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The insertion loss of perforated porous plates in a duct without and with mean air flow

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Abstract

Measurements of the acoustic insertion loss of poroelastic plates with different perforation ratios, mounted transversely across a flow duct, without and with flow have been made at various locations in the duct. For the lower perforation ratio the insertion loss is found to be approximately 3 dB higher with air flow than without over the frequency range of the measurements. A parallel impedance model has been formulated to predict the effects of perforation on the insertion loss without flow. Measurements and predictions without flow have been found to be in reasonable agreement.

Keywords: Absorption; Insertion loss; Perforated plates; Mean flow

1. Introduction

Poroelastic materials may be employed for noise control in aircraft, buildings and various other engineering applications. According to Goransson [1], ‘the fundamental characteristic of porous materials treated to reduce noise is that the produced fluid flow through the poroelastic material is opposed because of the frictional force produced by the fibres or the cell walls on the fluid. This mechanism allows the energy to be absorbed from the acoustic wave and to be converted into heat.’ Viscous friction due to relative motion between solid and fluid constitutes one mechanism. If the frame of the porous material is viscoelastic then other dissipative mechanisms are possible.

Horoshenkov and Sakagami [2] have used the Helmholtz integral equation formulation to produce the solution for the acoustic field reflected from a finite, thin, poroelastic plate in a rigid baffle with simply supported edges. The acoustic properties of the porous material were taken into account by using the effective fluid assumption.

Takahashi and Tanaka [3] have presented an analytical model of sound absorption and sound transmission through a single permeable membrane. They carried out a theoretical investigation of the sound absorption of structures composed of air layers, absorptive layers and permeable membrane facings and compared predictions with the experimental data measured by using the reverberation room method. Takahashi et al. [4] have presented a new method for analyzing acoustic coupling due to flexural vibration of perforated plates and plates of poroelastic materials. The analytical model they presented is developed by introducing flow continuity at the plate surface in a spatially mean sense and air–solid interaction within the plate material. They have analyzed some acoustic problems based on a classical thin-plate theory in relation to the interactive effect of flexural vibration and plate permeability to demonstrate the method of application.

Rao and Munjal [5] have calculated the normal incidence impedance of an orifice or cluster of orifices (perforated plate), and investigated their use in the prediction of noise reduction across concentric tube resonators The effects of the thickness of the plate and the diameter of the holes on the impedance of the perforate were taken into

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account. Dickev et al. [6] have presented an experimental investigation of the linear impedance of perforated interfaces exposed to grazing fluid flow, including a description of the branch-type setup used in the study.

Peat et al. [7] have compared the results from a recent theoretical analysis of the problem with a new set of experimental results.

Kirby and Cummings [8] have presented data for the acoustic impedance of multiply perforated plates exposed to fully developed grazing gas flow in a duct, both with and without a backing layer of porous material. Cummings and Astley [9] have described a FE formulation for sound attenuation in “bar-silencers”, consisting of rectangular prisms of sound-absorbing material placed in a rectangular lattice arrangement within a rigid-walled duct. They took the uniform mean gas flow in the “airway” of the silencer into account. Cummings [10] has analyzed the transmission of complex periodic and transient acoustic signals through orifice plates at high amplitude, and in the absence of mean fluid flow. He solved the equation of motion for the air in the orifice numerically in the time domain.

Ingard [11] has studied the absorption and scattering from resonators in a free field as well as in walls. Morfey [12] has extended the theory of sound transmission and generation in hard-walled ducts to include axial and swirling mean flow. The theory presented is based on the idea of a single-frequency mode response function rather than a Green function. Wendoloski [13] has examined the acoustic behaviour of a constricted duct when a mean flow is present.

The purpose of this manuscript is to investigate the effects of inserting a clamped perforated porous plate on the uniformity of flow and the sound absorption in a duct. These effects are assessed by measuring insertion loss at different locations in the duct. A parallel impedance model is used to model the effects of the perforation. The role of perforation is to increase the permeability and to decrease the pressure drop associated with the insertion of the layer. The plate is assumed to be governed by the clamped rectangular poroelastic plate theory developed by Leclaire et al. [14] and Aygun et al. [15]. The measured insertion loss of porous plate in the absence of mean air flow is compared to predictions.

