 100/1126 = +0.088
Step 5 – Calculate D_M using Equation (6); here, $D_0 = L_h$
The flow effect reduces the attenuation about 12%, $D_M = L_h \times 0.8747$
Step 6 – The number of diameters of the gas path is 48/3 = 16;
Step 7 – Multiply D_M by 16 to obtain the attenuation in decibels for the 4 feet of baffle.

Using this method one can calculate the required baffle length to achieve the attenuation needed or determine the flow resistivity of the packing material to be used. Now, HVAC catalog data would show higher attenuation rates because measured data is used versus calculated. Had another material been used (R), one would interpolate between the curves. For a different temperature, calculate R(T) using Equation (4) and adjust c (speed of sound accordingly).

One could include many more frequencies in the calculations, which is easy with a spreadsheet with imbedded algorithms. Final baffle attenuation is further adjusted to account for baffle entrance and exit losses which are usually small. Keep in mind that virtually all text book and similar databases are generally conservative.

6. Baffle – Silencer Modeling

Fortunately, the need to use graphical databases to perform lengthy calculation to size a silencer is no longer required. Just imagine the design challenges for all the different flow conditions, temperatures, configurations, applications, and materials. There are software packages available for sizing baffles and silencers that have the capability to rapidly develop silencers for a wide range of configurations and materials. Some specialized programs include the density and dynamic properties of the pack material in determining performance. These programs permit the rapid design and optimization of silencers.

7. Self Noise and Dynamic Insertion Loss

When air or gas flows through a baffle or silencer, the interaction of the fluid flow, particularly at the exit, causes flow disturbances resulting in flow noise. As the velocity increases the flow noise increases and there are no simple algorithms to calculate this affect as the geometry of the baffles and silencers are highly variable. Most published algorithms are fairly conservative and only a few include temperature affects. The baffle design spreadsheet used at Universal employs octave band algorithms for inlet and exhaust cases developed from self noise levels published in silencer catalogs and includes the temperature of the exhaust gas. An algorithm that incorporates temperature (other than ambient) for inlet silencers has not been developed yet. The interested reader can review this in detail by going to the acoustical spreadsheet user guide.

When low noise requirements are specified the self noise may limit performance. Self noise is generally a sound power level adjustment to the resulting attenuated sound power level (PWL). An example calculation is presented in the following table. The final emitted PWL2 is the logarithmic summation of the previous two rows, PWL1 and the Self Noise PWL.

Table 2 – Example of Self Noise on Performance, decibels (dB)

OBCF:	31.5	63	125	250	500	1k	2k	4k	8k	A-wt
Source PWL	126	124	130	131	123	124	123	122	122	131
Silencer TL	3	6	14	27	41	59	53	34	13	
PWL1	123	118	116	104	82	65	70	88	109	109
Self Noise	**91**	**83**	**74**	**71**	**69**	**67**	**71**	**69**	**63**	**76**
PWL2	123	118	116	104	82	69	74	88	109	109
Delta =						+4	+4			
DIL =	3	6	14	27	41	55	49	34	13	22

In this example the velocity between the baffles is 164 ft/s (9855 fpm, 50 m/s) and causes degraded performance in the 1k and 2k octave bands by 4 dB (compare PWL1 to PWL2). As can be seen, this reduces the effective insertion loss in those bands resulting in what is termed the dynamic insertion loss (DIL) since it includes the affect of flow noise.

Flow noise generally affects the middle bands and for very low noise requirements can be problematic resulting in large silencers to obtain a low velocity and with the added benefit of low pressure drop. However, the self noise is a power level that is also a function of the face area of the silencer and as the silencer area increases to reduce velocity, there may be an increase in self noise PWL because of the area increase. PWLs are typically an absolute value and do not change but in the case of flow noise, the increase in area also causes more self noise energy to be present. This becomes very problematic in very low noise applications or if the power level of the source is very low as the self noise PWL can then be more than the source PWL.

Very high or very low source sound power levels and the distribution of the energy across the bands will also impact the overall insertion loss performance. The following tables illustrate an example case; the only difference between the two tables is the sound power level (PWL) of the source listed in the second row. Table 3b lists sound power levels typical of large diesel engine exhausts and note that the overall sound power level is seven decibels less.

Table 3a – Example of Source PWL on Overall Performance

OBCF:	16	31.5	63	125	250	500	1k	2k	4k	8k	A-wt
Source PWL	112	116	122	124	124	137	142	146	141	134	149
Silencer TL	1	1	2	4	7	10	16	29	17	6	
PWL1	111	115	120	120	117	127	126	117	124	128	
Self Noise	96	96	88	79	76	74	72	76	74	68	
PWL2	112	115	120	120	117	127	126	117	124	128	132
DIL =	0	1	2	4	7	10	16	29	17	6	**18**

Table 3b – Example of Source PWL on Overall Performance

OBCF:	16	31.5	63	125	250	500	1k	2k	4k	8k	A-wt
Source PWL	136	154	156	151	138	136	135	133	133	131	142
Silencer TL	1	1	2	4	7	10	16	29	17	6	
PWL1	135	153	154	147	131	126	119	104	116	125	
Self Noise	96	96	88	79	76	74	72	76	74	68	
PWL2	135	153	154	147	131	126	119	104	116	125	134
DIL =	1	1	2	4	7	10	16	29	17	6	**8**

Note the dramatic drop in overall A-weighted performance, from a DIL from 18 dB to 8 dB even though the octave band level (16 Hz to 8k Hz) DIL is identical and the overall power level is seven decibels less, 142 dB versus 149 dB. In these cases, self noise was not a factor in performance.

