Measurement of the sound transmission loss of a small expansion chamber muffler to consider the effects of mean flow and wall compliance

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ABSTRACT

Two matters which may be important in the modelling of small mufflers for Continuous Positive Airway Pressure devices are the effects of mean flow and of compliant walls. A preliminary experimental investigation using a simple expansion chamber muffler has been undertaken to assess the likely importance of these matters and the results are presented in this paper. The frequency dependent sound transmission loss of the expansion chamber muffler in the presence of mean flow and when it was fitted with compliant walls, was measured using a novel technique involving passing transient acoustic waves through long conduits attached to the muffler. The validity of the measurement technique was demonstrated. Results show that at the low mean flow velocities of interest, the mean flow has little effect but wall compliance can be significant.

INTRODUCTION

Noise is produced by a small centrifugal fan, the essential component in a Continuous Positive Airway Pressure (CPAP) device. These sleep apnea assisted breathing devices function by using the small high speed fan to produce a positive pressure in the nasal passages of the user. The fan runs at varying speeds (up to 30,000 rpm), and because of its high and varying speed, generates noise which is annoying to the user and others. Often a small predominantly reactive muffler is used to attenuate one component of the noise, that which is transmitted along the air path between the fan and the user.

Using an idealised muffler, this paper presents the results of a preliminary experimental study of several potentially important matters which should be considered in the modelling of these mufflers for CPAP devices. The first aspect is the effect of the mean flow through the muffler. It is generally thought that quite high mean flow velocities (>Mach number M=0.1) must be present before the mean flow produces a significant effect on the acoustic performance of the muffler other than that due to flow generated noise (Peat 1988; Ji Zhenlin *et al.* 1994; Munjal 1997).

The second matter is the effect of compliant (yielding) boundaries of the muffler. Often the mufflers used for sleep apnea assisted breathing devices are very small, irregularly shaped, and are manufactured using injection moulded plastic. The moulds are expensive to make, and hence it is important to ensure that the design of the moulded mufflers provide high acoustic attenuation. It is of interest to consider the effects of compliant boundaries in the modelling of these mufflers. Although the use of sound absorbing materials (for example, glass fibre, polymeric fibrous materials, various types of foams) are commonly used as linings in ducts and mufflers (Craggs 1977; Song and Bolton 2000), and as multi-layered materials (Lee and Wang 2006), there has been very little research on the effects of compliant boundaries on muffler acoustic performance.

THE ACOUSTIC PERFORMANCE MEASURE

A muffler of the type of interest here is shown in Figure 1.

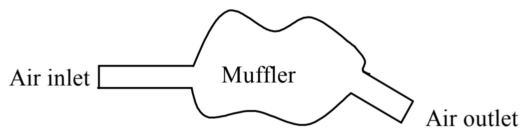

Figure 1. General predominantly reactive muffler.

The shape of the muffler body may be irregular to make use of the available volume. However the nomimal sizes of the inlet and outlet ports are often quite small, for example, 15 to 30 mm in diameter, hence nearly all of the acoustic energy entering and leaving the muffler up to quite high frequencies is carried by plane waves.

A useful measure of the acoustic performance which depends only on the muffler and not also on connected elements is the frequency dependent sound transmission loss. The frequency dependent sound transmission loss is defined as the ratio of the sound energy incident on the muffler inlet to that of the sound energy leaving the muffler at the outlet. As previously noted, most of the energy entering and leaving the muffler is carried by plane waves. Hence, the sound energy incident on and transmitted from the muffler can be found from the acoustic pressures associated with the incident and transmitted acoustic waves. If the acoustic pressure time histories of an appropriate transient incident wave and the corresponding transmitted wave are Fourier Transformed, and the ratio of the modulii of their transforms are taken and expressed in logarithmic form, the required frequency dependent sound transmission loss of the muffler can be found, that is:

$$TL = 20 \log \frac{\mathrm{FFT}_1}{\mathrm{FFT}_2} \qquad (1)$$

where FFT_1 and FFT_2 are the Fourier Transforms of the time histories of the incident and transmitted waves, respectively.