2. Theoretical analysis

2.1. Acoustical coupling for perforated porous plate

A schematic diagram of a perforated poroelastic plate in a baffle and the symbols used for modelling this arrangement are given in Fig. 1. The theoretical analysis does not include the absorption due to the open pores on the inside surfaces of the perforation holes. Such absorption can be neglected due to the relatively small size of the pores compared with the holes and the relatively small surface area involved. It is assumed that the perforated porous plate and the baffle separate two fluid half-spaces.

The pressure difference $\Delta P$ across the perforated poroelastic plate causes plate vibration in a flexural mode with velocity $v_p$. If the porous plate is vibrating under acoustic loading ($v_p \neq 0$), then the mean particle velocity $v_m$ is given by [3]

$$v_m = v_p + (v_r - v_p) \lambda$$  \hspace{1cm} (1)$$

where $v_r$ is the spatially averaged velocity in each hole, and $\lambda$, is the perforation ratio. Also, introducing $Z_0 = Z_{\text{resist}} + Z_{\text{react}} = \Delta P/v_r$, the impedance of each hole when the plate is at rest, it is possible to write [3]

$$Z_{\text{resist}}(v_r - v_p) + Z_{\text{react}}v_r = \Delta P$$  \hspace{1cm} (2)$$

The real part of the impedance, $Z_{\text{resist}}$, is related to the air-solid interaction in each hole. Rearranging Eq. (2), the relative velocity between each hole and the porous plate is given by [3]

$$v_r - v_p = \frac{\Delta P}{Z_0} - \frac{Z_{\text{react}}}{Z_0} v_p$$  \hspace{1cm} (3)$$

By inserting Eq. (3) into Eq. (1), the mean particle velocity becomes

$$v_m = v_p - \left(\frac{Z_{\text{react}} v_p + \Delta P}{Z_0}\right) \lambda$$  \hspace{1cm} (4)$$

The acoustic impedance of a perforation hole is given by [16], i.e.

$$Z_0 = \left(\frac{h}{R} + 2\right) \sqrt{2\eta_0 \omega \rho_0} + j \omega \rho_0 (0.96 S_1^{1/2} - 7.7 R S_1^{1/2} + h) + Z_c \frac{\pi R^2}{\rho_0} 0.0625$$  \hspace{1cm} (5)$$

where $R$ is the diameter of the hole, $h$ is the thickness of the plate, $\eta_0$ is the air viscosity which is equal to $1.839 \times 10^{-5}$ Pa s for standard temperature and pressure, $\rho_0$ is the air density, $\omega$ is the angular frequency, $S$ is the area of the hole, and $Z_c$ is the characteristic impedance of the free air.
The boundary condition for the acoustic velocities at the source side of the perforated porous plate are given by [2]

$$\nabla P_s = i\omega p + \frac{P_s}{Z_{in}}$$

where $Z_{in}$ is the front specific acoustic impedance of the plate, $P_s$ is the sound pressure on the source side, and $\xi = 1 - (Z_{out}/Z_0)\lambda$.

The boundary conditions for the acoustic pressures and velocities at the receiver side of the plate are given by [2]

$$P_1 = P_s e^{ik_s} + P_h e^{-i\omega t}$$

$$\nabla P_1 = i\omega p + \frac{\Delta P_s}{Z_{in}}$$

where $P_1$ is the transmitted sound pressure, $\gamma_h = k_d h$ and $k_p$ is the complex wave number in the porous medium.

When the plate is loaded by a fluid half-space, the front surface acoustic impedance is given by [16]

$$Z_{in} = Z_p - iZ_o \cot g(k_d h)$$

where $Z_p = \rho_c c_p$ is the characteristic impedance of the porous medium, $c_p$ is the complex sound speed in the porous medium and $\rho_c$ is the effective density of the fluid in porous medium. $Z_o$ is the acoustic impedance of the fluid half-space, given by $\rho_0 c_0$. The effective density and the bulk modulus are calculated by assuming that the material is rigid-porous and using an appropriate model (e.g. [16–18]).

