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Insertion Loss of Clamped Perforated and Non-Perforated Poroelastic Plate Silencers with Flow

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Summary

Idealized silencers consisting of cavity-backed clamped porous elastic plates mounted on fairings parallel to the direction of flow can result in low frequency attenuation of noise generated in a duct. The insertion loss (IL) of such silencers has been measured without and with perforations in the plates, without and with mean air flow and without and with noise generated by loudspeakers placed downstream in the flow duct system. Whether the plate is perforated or not but without noise generated by loudspeakers in the duct, the IL due to the poroelastic plate silencer is found to increase in the presence of mean air flow at speeds of 37.65 m/s and 5.5 m/s. With noise from the loudspeakers and a mean air flow of 37.65 m/s, the IL when the plate is perforated is higher than that achieved with the non-perforated plate at frequencies less than 100 Hz. To investigate the extent to which there may be structure-borne components of the plate deflection, the displacement of the duct wall has been measured and plotted versus frequency. The displacement of the non-perforated plate is found to be higher than the displacement of the duct wall. However the displacement of the perforated plate is less than that of the duct wall which suggests that low frequency improvement in the performance in the presence of mean air flow and perforations is a consequence of the aeroacoustic effects of the perforations rather than increased plate deflection.

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1. Introduction

Passive noise control in ducts has been achieved using structural, reactive and dissipative devices [1, 2, 3, 4, 5, 6, 7]. Huang [1, 2] has carried out theoretical and experimental work on structural devices involving the duct walls. Ramamoorthy *et al.* [3] have used experimental and theoretical methods to investigate the insertion loss of a passive structural acoustic silencer taking the effect of the external fluid into account. Also they have presented an interesting relationship between the insertion loss and plate dispersion. Tang and Lin [6] have studied the resonant mass-spring behaviour of a stiff light composite panel absorber, and have investigated its applicability for noise reduction. Astley *et al.* [6] have examined some of the effects of wall flexibility on sound propagation and attenuation in a duct with a bulk porous liner and negligible mean flow. Cummings and Chang [7] have presented models that describe the effect of internal mean flow on the bulk acoustic properties of a porous medium and have predicted the effect of the absorbent on the sound transmission loss of the silencer. Typically dissipative silencers are designed to exploit the most widely-acknowledged absorption mechanism in porous materials viz. viscous friction due to relative motion between solid and fluid. If the frame of the

porous material is viscoelastic then other dissipative mechanisms are possible. To obtain good low frequency absorption, say below 300 Hz, exploiting the viscous friction mechanism alone may require an unacceptably large thickness of material. On the other hand, when designing passive sound absorbers based on porous materials or membranes, it is known that the presence of a backing air gap with properly chosen dimensions can enhance their low frequency performance. Furthermore, if an elastic porous plate is clamped at its edges, then a backing air cavity will allow bending modes which could lead to significant absorption through viscoelastic losses at relatively low frequencies. Indeed useful low frequency absorption maxima have been observed for configurations consisting of clamped poroelastic plates with an air gap behind them [8]. Perforated plates are used in exhaust silencers, noise attenuators in ducts, and duct linings in jet engines. Recently Aygun *et al.* [9, 10] have investigated the effects of inserting a perforated porous plate placed transversely across the duct on the uniformity of flow and the sound absorption in the duct. These effects were assessed by measuring the insertion loss at different locations in the duct. A parallel impedance model was used to model the effects of perforation.

This paper explores the possibility that the combination of bending vibration, the presence of an air gap and the higher frequency visco-thermal sound attenuation in a duct-wall silencer using a porous elastic plate mounted

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parallel to flow and with an airgap might give useful acoustic performance over a wide frequency range. The effects of mean flow rate and of perforations in the plate are explored also.

Measurements have been made in a flow duct of the sound insertion loss (IL) of simple silencer systems consisting of clamped non-perforated and perforated porous elastic plates with an air cavity backing. The open cell foam plates were made from recycled materials at the University of Bradford. Each of the porous elastic plates were mounted on fairings near to the duct wall parallel to the direction of the air flow and separated from the walls by an air cavity. The purpose of separating the plate from the walls by an air cavity is to allow bending vibration which should result in attenuation due to viscoelasticity as well as flow resistivity. Separating the plate from the wall by an air cavity causes the resonance frequencies of the plate to increase because the stiffness of the system is increased by the air cavity.