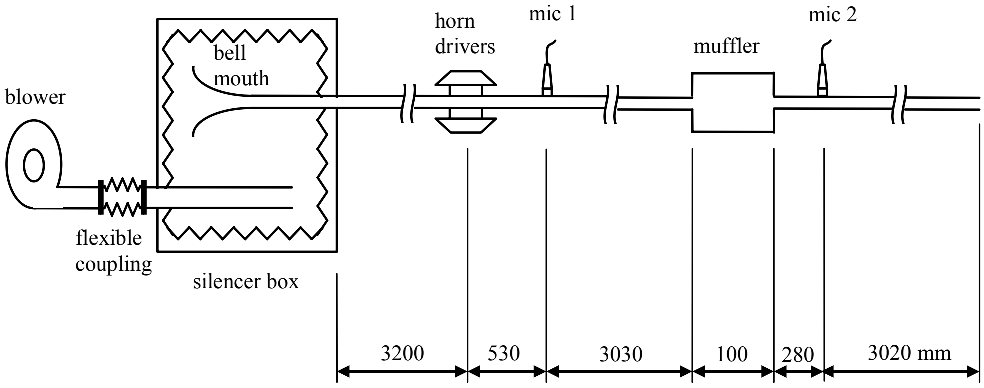

Figure 2. Schematic diagram of the experimental set-up to measure the transmission loss of an expansion chamber muffler.

THE MEASUREMENT TECHNIQUE

The frequency dependent sound transmission loss was measured using a rig with the features shown in Figure 2. The air from a small centrifugal blower was directed via a flexible connector to a silencer box, a cube with nominal internal dimensions of 600 mm x 600 mm x 600 m and lined with 50 mm thick mineral wool. The air was then directed by a 130mm diameter bell mouth into a 16 mm internal diameter plastic conduit pipe. A photograph of the blower, flexible connecter, silencer box and bell mouth on the inlet pipe is shown in Figure 3. Approximately 3 m along this pipe was a fitting which allowed a pair of horn drivers to be connected to the pipe. These horn drivers acted as the sound source. Downstream of the sound source was a fitting which carried the flush mounted inlet (or upstream) microphone. Downstream of this microphone was approximately 3 m of plastic conduit pipe which lead to the muffler to be tested. Close to the outlet of the muffler, the outlet (or downstream) microphone was mounted, followed by a further 3 m of plastic conduit. At the centre of the outlet of this conduit, a hot wire anemometer was located to allow an estimate of the mean flow velocity in the pipe to be made. Figure 4 shows a photograph of the experimental rig.

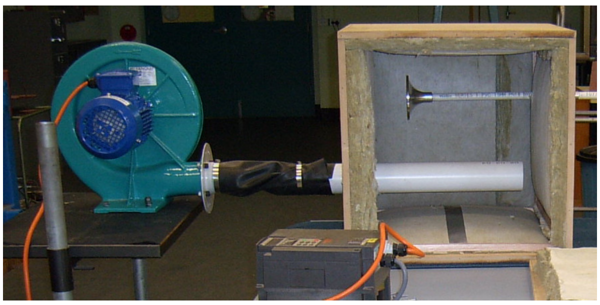

Figure 3. Photograph of the blower, flexible connecter and interior of the silencer box, showing the bell mouth on the inlet pipe.

Relative to the conduit internal diameter, long lengths of pipe were used such that the incident and transmitted waves could be studied without interference from reflected waves generated at the muffler and the pipe ends. The wave reflected from the muffler inlet travels approximately 6 m before it passes the inlet microphone, roughly 17.5 ms after the incident wave. Similarly, a transient wave from the sound source (horn drivers) will propagate upstream into the silencer box and be reflected, and will also pass the inlet microphone. The transient wave emerging from the muffler is sensed by the outlet (downstream) microphone. Again a reflection will occur at the open end where the hot wire anemometer is located, and will be reflected back to the muffler.

Time histories were captured for a number of individual pulses with the results averaged in the time domain before obtaining the spectrum of this averaged time history. This process removes some of the noise associated with the mean flow in the pipe and is known as signal enhancement. It is important to use appropriate triggers and windows using this technique such that the signal is always captured at the same time instant, in this case the best results were achieved when the analysis was triggered off the inlet microphone signal. Prior to carrying out a Fourier Transform, the averaged time signals were weighted using a transient window such that only the initial pulse was used for the analysis and not the subsequent pulses due to reflections. The lengths of the conduits attached to the muffler were chosen in order to allow the windowing to be carried out effectively. After processing, the captured time histories were used to calculate the transmission loss by the method previously described.

Figure 4. Photograph of the experimental rig.

An example of the time histories recorded by the inlet and outlet microphones is shown in Figure 5, along with the windows which allow the time histories of the unwanted reflected waves to be windowed out. The modulii of the Fourier Transforms of the windowed time histories presented in Figure 5 are shown in Figure 6, and the frequency dependent sound transmission loss found from these two Fourier Transforms as stated in equation (1) is given in Figure 7.