### 2.2. Reflected sound pressure for a perforated porous plate

Consider that the plate shown in Fig. 1 is vibrating under a plane harmonic wave with the sound pressure $P_i$ given by $e^{i\omega t}$. Time dependence $\exp(-i\omega t)$ is understood. According to the Helmholtz integral formulation for a two-dimensional problem, the sound pressure $P_s(x)$ on the surface of the source side of the plate can be expressed as follows [19]:

$$P_s(x) = 2[P_i(x)]_{x=0} - 2 \int \left( P_s(x_0) \frac{\partial G(x|x_0)}{\partial n} - G(x|x_0) \frac{\partial P_s(x_0)}{\partial n} \right) dx_0$$

where $G(x|x_0)$ is the two-dimensional free-space Green’s function, which is given by $G(x|x_0) = (i/4) H_0^{(1)}(k_0|x-x_0)$. $H_0^{(1)}$ is the Hankel function of the first kind of order zero, which is given by $H_0^{(1)}(k_0|x-x_0) = \int_0^a \frac{\sin(k_0 x) dx_0}{\sin(k_0)}$, $n$ is the normal taken outward, perpendicular to the surface of the plate and $a$ is the dimension of the plate in $x$-direction. For a plane surface, the term involving $\frac{\partial G(x|x_0)}{\partial n}$ vanishes. Then Eq. (10) becomes

$$P_s(x) = 2[P_i(x)]_{x=0} + 2 \int \left( G(x|x_0) \frac{\partial P_s(x_0)}{\partial n} \right) dx_0$$

The normal gradient of the pressure at the boundary is $\frac{\partial P_s(x_0)}{\partial n} = i\rho_0 c_0 n(x_0)$. Then Eq. (11) becomes

$$P_s(x) = 2P_i(x) + \frac{1}{2} \int_0^a \rho_0 c_0 n(x_0) H_0^{(1)}(k_0|x-x_0) dx_0$$

Using the boundary condition (6) the acoustic pressure on the surface of the source side of the plate can be expressed as

$$P_s(x) = 2P_i(x) + \frac{1}{2} \int_0^a |\sigma|^2 \rho_0 c_0 \xi \omega w + i\omega h \chi_1 P_s(x_0) H_0^{(1)}(k_0|x-x_0)) dx_0$$

where $\chi_1$ is the specific acoustic admittance of the source side of the plate surface which is given by $(\rho_0 c_0 \lambda / Z_{in})$, and $w$ is the displacement of the plate.

Eq. (13) can be solved analytically by using the Fourier transform technique. The detailed procedures are given elsewhere [3,20]. Finally, the solution for the acoustic pressure can be expressed as [22]

$$P_s = \frac{i\rho_0 w_0^2 \xi / k_0 + 2(1 + \chi_1)}{1 + 2\chi_1}$$

The reflected sound pressure $P_r$ can be calculated by using the Helmholtz integral equation for a two-dimensional problem at a certain point $(x,z)$ and by substituting the boundary value $P_s$. Then $P_t$ becomes [22]

$$P_t(z) = \frac{i\rho_0 w_0^2 \xi / k_0 + e^{-i\omega t}}{1 + 2\chi_1}$$

The absorption coefficient of the plate can be calculated from

$$x_{abs} = 1 - |P_r / P_i|^2$$

### 2.3. Sound transmission through a perforated porous plate

In an analogous manner, by using the boundary condition (8) and the Helmholtz integral formulation the surface acoustic pressure $P_r$ on the receiver side can be expressed as follows [2]:

$$P_i(x) = -\frac{i}{2} \int_0^a [\rho_0 c_0^2 \xi w + i\omega h \chi_2 e^{i\omega t} P_s(x_0)] H_0^{(1)}(k_0|x-x_0) dx_0$$

where $\chi_2$ is the specific acoustic admittance of the receiver side of the plate surface, which is given by $(\rho_0 c_0 \lambda / Z_{in})$, and $e$ is the velocity transfer coefficient, which is given by