In mean flow, the poroelastic plate silencer is excited by lateral components of the air flow and by acoustic plane waves produced from loudspeakers. A fraction of the acoustic energy vibrates the plate in flexural modes. The resulting vibrational energy is partly converted into heat and dissipated in the structure of the porous frame and partly re-radiated at the resonance frequencies of the plate. Another fraction of the sound energy is dissipated through the visco-thermal interactions between the solid and the fluid in the pores. The vibration losses have the greatest effect at low frequencies while the visco-thermal dissipation has a significant effect at high frequencies. The sound insertion loss has been calculated from the differences between averages of several sound pressure measurements either side of the porous plate structure in the flow duct. To investigate the contributions of plate vibrations to the measured IL, the displacements of a perforated and non-perforated porous elastic plates separated from duct wall by an air cavity in the duct have been measured and compared with those of the duct wall.

2. Construction of the clamped plate silencer

A schematic of the porous plate silencer is shown in Figure 1 which also specifies x - y coordinates. A 20 mm thick poroelastic plate with length $a = 1$ m in the x -direction and height 0.8 m in the y -direction is mounted parallel to one side of the duct wall inside the flow duct, and separated from the wall by an air cavity. The length and breadth of the cavity are the same as those of the porous plate, and the cavity depth is 60 mm. The plate is mounted parallel to the direction of the air flow.

A steel frame having $0.7 \text{ m} \times 0.9 \text{ m}$ internal dimension with 3 mm thickness is used to hold the plate. The plate dimensions are reduced to $0.9 \text{ m} \times 0.7 \text{ m}$ as a result of using this steel frame. Two steel fairings are attached to edges of the plate to encourage the sound waves and air flow to travel in the direction parallel to the plate. There

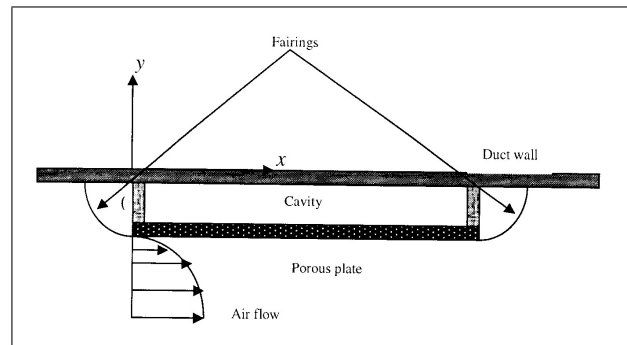


Figure 1. A poroelastic plate mounted parallel to the duct wall with an air cavity to the rear.

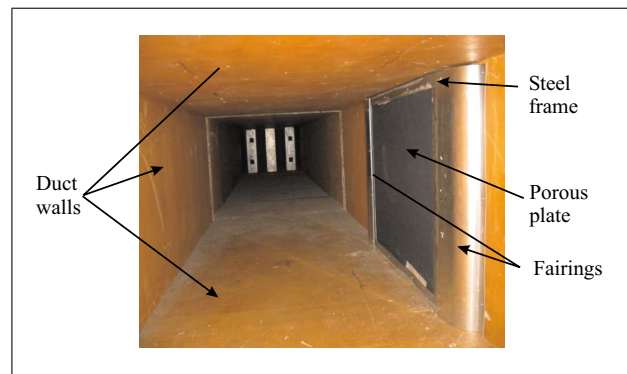


Figure 2. A $1 \text{ m} \times 0.8 \text{ m} \times 20 \text{ mm}$ thick porous elastic plate mounted in fairings on a side wall in the duct.

are two fluid/plate interfaces at the front and rear of the plate. The plate is excited by the incoming acoustic plane wave and the air flow. Plane waves propagate only in the regions $x < 0$ and $x > a$. Figure 2 shows the porous plate mounted in the steel fairings on the wall of the flow duct.

