Micro-perforated Panels for Duct Silencing

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Micro-perforated panels have been used for duct silencing for several years and are gaining popularity. This paper presents equations for predicting the sound attenuation of micro-perforated panel silencers. It also presents guidelines for the silencer geometric parameter selection to improve silencer performance. The guidelines are taken from both silencer tests and theoretical analyses.

Primary subject classification: 34, Secondary subject classification: 37

1. INTRODUCTION

A micro-perforated panel is a sheet metal panel with many small diameter holes (usually less than one millimeter) distributed over its surface. Past research work has shown that when the holes are reduced to such a small size, the panel acoustic resistance increases tremendously. As a result the ratio between its acoustic resistance and its acoustic mass increases tremendously. With a high acoustic resistance and high resistance-to-mass ratio, the perforated panel itself forms an efficient sound-absorbing construction without the use of any porous material [1 - 4].

In recent years, micro-perforated panels have been used to build silencers. Such silencers are similar to conventional silencers except that there is only air behind the perforated panel as compared to conventional silencers which have porous material behind the panel. Without porous materials, such silencers are free of the problems usually associated with porous materials, such as bacteria contamination and small particle discharge. They are widely used in hospitals, food industries, pharmaceutical industries and micro-electronics industries. As the application of micro-perforated panel silencers widens, the interest in predicting and improving the silencer's performance increases accordingly.

Although much work has been done on developing such silencers and several patents have been granted [5], few papers have been published on micro-perforated panel silencer design [6]. In the early 80's P. T. Soderman published his research results on duct silencing using perforated panels [7]. However, his silencers are constructed with panels with large holes (1.6 to 3.2 millimeters diameter). Such large-hole silencers have low and narrow-banded insertion losses, generate tones in ducts carrying high speed air flow, and, therefore, are not widely used. On the other hand, research work on micro-perforated panel absorption mechanisms has been carried out for many years and many papers have been published [1 - 3, 8 - 10]. However, those papers focus on the panel application as a sound absorber in rooms, the results cannot be readily used in silencer design.

In this paper, equations are presented to predict the plane-wave sound attenuation in micro-perforated panel silencers. The predicted sound attenuations are converted into silencer dynamic insertion losses and compared with laboratory test results. The comparison shows that the predicted sound attenuations agree with the test results well enough to predict silencer performance changes due to geometric parameter changes, but not well enough to generate catalogue data. In Chapter 4 of the paper, the effects of the silencer geometric parameters on the silencer performance are discussed. Guidelines, obtained from tests and theoretical analyses, are offered for selecting the key parameters to achieve optimum performance in micro-perforated panel silencers.

2. PREDICTION OF SOUND ATTENUATION OF MICRO-PERFORATED PANEL SILENCERS

A typical rectangular duct silencer is shown in Figure 1. It consists of a straight sheet metal duct with parallel sound absorbing panels inserted inside. The width of a single unit is defined as the silencer unit size. The duct areas between the panels are called the open path areas of the silencer. The ratio of the open path areas to the total cross-section of the silencer is called the open path ratio. The open path ratio is usually between 20% and 60%.

The insertion loss of a silencer is generally defined as the difference, in decibels, between two sound pressure levels which are measured at the same point in a space before and after the silencer is inserted between the measurement point and the noise source. The insertion loss of a silencer includes three major parts: plane-wave sound attenuation, area correction factor and random-incident wave correction factor.

The plane-wave sound attenuation is the sound reduction caused by the sound absorption of the silencer panels. Its value depends on the acoustical properties of the panels and the geometric parameters of the silencer. The major goal of this paper is to explore the effects of the panel properties and the silencer geometric parameters on sound attenuation, although we will also add the area correction factor and the random-incident wave correction factor to the sound attenuation to obtain the silencer insertion loss, and dynamic insertion loss (the insertion loss measured with air flow), so that it can be compared with the measurement results.