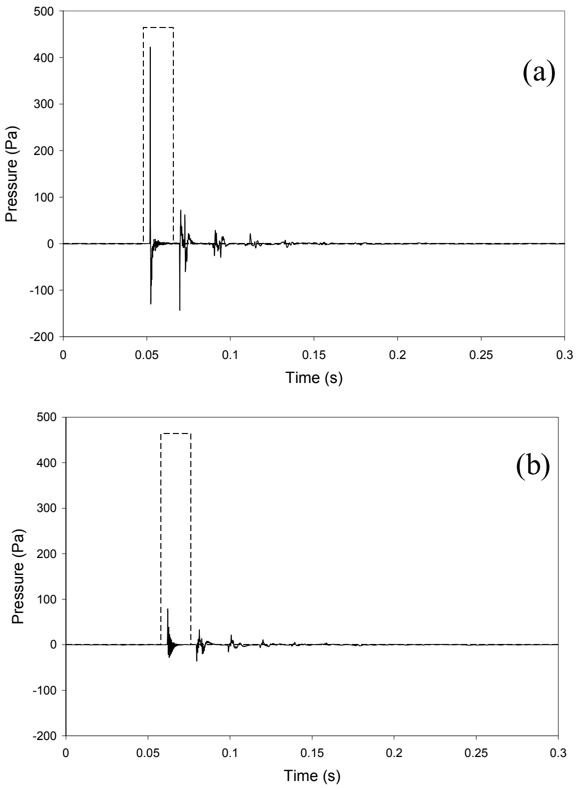

Figure 5. Time histories recorded by the inlet (a) and outlet (b) microphones and the windows applied to them.

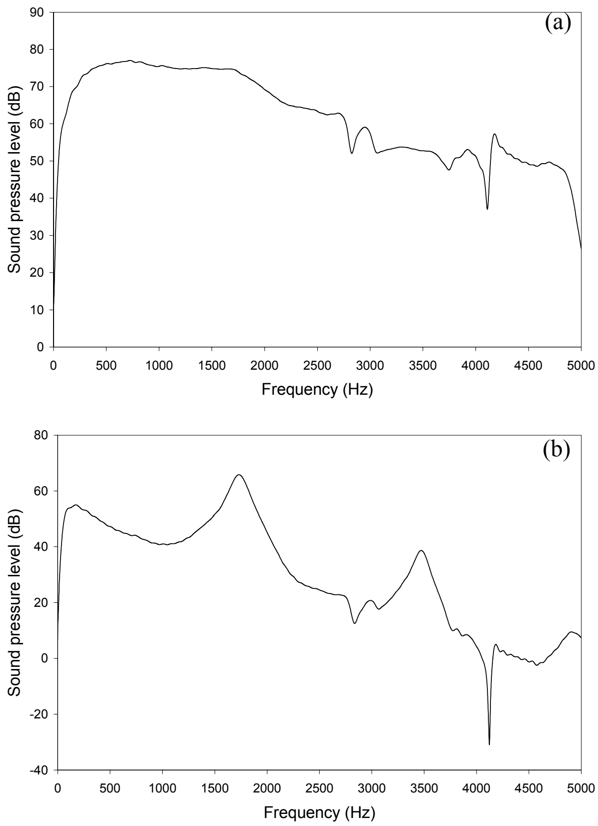

Figure 6. Modulii of the Fourier Transforms of the windowed time histories recorded by the inlet (a) and outlet (b) microphones.

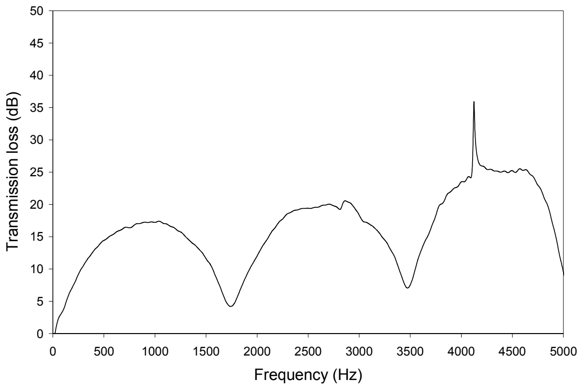

Figure 7. Frequency dependent sound transmission loss of the simple expansion chamber muffler.

The simple expansion chamber muffler used in these preliminary experiments was constructed from aluminium sections bolted together. Figure 8 shows a photograph of the rectangular aluminium expansion chamber muffler and the downstream microphone. The expansion section had an internal cross-section of 50 mm x 50 mm and a length of 100 mm, the inlet and outlet had a diameter of 16.2mm. The area expansion ratio was calculated to be 12.13. The lid could be removed and various compliant walls could be inserted to replace the rigid aluminium section.