3. Flow duct measurement system

The flow duct measurement system is shown in Figure 3. The internal cross-section of the duct is $1.2 \text{ m} \times 0.8 \text{ m}$. By means of the fan impeller, volume flow rates of up to $17 \text{ m}^3/\text{s}$ can be achieved at a pressure differential of approximately 120 mm water across the fan inlet. The flow rate is changed by varying the fan blade pitch angle between -2° and $+24^\circ$. As well as the noise caused by the fan and air flow, sound is created inside the duct by four 600 W bass loudspeakers fed with random noise filtered in octave frequency bands from a B&K noise generator, type 1405, and a 1.6 kW power amplifier, type SR707. The loudspeakers are mounted downstream of the microphone positions i.e. the airflow was in the direction towards the loudspeakers from the microphone. A $\frac{1}{2}$ " B&K microphone, type 4133, fitted with a B&K turbulence cancelling tube, type UA0436, is used to sample the acoustic sound field. The microphone is supported by a preamplifier that is connected to a B&K measuring amplifier, type 2609. The output of the measuring amplifier is fed to a two channel FFT analyzer, type CF350Z.

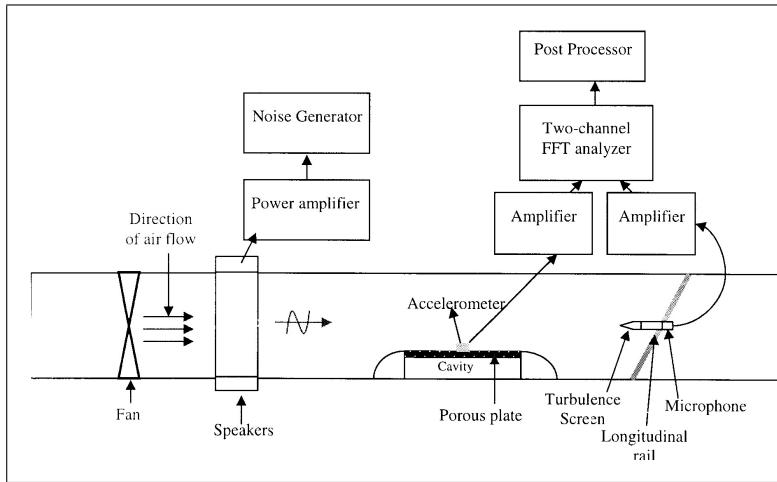


Figure 3. Flow duct measurement arrangement; dimensions in the figure are not to scale.

The signal to noise ratio was between 12 dB and 17 dB for measurements made in the presence of the silencer, and was between 12.5 dB and 16.5 dB for measurements made without the silencer. The usable frequency range of the flow duct for insertion loss measurements is between 31.5 Hz and 5000 Hz.

The porous elastic plate, made of recycled car dashboard materials, is clamped between a steel frame and a wooden support. The properties of this plate have been measured at the University of Bradford and are given in Table I. The insertion loss data reported here has been measured with the clamped plate mounted on a vertical wall inside the flow duct (see Figure 2). The IL of the clamped plate system mounted at the top or bottom inside the duct has been measured but the corresponding values are negative throughout the frequency range and are not shown here. With top or bottom mounting, the clamped poroelastic plate is curved and under tension as a result of gravitational self-loading. The observed negative insertion loss might be the result of such self-loading.

The perforations consisted of nine 42 mm diameter holes giving an area perforation ratio of 1.6%.

4. Insertion loss results

The procedure described elsewhere [9, 10] and the recommendations given in ISO standard 7235 [11] have been followed during measurements. The insertion loss (IL) of the poroelastic plate silencer has been calculated from sound pressure measurements in the duct according to [11],

$$IL = \bar{L}_{pII} - \bar{L}_{pI}, \quad (1)$$

where $\bar{L}_{pI}$ is the spatial average sound pressure level in the frequency band in the test duct, when the test silencer is installed, and $\bar{L}_{pII}$ is the spatial average sound pressure level in the frequency band in the test duct, when the silencer is not present.