The sound absorbing panels of a silencer are usually supported on the back by solid partitions. These partitions divide the back space into small cells and prevent sound propagation behind the panels. In this case, the sound absorbing panel is called "locally reacting". The impedance of the panel is independent of the distribution of sound along the duct and can be regarded as a quantity known a priori [11]. In this paper, we restrict our discussions to the locally reacting case only. The limitations and advantages of this restriction will be discussed after Equation (17).

For ducts with locally reacting panels, the task of predicting sound attenuation along the duct is to determine the sound pressure amplitude in the duct subject to the boundary condition of a known normal impedance. As shown in Figure 2, let x be the direction along the duct; y be the direction normal to the panels; the center line of the silencer be at y=0; and the sound absorbing panel be at y=h (where h is half the distance between the panels). According to Ingard, we can express the complex amplitude of the sound pressure as follows [12].

$$p(x,y,\omega) = A\cos(k_y y)e^{jk_x x}, \tag{1}$$

where A is a constant; $\omega = 2\pi f$; f is frequency in Hz; $j = \sqrt{-1}$; k_x and k_y are the complex wave numbers in the x and y directions, respectively. Let the real part of k_x be β and the imaginary part be γ,

$$k_x = \beta + j\gamma. \tag{2}$$

In fact, the x-dependence of the magnitude of the complex amplitude of the sound pressure is given by

$$\frac{|p(0)|}{|p(x)|} = e^{\gamma x}$$

and the corresponding attenuation in decibels (dB) from x=0 to x=L (for a silencer of length L) is a function of γ

$$D = 20\cdot\log_{10}\left|\frac{p(0)}{p(L)}\right| = 20\cdot\log_{10}(e)\cdot\gamma L \approx 8.72\cdot\gamma L. \tag{3}$$

When there is air flow in the silencer, the attenuation is [13]:

$$\mathrm{D_m} = \frac{\mathrm{D}}{1+\mathrm{m}}. \tag{4}$$

D is the attenuation without air flow; D_m is the attenuation with air flow; and m is the Mach number - the ratio of the average air flow velocity between the panels to the speed of sound in the air. When the air flows from the sound source to the receiver, m is positive. Otherwise, m is negative.

The value of γ is determined by k_y, which is further determined by the normal acoustic impedance of the sound absorbing panels.

The relationship between k_x and k_y is

$$k_x = \sqrt{k^2 - k_y^2}, \tag{5}$$

where k=/c, is the wave number, in meter^{-1}; c is the speed of sound in the air, in meters per second.

The amplitude of the y-component of the vibration velocity of the air particles at the surface of the sound absorbing panel is

$$u_y = \frac{1}{-j\omega\rho}\frac{\partial p}{\partial y} = \frac{A}{-j\omega\rho}k_y\sin(k_y y)e^{jk_x x}, \tag{6}$$

where is the air density in $\mathrm{kg/m^3}$. The boundary condition at y=h, i.e. the surface of the sound absorbing panel is

$$\frac{u_y}{p} = \frac{1}{Z}, \tag{7}$$

where Z is the normal specific acoustic impedance of the absorptive panel, defined as the complex ratio of the sound pressure to the particle vibration velocity. Z is in units of N-sec./m^3;

Substituting Equations (1) and (6) into (7), we have

$$k_y h \tan(k_y h) = -jkh\rho c / Z \quad (8)$$

Having obtained k_y in terms of the known quantities on the right hand side of Equation (8), substitute k_y into Equation (5) to determine the propagation constant k_x for the silencer. The imaginary part of k_x, , is then used to yield the attenuation of the silencer from Equations (3) and (4) [12].