$$e = \frac{e^{i\omega t} - r_b e^{-i\omega t}}{1 - r_b}$$

where $r_b$ is the reflection coefficient of the receiver side of the plate, which is given by

$$r_b = Z_s - Z_p e^{2i\omega b}$$

By substituting the sound pressure and its normal derivative on the boundary surface into a Helmholtz integral formula, the transmitted sound pressure can be obtained as [4]
\[ P_1(z) = \frac{2\gamma_2 e^{\gamma_2 z} - i\rho_0 \omega^2 z}{1 + 2\gamma_2 e} \]  

(20)

The insertion loss IL is given by

\[ \text{IL} = 10 \log \left( \frac{|P_1|}{|P_2|} \right)^2 \]  

(21)

The acoustic impedance of the plate \( Z_p \), the complex wave number \( k_p \), the complex sound speed in the porous plate \( c_p \), and the effective density of the fluid in porous plate \( \rho_p \) are calculated by using the method of Allard et al. [16].

3. Measurements

3.1. Experimental procedure

Sound insertion loss measurements on two perforated porous plates have been carried out in the flow duct at the University of Hull designed by Cummings [9]. A diagram showing the flow duct test arrangement is given in Fig. 2. This flow duct has an internal cross section of 1.2 m \( \times \) 0.8 m and is 27.7 m long. The walls are made of 24 mm thick plywood and an outer cladding of 1.3 mm lead. The surface density of the wall is about 32 kg/m\(^2\). The air-flow rate can be changed by varying the fan blade pitch angle between \(-2^\circ\) and \(+24^\circ\). Four speakers of 600 W are mounted in the sound source section in the flow duct walls as shown in Fig. 2. A noise generator, type 1405, and a 1.6 kW power amplifier, type SR707, are used to feed the speakers. The sound in the test section is measured by a 1/2 in B&K microphone, type 4133, fitted in a B&K turbulence cancelling tube, type UA0436. A turbulence screen is fixed on a longitudinal rail fitted diagonally across and along the duct.

The output of the microphone was connected to a B&K amplifier, type 2609. The output of the amplifier was fed to an Ono Sokki two-channel FFT analyzer, type CF350Z. Two porous plates and a plywood plate were used in the insertion loss measurements. The plates were transversely mounted between two heavy steel frames in the test section of the flow duct. All of the gaps between the walls of the duct and the edges of the frame were sealed by putty.

3.2. Insertion loss measurements

The insertion loss can be determined from (22) [21]

\[ \text{IL} = T_{\text{pl}} - T_{\text{pl}} \]  

(22)

where \( T_{\text{pl}} \) is the spatial average sound pressure level in the frequency band in the test duct, when the porous plate is installed, and \( T_{\text{pl}} \) is the spatial average sound pressure level in the frequency band in the test duct, when the porous plate is removed.

The spatial averaged sound pressure level, \( Z_p \), has been calculated by measuring the local sound pressure levels at a minimum of three key positions equally spaced on the diagonal line shown in Fig. 3a. The local sound pressure levels were measured by a 1/2 in B&K microphone which was fitted in a B&K turbulence cancelling tube as shown in Fig. 3b. The length of this tube was bigger than a quarter of wavelength at the lowest frequency of interest. The spatial average sound pressure level, \( Z_p \), in dB, has been determined from the local sound pressure levels, \( T_{\text{pl}} \), using (23) [21]:

\[ T_p = 10 \log \left[ \frac{1}{n_m} \sum_{i=1}^{n_m} 10^{\frac{T_{\text{pl}}}{10}} \right] \]  

(23)

where \( n_m \) is the number of measurements.

The background noise produced by the air flow at each measurement position was at least 10 dB below the test signal. This was checked by measuring the sound pressure levels with the loudspeaker unit turned on and off. The air flow rate in the test series with and without the porous plate installed was the same. The microphone signal was analyzed in one-third octave bands. The perforated Black plate was transversely mounted in the flow duct as shown in Fig. 3c. The edges of the plate were clamped by using two steel frames. The area of the plate affected by the sound was reduced to 0.9 m \( \times \) 0.7 m because of the steel frame. The 1.8% and 14.4% perforations of the plate were provided by 10 holes of 44 mm diameter, and 25 holes of 84 mm diameter, respectively. The measured characteristics of the plates are given in Table 1.