The sound pressure level, $\bar{L}_p$, has been calculated by measuring the local sound pressure levels at least at three key positions equally spaced on the longitudinal span of

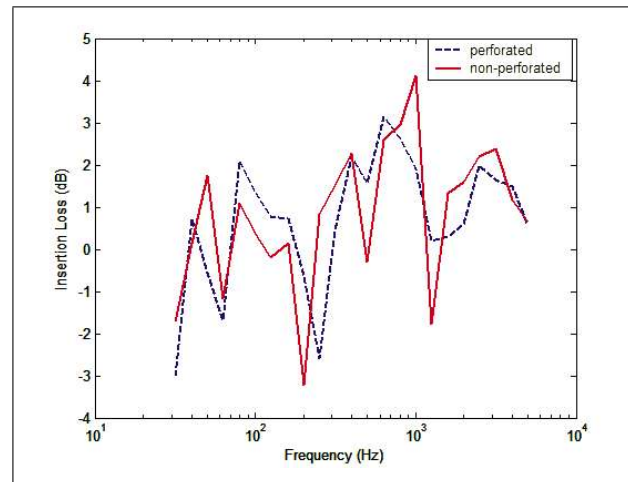


Figure 4. Measured insertion loss of a perforated and non-perforated porous plates with loudspeaker noise but without mean flow.

the line. The spatial average sound pressure level, $\bar{L}_p$, in dB, has been determined from the local sound pressure levels, $\bar{L}_{pi}$, using [11]

$$\bar{L}_p = 10 \log \left[\frac{1}{n_m} \sum_{i=1}^{n_m} 10^{\bar{L}_{pi}/10} \right], \quad (2)$$

where n_m is the number of measurements.

The measured IL spectra of non-perforated and perforated poroelastic plate silencers without mean flow but with noise from the loudspeakers are shown in Figure 4. The maximum IL when the plate is not perforated has a value of 4.1 dB at 1 kHz. With perforations, the maximum IL is 3.2 dB at 630 Hz. Negative minima in IL occur near 30, 60 and 200 Hz. For the non-perforated plate there is an additional negative minimum near 1100 Hz. The high frequency minimum might be the result of the waves not being plane and flanking paths. At low frequencies the negative IL minima might be the result of scattering from the fairings.

IL measurements have been carried out also in the presence of average air flow speeds of 5.5 m/s and 37.65 m/s

Table I. The assumed characteristics of the porous plate.

L_x	L_y	h	ρ	E	Loss factor	Porosity ϕ	Poisson ratio ν	Flow resistivity	α_∞
0.9 m	0.7 m	21 m	223 kg/m ³	2.46·10 ⁶ Pa	0.35	0.75	0.3	4633 N s/m ⁴	1.85

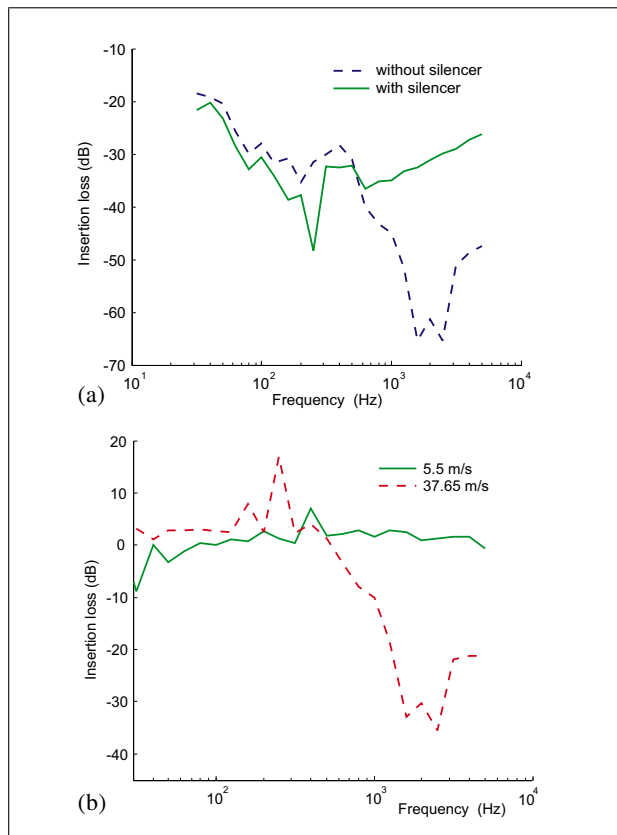


Figure 5. (a) Measured sound levels without and with the silencer during air flow at a mean speed of 37.65 m/s. (b) Measured insertion losses of the poroelastic plate silencers with mean air flow at speeds of 5.5 m/s and 37.65 m/s.