In general, Equations (8) and (5) have to be solved numerically. However, according to Kurze, can be approximately expressed as follows [14].

$$y = \mathrm{Im}\left(k\sqrt{1 - \left(\frac{2}{k \cdot h}\right)^2 \left\{1 + \left[\frac{1}{1 + 4Z/jkh\rho c}\right] \pm \sqrt{1 + \frac{1}{[1+4Z/jkh\rho c]^2}}\right\}}\right) \quad (9)$$

In the equation, Im( ) represents the imaginary part of the complex function in the parentheses. The proper sign in Equation (9) is the one that gives the smaller value of .

The specific acoustic impedance Z of a Helmholtz panel system, which includes a perforated panel with an air space behind it (see Figure 2), consists of a resistance term R, a mass term M and a compliance term C.

$$Z = R + j \cdot \omega M - j \cdot C \quad (10)$$

According to Maa [1 through 3], the resistance term, mass term and compliance term of a micro-perforated Helmholtz panel system can be expressed as below.

$$R = (32 \cdot \mu \rho \frac{t}{P \cdot a^2}) \cdot \left\{\sqrt{1 + \frac{x^2}{32}} + 0.177 \cdot x \cdot \frac{a}{t}\right\}, \quad (11)$$

$$M = t \cdot \frac{\rho}{P} \cdot \left\{\left[1 + \frac{1}{\sqrt{9 + \frac{x^2}{2}}}\right] + 0.85 \cdot \frac{a}{t}\right\}, \quad (12)$$

$$C = \rho \frac{c}{\tan(k \cdot d)} . \quad (13)$$

In Equations (11) through (13), $x = 10 \cdot a \cdot \sqrt{f}$; is the kinematic viscosity of air ($1.56 \cdot 10^{-5}$ meter^2 / second); t is the thickness of the panel, in meters; a is the panel perforation diameter, also in meters; P is the porosity of the panel, equal to the ratio of the perforated open area to the total area; and d is the depth of the cavity behind the panel, in meters. The other symbols have been defined previously.

For the micro-perforated panels used in silencers, the perforation diameter a is usually less than 1 millimeter. Therefore, the value of x is greatly less than unity. Equations (11) and (12) can be simplified to

$$R = (32 \cdot \mu \rho \frac{t}{P \cdot a^2}) \cdot \left[1 + 0.177 \cdot x \cdot \frac{a}{t}\right] \quad (14)$$

and

$$M = t \cdot \frac{\rho}{P} \cdot \left\{1.33 + 0.85 \cdot \frac{a}{t}\right\} . \quad (15)$$

Furthermore, in silencers, the ratio between the perforation diameter and panel thickness equals, approximately, unity. Since x is much less than unity, the second term in Equation (14) is much less than the first term. Equation (14) further simplifies to

$$R = 32 \cdot \mu \rho \frac{t}{P \cdot a^2} . \quad (16)$$

The significance of this simplification is to change the frequency-dependent panel resistance into a frequency-independent constant, whose value equals the steady flow resistance, which can be measured in a conventional flow resistance apparatus [15].
The compliance term shown in Equation (13) applies when the sound impinges normal to the panel. When the sound impinges the panel at an angle to the normal direction, the effective wave number becomes k cos() and Equation (13) becomes [16]:

$$C = \rho \frac{c}{\cos(\theta) \cdot \tan(k \cdot d \cdot \cos(\theta))} \ . \qquad (17)$$

Since we have limited our discussion to the locally reacting case only, we have defined = 0, and Equation (17) reduces back to Equation (13). When the distance between the supporting partitions behind the perforated panel is smaller than half a wavelength, it is a good approximation to regard the air backing as a locally reacting element and = 0 [13]. At high frequency, when the locally reacting assumption does not apply, the panel becomes non-locally reacting. The velocity component normal to the boundary at a certain point depends not only on the local sound pressure at that point but also on the sound pressure distribution over the entire boundary. The normal impedance of the boundary is not known a priori. It can be found only after the wave fields in the absorber and in the air duct have been determined. The non-locally reacting case is discussed in detail in Reference [17].
When air flows between two parallel panels, as in the case of a silencer, the acoustic impedance of the panel system changes. The major effect of the flow on the panel impedance is the pumping of air through the panel due to sound-flow interaction [18]. A simple expression to approximate the flow effect is [19],

$$Z_m = Z_0 \cdot (1 + m) \ , \qquad (18)$$

where Z_0 is the impedance without air flow; Z_m is the impedance with air flow and m is the Mach number, defined previously.
By substituting Equations (11) through (18) into Equations (3), (4) and (9), the sound attenuation of a micro-perforated panel silencer can be calculated.