Fig. 4 shows the measured insertion loss data for the perforated Black plate without air flow. For 1.8% perforation the insertion loss is higher throughout the frequency range due to the narrower air ways, and has maximum values between 1000 Hz and 5000 Hz. The insertion loss at 50 Hz for 14.4% perforation is not plotted being negative due to the undulations in the measured insertion loss.
Fig. 4 shows that increasing the perforation ratio decreases the measured insertion loss by about 6 dB. The measured insertion loss spectra for the Black plate in presence of air flow are shown in Fig. 5. The air flow speed in front of the hole at the receiver side of the plate is 52.37 m/s for 1.8% perforation ratio, and is 30.24 m/s for 14.4% perforation ratio, corresponding to Mach numbers of 0.153 and 0.088, respectively. The air flow speed on source side of the plate is 5.39 m/s. For 1.8% perforation the presence of air flow increases the measured insertion loss throughout the frequency range by about 3 dB. But the measured IL of the Black plate for 14.4% perforation ratio is not affected by air flow except for a very small change between 200 Hz and 700 Hz. According to Figs. 4 and 5 it seems that air flow does not affect the insertion loss of the Black plate for higher perforation ratios. The insertion losses below 125 Hz for 14.4% perforation ratio and below 15 Hz for 1.8% perforation ratio are not recorded because they were negative at these frequencies.

Table 1
The measured characteristics of the porous plates

<table>
<thead>
<tr>
<th></th>
<th>$L_x$ (m)</th>
<th>$L_y$ (m)</th>
<th>$h$ (m)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$E$ (Pa)</th>
<th>Loss factor</th>
<th>Porosity $\phi$</th>
<th>Poission ratio $\nu$</th>
<th>Flow resistivity (N s/m$^4$)</th>
<th>$x_\infty$</th>
</tr>
</thead>
<tbody>
<tr>
<td>YB10</td>
<td>0.5</td>
<td>0.5</td>
<td>0.01</td>
<td>353</td>
<td>$2.1 \times 10^4$</td>
<td>0.1</td>
<td>0.69</td>
<td>0.35</td>
<td>68 111</td>
<td>1.2</td>
</tr>
<tr>
<td>Black plate</td>
<td>1</td>
<td>1</td>
<td>0.021</td>
<td>223</td>
<td>$2.46 \times 10^6$</td>
<td>0.35</td>
<td>0.75</td>
<td>0.3</td>
<td>46 933</td>
<td>1.85</td>
</tr>
</tbody>
</table>

Fig. 3. (a) Microphone positions, (b) turbulence screen with a microphone fitted on a longitudinal rail, and (c) a perforated porous plate transversely mounted in the flow duct [22].
The measured insertion losses of YB10 plates with different perforation ratios in the absence of air flow are shown in Fig. 6. For both perforation ratios, the IL curves exhibit peaks between 315 Hz and 1000 Hz. The IL for 1.8% and 10.39% perforation ratios are not plotted below 31.5 Hz and 50 Hz, respectively. The measured IL of the YB10 is decreased by about 4 dB by increasing the perforation ratio.

The measured insertion losses of the YB10 plates with air flow are shown in Fig. 7. The air flow increases the IL by between 2 and 5 dB at low frequency for 1.8% perforation ratio, and decreases the IL of the plate by between 1 and 2 dB throughout the frequency range for 10.39% perforation ratio in comparison with no air flow. The air flow speed on the receiver side of the YB10 plate was 57.7 m/s for 1.8% perforation ratio, and 37.71 m/s for 10.39% perforation ratio, corresponding to Mach numbers of 0.168 and 0.110, respectively. The air flow speed on source side of the plate was 5.69 m/s.