in the duct, corresponding to Mach number (M) values of 0.016 and 0.11 respectively but without the loudspeakers operating, so that the noise in the duct was due only to the air flow, the fan and background noise in the laboratory. As shown in Figure 5a, above 500 Hz in the presence of the silencer the noise with air flow at a mean speed of 37.65 m/s is higher than without the silencer. This suggests that the negative IL at higher frequencies, when there is only mean flow and no noise from the loudspeakers, is caused by additional aerodynamic noise due to the presence of the silencer. The corresponding IL spectra are shown in Figure 5b. The measured IL with a mean air flow speed corresponding to $M = 0.11$ has a peak of 16.4 dB at 250 Hz. As might be expected from Figure 5a data, above 500 Hz, the IL values for the higher Mach number flow are below zero.

IL measurements for perforated and non-perforated porous plate silencers have been carried out also in presence of both mean air flow (with flow rates corresponding to $M = 0.056$ and 0.11) and noise from the loudspeakers.

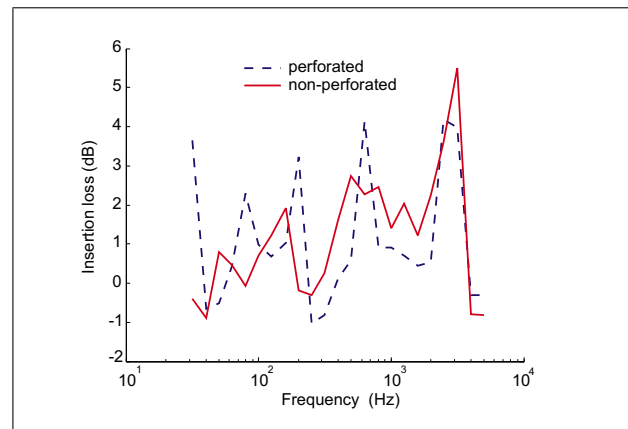


Figure 6. Measured insertion loss of perforated and non-perforated porous plate silencers with the lower air flow rate ($M = 0.056$) and in the presence of noise from the loudspeakers.

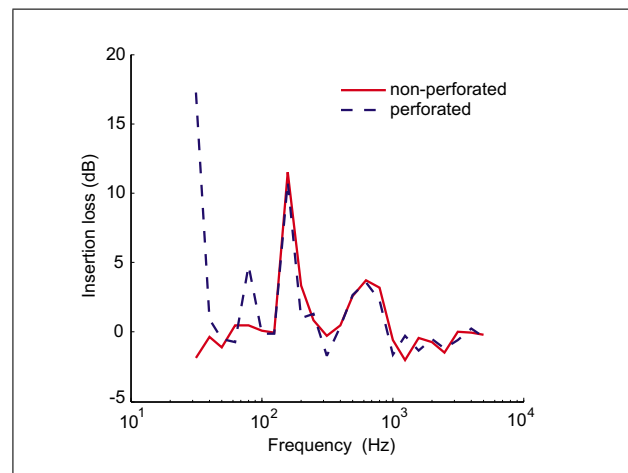


Figure 7. Measured insertion loss of a perforated and non-perforated porous plate with the higher rate of air flow ($M = 0.11$) and in the presence of noise from the loudspeakers.

The latter increases the in-duct noise spectrum particularly above 300 Hz but, in the presence of mean flow, noise due to the loudspeakers has been observed to decrease at the higher flow rate, possibly as a consequence of refraction within the duct between microphone and loudspeaker positions. The resulting data are compared in Figures 6 and 7. Figure 6, for the lower flow rate shows a more or less positive IL at all frequencies with several peaks. There is a maximum of 5.4 dB for the non-perforated porous elastic plate at 3150 Hz. With perforations in the plate, the IL peaks observed at frequencies below 1000 Hz are higher than the IL peak observed without perforation. At the higher flow rate, Figure 7 shows that the IL values for perforated and non perforated porous elastic plates are similar

Table II. Insertion loss peaks IL obtained with the PP (perforated plate) and NPP (non perforated plate) with and without air flow and noise for Mach numbers $M = 0.016, 0.056$ and 0.11 . All values in dB.