3. COMPARISON OF PREDICTED SILENCER INSERTION LOSS TO MEASUREMENT

The insertion loss of a silencer includes three major parts. They are the plane-wave sound attenuation, area correction factor and random-incident wave correction factor. The plane-wave attenuation is the sound reduction caused by the sound absorption of the silencer panels. It can be calculated by the equations given in Chapter 2. The area correction factor is the transmission loss caused by sound reflections at the entrance and the exit of the silencer due to duct cross-section mismatch. It is a function of sound wavelength, the length of the changing section and the ratio between the cross-section areas. It can be determined using the figure given in Reference [20]. For a silencer of 25% open path ratio, the transmission loss is approximately 2 dB at both entrance and exit of the silencer. The random-incident wave correction is defined by Doelling [21]. It is the sound attenuation in addition to the plane-wave attenuation. The validity of this correction factor depends on the relative amounts of plane-wave and random-incidence energy in the duct. The correction factor rises from zero, when $\sqrt{S} / \lambda \leq 0.1$, to a maximum of 10 dB, when $\sqrt{S} / \lambda \geq 1$, where is the wavelength and S equals the duct area between two panels. Doelling has shown by testing that this correction factor is limited to 10 dB regardless of the silencer length and the frequency.
Figures 3 through 6 show the predicted one-third octave band dynamic insertion loss curves of four silencers. The curves are calculated by adding the area correction factor and random-incident wave correction factor to the plane-wave sound attenuation obtained from the equations given in Chapter 2. When the value of insertion loss is greater than 50 dB, it is assumed to be 50 dB. This assumption is based on the fact that practical silencers do not achieve attenuations above 50 dB (due to flanking and other transmission paths).
Also shown in Figures 3 through 6 are the measured octave band dynamic insertion loss curves of the same silencers. The measured curves are obtained according to ASTM Standard E477, “Measuring Acoustical and Airflow Performance of Duct Liner Material and Prefabricated Silencers”. During the measurements, the silencer is installed in a long straight duct with one end connected to a reverberant room and the other end connected to a fan and noise source. The sound pressure levels in the reverberant room are measured with the silencer installed in the long straight duct and with a replacement duct installed at the silencer location. A

replacement duct is a straight sheet metal duct with negligible sound attenuation. The difference between the two sound pressure levels is the dynamic insertion loss of the silencer.

The silencers used to obtain the data shown in Figures 3 through 6 are 1.8 meters long and 0.6 meters high. Silencer A has a unit size of 0.61 meters and an open path ratio of 25%. Silencers B, C, and D have unit sizes 0.31, 0.76, and 0.38 meters, open path ratios 25%, 33%, and 33%, respectively. The dynamic insertion loss curves are obtained by calculating or testing the silencer at 10 meters per second face velocity. Face velocity is defined as the average air flow velocity in the duct before entering the silencer.

The figures show that the predicted dynamic insertion losses agree with the tested dynamic insertion losses reasonably well. Qualitatively, the curves show the effects of parameters correctly in a sense that an increase or decrease in the calculated insertion loss and in the frequency of the peak attenuation correspond to an increase or decrease in the measured insertion loss and the frequency of the peak attenuation, respectively, although the values may not agree exactly. We can also see that the predicted insertion loss curves have higher values at the Helmholtz resonant peak and lower values beyond the peak. This is probably because we ignored some damping factors in the micro-perforated panel system.