### 3.3. The measured IL of the porous plates at different locations in the duct

To assess the effects of inserting perforated plates on the uniformity of the flow in the duct, the insertion loss has been measured with different plate locations. The plates which were previously mounted at a distance of 4.5 m from the microphone were moved to 2.5 m from the microphone. The perforated poroelastic plates were excited by the acoustic sound field without air flow.

The measured insertion losses of the Black plate mounted at different locations in the duct, in the absence of air flow, are compared in Fig. 8. It seems that mounting the plate at different locations does not affect the measured IL very much in the absence of air flow. In the case of 1.8% perforation ratio the IL of the Black plate (when distance between Microphone and plate was 2.5 m) is not recorded below 50 Hz where it was negative. The measured IL only changes between 100 Hz and 300 Hz with different plate positions. For 14.4% perforation the measured IL data at both locations have similar values, even though there is a small difference at 50 Hz. Fig. 9 compares the measured IL of the YB10 plates at different locations in the duct without mean air flow. For 1.8% perforation ratio the IL data are not completely similar, and differ by between 0.5 and 2 dB throughout frequency range. The peak IL is decreased by about 2 dB when the plate is mounted closer to microphone. For the higher perforation, the IL results

![Fig. 5. The measured insertion loss of the perforated Black plate with air flow.](image1)

![Fig. 6. The measured insertion loss of the perforated YB10 plates without air flow.](image2)

![Fig. 7. The measured insertion loss of the perforated YB10 plates with air flow.](image3)
are similar at low frequency, and are little different between 1000 Hz and 2500 Hz. The peak insertion loss is similar at both locations. The IL data are not plotted at below 100 Hz where they were negative.

Fig. 10 compares the measured IL of YB10 plates for different perforations and different locations in the duct in the presence of mean air flow. For the higher perforation, the IL is increased at all frequencies by mounting the plate closer to microphone. For 1.8% perforation the results differ only slightly for frequencies between 40 Hz and 70 Hz, between 800 Hz and 1250 Hz, and between 4000 Hz and 5000 Hz.

4. Comparisons between predicted and measured IL without flow

In this section, we compare numerical predictions of insertion loss of perforated infinite plates to the measured insertion loss of two finite perforated poroelastic plates for different perforation ratios but without mean air flow. Eq. (20) was used to calculate the insertion loss of the clamped rectangular perforated plates. Fig. 11 shows the measured and predicted insertion loss spectra for the perforated YB10 plates. For 1.8% perforation ratio, the agreement between measured and predicted IL is reasonably good at low frequency but there is a discrepancy of between 1 dB and 2 dB at high frequency between measurements and predictions. For 10.39% perforation ratio, the
measured insertion loss, agrees reasonably well with the predicted insertion loss except for a discrepancy of about 4 dB at frequencies between 3000 Hz and 5000 Hz.

A comparison of the measured and predicted insertion loss of the Black plates in the absence of air flow is shown in Fig. 12. For 1.8% perforation the measured and predicted IL agree well apart from the discrepancies observed at higher frequencies. For 14.4% perforation ratio, the agreement between the predicted and measured IL is satisfactory although there are discrepancies at very low and high frequencies.

5. Conclusion

The insertion losses of two different types of perforated porous plates have been measured for two different perforation ratios and in two different locations in a flow duct with and without air flow. For the higher perforation ratio, the measured insertion loss of porous plates decreases. Mounting the plates at different locations in the duct did not affect the measured insertion loss significantly in the absence of air flow. But in presence of air flow, the measured insertion loss of YB10 plate with the higher perforation ratio increases when the plate is mounted closer to the microphone. The measured insertion losses of perforated porous plates with air flow are not compared with the predictions since a model for the insertion losses of perforated porous plates with air flow has not yet been formulated. However, an analytical model that takes into account the effect of perforations and the effect of the flexural vibrations in the plates has been formulated and used to calculate the insertion loss in the absence of air flow. The agreement between measured and predicted insertion loss of perforated porous plates in the absence of air flow is satisfactory. The observed discrepancies between data and predictions for insertion loss without flow may, in part, be the results of the assumption of rigid-frame when calculating the acoustical characteristics of the plate materials.

References