Frequency (Hz)	40	50	80	160	200	250	400	500	630	1000	2500	3150
IL _{noise} , NPP		1.5	1				2.25			4		2.4
IL _{noise} , PP	0.75		2				2.2		3.2		2	
IL _{airflow} , NPP, $M = 0.016$					2.8		7					
IL _{airflow} , NPP, $M = 0.11$				8		17	4					
IL _{noise+airflow} , NPP, $M = 0.056$		0.9		1.9	3.2			2.8				5.5
IL _{noise+airflow} , PP, $M = 0.056$			2.2						4.1		4.2	
IL _{noise+airflow} , NPP, $M = 0.11$				11.5					3.6			
IL _{noise+airflow} , PP, $M = 0.11$			4.8	11					3.6			

above 100 Hz. However the peak IL at 160 Hz is significantly higher, 11.5 dB without perforations and 11 dB with perforations, than any observed at the lower flow rate. At frequencies below 100 Hz for the higher flow rate, the IL measured for perforated plate has been found to be higher than that obtained using the non-perforated plate. Overall, it seems that increasing air flow velocity decreases the insertion loss of poroelastic plate silencers at high frequency and increases their insertion loss at low frequency. Somewhat counter-intuitively, the perforations in the plate results in higher insertion loss peaks at low frequencies as shown in Figures 4, 6 and 7.

The magnitudes of the measured positive insertion loss peaks for perforated and non perforated porous elastic plate silencers with and without air flow and noise are shown in Table II. The maximum insertion loss observed throughout measurements is 17 dB achieved at 250 Hz without perforation in the presence of average air flow speed 37.65 m/s in the duct ($M = 0.11$).

5. Results for the displacement of the plate

Separating the plate from the wall by an air cavity causes an increase in the resonance frequency of the plate because stiffness of the system is increased by the air cavity. To investigate the extent to which the poroelastic plate silencer performance depends on the plate deflection, measurements have been performed of the deflection of the perforated and non perforated poroelastic plates in the silencer arrangement.

The measured porous elastic plate displacements are plotted versus frequency in Figure 8 and Figure 9 for three different cases. The measured displacements of the non perforated poroelastic plate, with and without noise, in the presence of air flow at a speed of 5.5 m/s are similar to each other. This suggests that the addition of noise from the loudspeakers does not change the displacement of the plate very much when there is air flow. The measured displacements of the perforated poroelastic plate with and without noise, in the presence of air flow, are similar to each other as shown in Figure 9. However the displacement of the perforated plate is less than that of the non-perforated plate.

To investigate the extent to which there may be structure-borne components of the plate deflection, the dis-

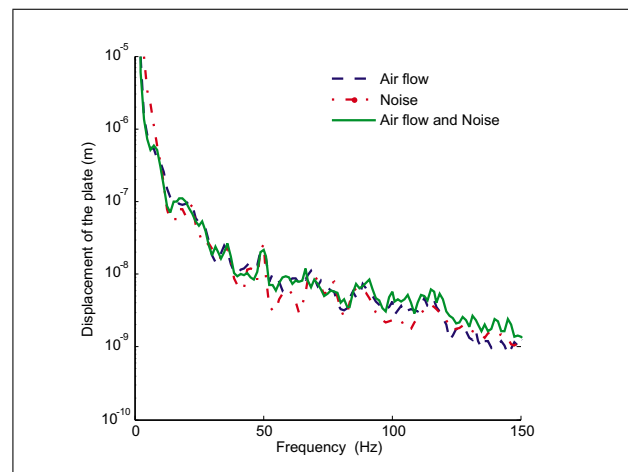


Figure 8. Measured displacement of the non-perforated porous elastic plate in the Flow Duct with loudspeaker noise alone, with air flow with a mean speed of 5.5 m/s, and with airflow and noise.

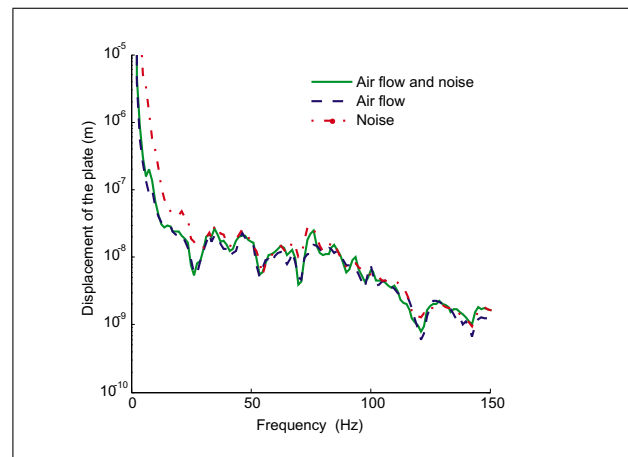


Figure 9. Measured displacement of perforated porous elastic plate in the Flow Duct with loudspeaker noise alone, with air flow with a mean speed of 5.5 m/s, and with airflow and noise.