According to Guess [8], for a perforated panel backed by an air space and a solid wall, its acoustical impedance includes four items: viscous and mass impedance, radiation impedance [22], non-linear resistance due to high sound amplitude and steady tangential airflow [18], and backing impedance. In this paper, we included in Equations (11) through (13) only the viscous and mass impedance and the backing impedance. Radiation impedance and non-linear resistance have been neglected. According to Ingard [18], the major effect of flow on the impedance of a perforated panel is the pumping of air through the panel due to sound-flow interaction. The magnitude of the non-linear resistance depends on the porosity of the panel, the magnitude of the turbulent velocity fluctuation due to a turbulent boundary layer, and the dissipation due to jet formation on the exit side of the perforation (with oscillatory acoustic motion). The negligence of the non-linear resistance term may be the reason the predicted insertion loss curves look under-damped comparing to the test results.

In the following section, we summarize the effects of silencer geometric parameters on the silencer performance and illustrate the effects by calculated sound attenuation curves. The reason for using calculated plane-wave attenuation instead of measured insertion loss is that the former shows the parameter effects more systematically and without the random measurement errors. Furthermore, it shows the parameter effects more clearly because it does not include the area correction factor and the random-incident wave correction factor.

4. EFFECTS OF SILENCER PARAMETERS

In this chapter, we discuss the effects of silencer geometric parameters on silencer performance and present guidelines for silencer parameter selection. These guidelines are obtained from laboratory tests as well as theoretical analyses of the plane-wave attenuation equations. The effects of silencer geometric parameters are illustrated by calculated plane-wave attenuation curves. It should be cautioned that these curves are given only to illustrate the effects of geometric parameters. They should not be used as quantitative guidelines in silencer design because the quantitative effects of one parameter may vary with the change of other parameters. The proper way to examine the effect of one parameter for a particular silencer is to calculate the plane-wave attenuation using the baseline silencer parameters, then change the subject parameter and re-calculate the attenuation.

As shown in Figures 3 through 6, the typical attenuation curve of a micro-perforated silencer has one major peak. Beyond the peak, the attenuation decreases. The major goal of silencer performance improvement is to widen this attenuation peak. The other goal is to control the frequency of this attenuation peak.

A. Effects of Perforation Hole Diameter

Reducing the perforation hole diameter is the most effective way to widen the attenuation peak. When the hole diameter is reduced, the acoustic resistance of the panel increases, the damping of the panel Helmholtz system increases and the attenuation peak widens. The major reason for the micro-perforated panel silencer to have significantly improved performance over the silencers tested by Soderman [7] is that the micro-perforated panel silencers have smaller holes.

The attenuation curves shown in Figure 7 illustrate the above statement. The three silencers have the same configuration except for the perforation hole diameters. Silencer A has a hole diameter of 0.635 millimeters. Silencers B and C have hole diameters of 0.508 and 0.381 millimeters, respectively. Silencer attenuation clearly increases from silencer A to B and again from B to C.

B. Effects of Panel Thickness

Increasing the thickness of the panel is another way to widen the silencer attenuation peak. However, it is not as effective as reducing the perforation hole diameter. Test results have shown that the difference in the insertion loss between silencers made of panels with the same perforation hole diameter, but a 19% difference in panel thickness, is smaller than the test error.

Equation 16 shows that the panel's acoustical resistance is inversely proportional to the second power of perforation hole diameter while proportional to the first power of panel thickness. That explains why decreasing hole diameter is more effective than increasing panel thickness in increasing the panel acoustic resistance and therefore the silencer sound attenuation.