This example demonstrates the importance of having the sound power level of the source in order to assess the silencer performance.

One element not addressed herein is pressure drop and is largely a function of the gap velocity. The calculation of acoustical performance must be balanced with pressure loss and the available area for the silencing system. In critical applications for high acoustical performance, low pressure loss, and limited size, the general rule is “pick any two.”

8. Insertion Loss

Most times it is either insertion loss (IL) or dynamic insertion loss (DIL) specified as the performance requirement. It is critical to realize that the actual insertion loss is dependent upon the system duct geometry, the source's sound power level, temperature of the fluid medium, and for dynamic insertion loss, the flow rate. Flow through a silencer generates self noise that may degrade performance and to meet a DIL means all the cited information above is required.

One important item to know is insertion loss is the difference in sound pressure level at a set point with and without the silencer in place as illustrated below. This is not done as it is obvious that the expense of such a test would be prohibitive except for smaller silencers where a spool piece (empty duct section) could be easily inserted in place of the silencer.

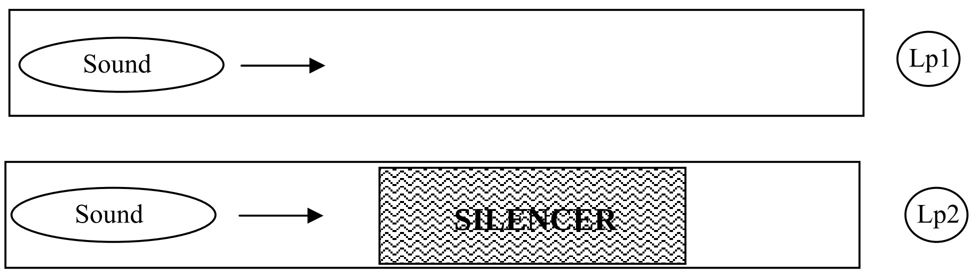

Figure 2 – Insertion Loss Measurement

$$\text{DIL or IL} = L_{P1} - L_{P2} \quad \text{dB} \qquad (7)$$

Obviously, to verify performance would be expensive by having the equipment and manpower to swap out the silencer for an empty duct section and make measurements.

9. Transmission Loss

When specifying silencer performance, a more convenient method is to specify the transmission loss (TL) of the silencer. The TL is the reduction of sound power level and can be measured upstream and downstream of the silencer and the ducting system is not a factor. Refer to ISO 11820 *Acoustics – Measurements on silencers in situ*; it provides guidelines for measuring both insertion loss (IL) and transmission loss (TL). Universal calculates the TL (as shown in Tables 2 and 3) and includes self noise, thus effectively showing the dynamic transmission loss (DTL) reported as DIL as shown in the cited tables. The TL is used because it is very rare that a full system description is provided when a silencer is requested and frequently system arrangements change during project development. In the case of changing ducting system arrangements, a revised calculation of the insertion loss would be necessary and most projects do not perform this as it causes delays and added costs if the silencer needs to be modified.

The following exercise demonstrates that TL is a very good approximation of IL. We start with the classical approach of measuring or defining the sound levels (L_P) with and without the silencer in place associated with insertion loss measurement. The sound power emitted from the duct is noted as L_{W1} and L_{W2} for the two measurement conditions.

$$L_{P1} = L_{W1} + C \quad \text{dB (with the silencer)} \tag{8a}$$

$$L_{P2} = L_{W2} + C \quad \text{dB (without silencer)} \tag{8b}$$

$$IL = [L_{P2} - L_{P1}] = [(L_{W2} + C) - (L_{W1} + C)] = L_{W2} - L_{W1} \quad \text{dB} \tag{8c}$$

$$TL \approx L_{W2} - L_{W1} \quad \text{dB} \tag{8d}$$

Assuming C is constant for both boundary conditions, with and without the silencer, it is shown that the insertion loss (IL) is approximately equivalent to the TL. Now there are some liberties here as it assumes L_{W2} is the same value upstream of the silencer (in place). But lacking detailed design data, this is adequate to show that TL works well for estimating the IL of a system. The uncertainty is the very low frequencies which are generally not important for meeting an overall sound level reduction. When having to meet very low frequency TL or IL requirements, safety margin (increased silencing) is added to account for any uncertainty.