placement of the duct wall has been measured and is plotted versus frequency in Figure 10. The displacement of the duct wall is comparable with the displacement of the perforated porous plate. The displacement spectra shown in Figures 9 and 10 are similar except for the resonant peaks observed at 35 Hz and 75 Hz in Figure 10 which are close

Table III. Experimentally-observed resonance frequencies (Hz) for the deflection of the porous plates mounted with an air cavity on the duct wall. I: Non-perforated plate, II: Perforated plate, III: Duct wall.

I	Air flow and noise	18.75	36.25	48.75	61.25	71.25	86	102.5	113.5	130	148.8
	Air flow	18.75	36.25	48.75	61.25	71.25	86.25	100	115	127.5	148.8
	Noise	18.75	36.25	50.00	60	71.25	86.25	101.3	115	126.3	
II	Air flow and noise	7.50	35	47.5	62.5	67.5	76.25	85	93.75	100	128.75
	Air flow	7.50	36.25	46.25	62.5	67.5	76.25	86.25	100	127.5	
	Noise	21.5	36.25	46.25	62.5	67.5	75	83.75	115	128.75	147.50
III	Air flow and noise	11.25	18.75	28.75	35.00	48.75	52.25	73.75	112.50		
	Air flow	11.25	16.25	27.50	35.00	48.75	52.50	73.75	80.00	96.25	101.25
	Noise	11.25	21.25	27.50	35.00	50.00	53.75	60.00	75.00	81.25	113.75

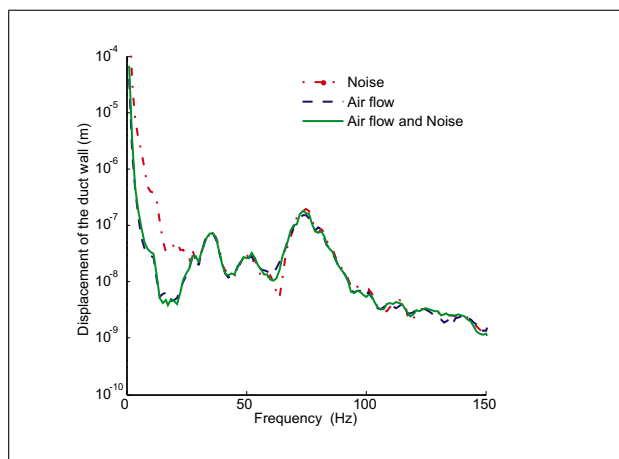


Figure 10. Measured displacement of the Duct wall with loud-speaker noise alone, with air flow with a mean speed of 5.5 m/s, and with airflow and noise.

to the frequencies at which the insertion loss of the porous plate is found to be negative (see Figure 4). The measured resonance frequencies (in Hz) for the displacement of the porous elastic plate and the displacement of the duct wall are shown in Table III.

6. Conclusion

The insertion loss spectra of perforated and non perforated poroelastic plate silencers mounted parallel to a duct wall without and with air flow have been deduced from sound pressure measurements versus frequency. The results show that, in the presence of airflow, perforation increases the low frequency IL. It has been observed that, whether or not the plate is perforated, increasing air flow velocity decreases the insertion loss at high frequency and increases the insertion loss at low frequency. The plate and duct wall deflections have been measured also. The introduction of air flow has been observed to increase the plate deflection. In the case of the non-perforated plate this is consistent with the hypothesis that the plate deflection is due to the lateral components of both the air flow and the acoustic plane wave. In the presence of air flow, perforation has been found to increase the insertion loss of the plate at low

frequencies but to decrease the displacement of the plate in comparison with non perforated poroelastic plate. Measurements of the displacement of the duct wall show that, in the case of the perforated plate, the plate deflection is mainly structure borne which suggests that the low frequency improvement is associated with the perforations.

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