The effect of panel thickness is further dimmed due to the so called "effective mass" of the vibrating air. When the air inside an orifice (i.e. a perforated hole) vibrates, the air entering and exiting it also vibrates. This added vibrating air effectively adds mass to the air column inside the orifice and thus makes the equivalent length of the orifice longer than its geometric length. This added effective length at each end of the orifice is approximately 0.85 times the orifice diameter [23]. For the micro-perforated panels typically used for silencers, the perforation hole diameter is approximately the same as the panel thickness. Therefore, this added length is 1.7 times the geometric length of the orifice, i.e. the thickness of the panel. As a result, doubling the panel thickness only increases the total effective thickness of the panel by 37%. Hence, although an increase in panel thickness should theoretically increase the panel system resistance and widen the silencer attenuation, its practical effect is minimal. The positive side of this phenomenon is that reducing the panel thickness does not reduce the panel acoustic resistance much either.

Due to limitations in panel perforation technology, the hole diameter is limited to approximately the same dimension as the panel thickness. Either reducing the hole diameter or increasing the panel thickness from this limit will increase panel cost dramatically. Since reducing hole diameter is a more effective way to improve silencer performance than increasing panel thickness, it is more beneficial to reduce both the panel thickness and the hole diameter.

The curves of Figure 8 show the attenuations of three silencers with the same configuration except for panel thickness. All three silencers have 0.787 millimeter diameter holes. Silencer A has a panel thickness of 0.635 millimeters. Silencer B has a thicker panel of 0.508 millimeters and C has a thinner panel of 0.381 millimeters. The figure shows that the changes in panel thickness cause little change in attenuation.

Since increasing the panel thickness increases the air mass per hole, and reduces the resonant frequency of the Helmholtz panel system, the frequency of peak attenuation is slightly lower for a silencer with a thicker panel. This trend is shown in Figure 8.

C. Effects of Depth of Cavity Behind Panel

The frequency of a silencer's peak attenuation can be controlled by the depth of the cavity behind the panel. If the panel is a purely resistive locally reacting element, the first attenuation peak occurs at a frequency for which the cavity depth equals a quarter wavelength, and the higher order peaks occur when the depth is an odd number of quarter wavelengths. However, when the panel has a mass reactive component, the peaks occur at somewhat lower frequencies. This reduction factor in frequency can be shown [13] to be, approximately, $d/(d+a/P)$, where d is the depth of the cavity, a is the panel perforation diameter, and P is panel porosity, the fractional open area of the total panel area. For example, for a silencer with a unit size of 0.61 meters and an open path ratio of 25%, depth d is 0.23 meters. The frequency with its quarter wavelength equals 0.23 meters is 375 Hz. If the panel porosity is 2.5% and the panel perforation diameter is 0.8 millimeter, the frequency reduction is about 12%. The attenuation peak frequency of the silencer should therefore be at 325 Hz. For a silencer with half this unit size, but all the other parameters the same, the cavity depth is reduced by half and the attenuation peak frequency almost doubles to 590 Hz. The attenuation curves shown in Figure 9 are those of silencers with 0.61 and 0.31 meter unit sizes. Their attenuation peak frequencies are at 325 Hz and 590 Hz, respectively.

D. Effects of Silencer Open Path Ratio

Similar to a porous material silencer, a decrease in the silencer open path ratio increases sound attenuation and therefore widens the attenuation bandwidth. Curves A, B, C and D in Figure 10 represent the attenuation of four silencers with open path ratios of 20%, 25%, 30%, and 35%, respectively. Obviously, the silencer with the smallest open path ratio, 20%, has the widest attenuation band. However, just as for porous material silencers, the smaller open path ratio, the higher pressure drops. Therefore, in many applications, decreasing silencer open path area is not an acceptable option for improving silencer performance.

For a given silencer unit size, a silencer with a smaller open path ratio has a deeper air cavity behind the panel, a lower Helmholtz panel resonant frequency and hence a lower frequency at peak attenuation. This trend is also shown in Figure 9.

E. Effects of Panel Perforation Porosity

One parameter which should be carefully chosen is the panel perforation porosity, the fractional perforation area to total panel area. Choosing a proper porosity can improve silencer attenuation without significantly increasing the panel cost, as in the cases of reducing the hole diameter or increasing the panel thickness. Neither will it increase silencer pressure drop, as does reducing silencer open path ratio. Figure 11 shows the sound attenuation of four silencers with the same configuration, except for panel porosity. With too large or too small porosity, sound attenuation decreases. The optimum porosity depends on many factors such as silencer open path ratio, silencer unit size, etc. Unfortunately, the author cannot give a general optimum porosity, but it can be determined from the equations for any particular case.

F. The Effects of Air Flow Speed

According to silencer test results, attenuation increases as the air flow speed increases. When the silencer face velocity increases from 5 meters per second (1000 fpm) to 10 meters per second (2000 fpm), the attenuation usually increases by 1 to 5 dB, depending on the unit size and the open path ratio of the silencer. Curves A, B and C in Figure 12 represent the calculated attenuation of a silencer at face velocities of 0, 5 and 10 meters per second, respectively. The calculated attenuation does not show enough increase with air flow speed. This is perhaps because the fact that the non-linear effects of air flow [18] has been neglected in our calculations.

G. Double Panel Silencer

Since the micro-perforated panel system is a single Helmholtz resonant system, we expect the silencer attenuation to have one major peak. The width of the peak is usually one to two octave bands. It is difficult to further widen the attenuation peak beyond this limit with the present panel perforation technology. One way to further widen the attenuation peak, is to use a double resonant system which consists of two micro-perforated panels in front of each other (see Figure 13). Maa has reported [3] work on architectural panel-absorbers. By adjusting the depth of the air cavities behind the panels, he is able to adjust the resonant frequencies of the double resonant system and achieve a system with a significantly widened absorbing bandwidth [3]. The same principles apply to silencers. Multi-panel silencers can be configured to have insertion loss as wide as those of porous material silencers [13].

H. Silencer Attenuation at High and Low Frequencies

As shown in the figures, silencer attenuation has one major peak. Below the peak, silencer insertion loss is mostly due to the cross-section area mismatch. Above the peak, insertion loss is mostly due to the random-incident wave correction factor. Changing the panel geometric parameters will not significantly improve silencer attenuation at low or high frequencies, unless the perforation hole diameter can be reduced beyond the present sheet metal perforation technology.

5. CONCLUSIONS

In this paper, equations are presented to predict the sound attenuation of micro-perforated panel silencers. Also, the effects of silencer geometric parameters are discussed. A few parameter selection guidelines are summarized below.

- Reducing the panel perforation diameter increases the sound attenuation bandwidth. However, the minimum achievable perforation diameter is usually paractically limited to about the panel thickness.
- Increasing panel thickness does not significantly improve silencer attenuation.
- The maximum sound attenuation occurs at a peak one or two octave bands wide. The frequency at the peak is controlled by the depth of the cavity behind the panel. When the depth is doubled, the frequency is reduced by approximately half.
- The effect of silencer open path ratio on its sound attenuation for a micro-perforated panel silencer is similar to that for a porous material silencer. Reducing the silencer open path ratio increases silencer attenuation,

but also increases the pressure drop.

- Changing the perforation porosity is an efficient way for improving silencer insertion loss. An optimum porosity can be found to achieve the best silencer performance.
- For the same silencer, attenuation increases as the air flow speed increases.
- Silencer insertion loss at low frequency is mostly due to the duct cross-section mismatch and is controlled by the silencer open path ratio. The insertion loss at high frequency is mostly due to the random-incident wave correction factor and is limited to approximately 10 dB. Changing the panel parameters does not change the low or high frequency attenuation much.

A comparison of predicted silencer insertion losses to measured insertion losses has shown that the equations presented in this paper can predict the plane-wave attenuation change caused by parameter changes. Thus, they can be used to estimate the attenuation of a new silencer slightly different from an existing silencer with known insertion loss. However, the equations are not accurate enough to generate catalogue data.

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