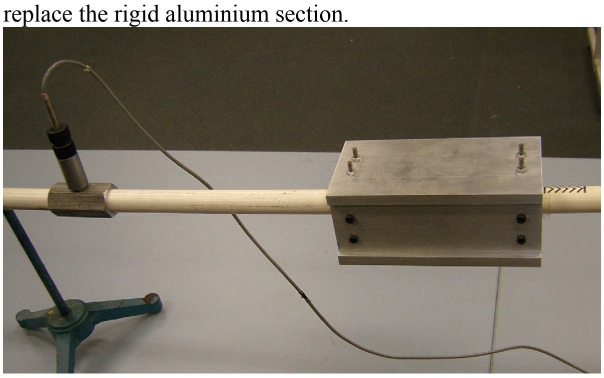

Figure 8. Photograph of the rectangular aluminium expansion chamber muffler and the downstream microphone.

Since the inlet and outlet microphones are more than 3 m apart, an acoustic wave propagating down the conduit whose internal diameter is only 16.2 mm will experience signficant attenuation. This frequency dependent attenuation must be subtracted from the total attenuation measured with the muffler in place to give the attenuation due to the muffler alone. To achieve this, the muffler was replaced with an equivalent length of pipe. This frequency dependent attenuation expressed as a transmission loss for a 3.41 m length of conduit (from inlet to outlet microphone) is shown in Figure 9.

It is of interest to note that there is a spike in the transmission loss at around 4200 Hz in the muffler results shown in Figure 7 and those of the straight pipe in Figure 9. As the spike is present in both sets of results it must be attributed to a loss in the straight pipe (and not the muffler). Although the exact cause of this spike was not determined in this investigation, it is shown in Figure 10 that it does not significantly affect the corrected muffler results.

VERIFICATION OF TECHNIQUE

The as measured transmission loss given in Figure 7 was corrected for the conduit attenuation using the curve shown in Figure 9. The result is shown in Figure 10. To validate the experimental measurement technique, the results were compared with those obtained analytically. An analytical expression for the transmission loss of a simple expansion chamber muffler is derived in the Appendix. Using equations (A15) and (A16), the transmission loss of a muffler with an area expansion ratio of 12.13 and an expansion length of 100 mm is presented in Figure 10. Figure 10 shows very good agreement between the theoretical and experimental results.

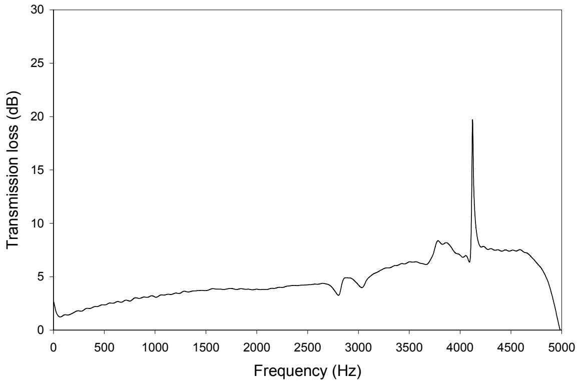

Figure 9. Transmission loss of the straight-through plastic conduit pipe of length 3.41 m.

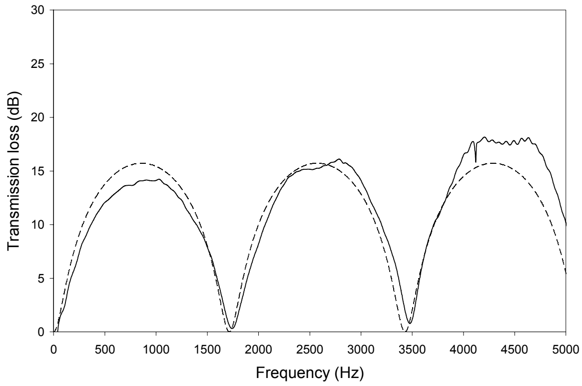

Figure 10. Sound transmission loss of the expansion chamber muffler after being corrected for the conduit attenuation, analytical result (dashed line), experimental result (solid line).

THE EFFECT OF MEAN FLOW ON THE SOUND TRANSMISSION LOSS

When mean flow is passed through the system, the microphones experience pressure fluctuations associated with the turbulent flow. Thus the transient acoustic pressure time histories become contaminated with noise. Since the flow induced pressures are random and not correlated with the acoustic transients generated by the horn drivers, it is possible to significantly remove their effect by averaging the recorded time histories. The time origin for each time history was defined with respect to a fixed point on the time history recorded by the inlet (or upstream) microphone. In the presence of mean flow, one hundred samples were taken for averaging.

The sound transmission loss of the straight-through plastic conduit was initially measured by this technique for four different mean flow velocities, corresponding to 1 m/s, 3 m/s, 5 m/s and 10 m/s. The results are given in Figure 11. Comparison of the results for the four different mean flow velocities given in Figure 11 (a) to (d) with each other, and with the transmission loss of the straight-through pipe given in Figure 9, shows that these mean flow velocities have little effect on the conduit attenuation.

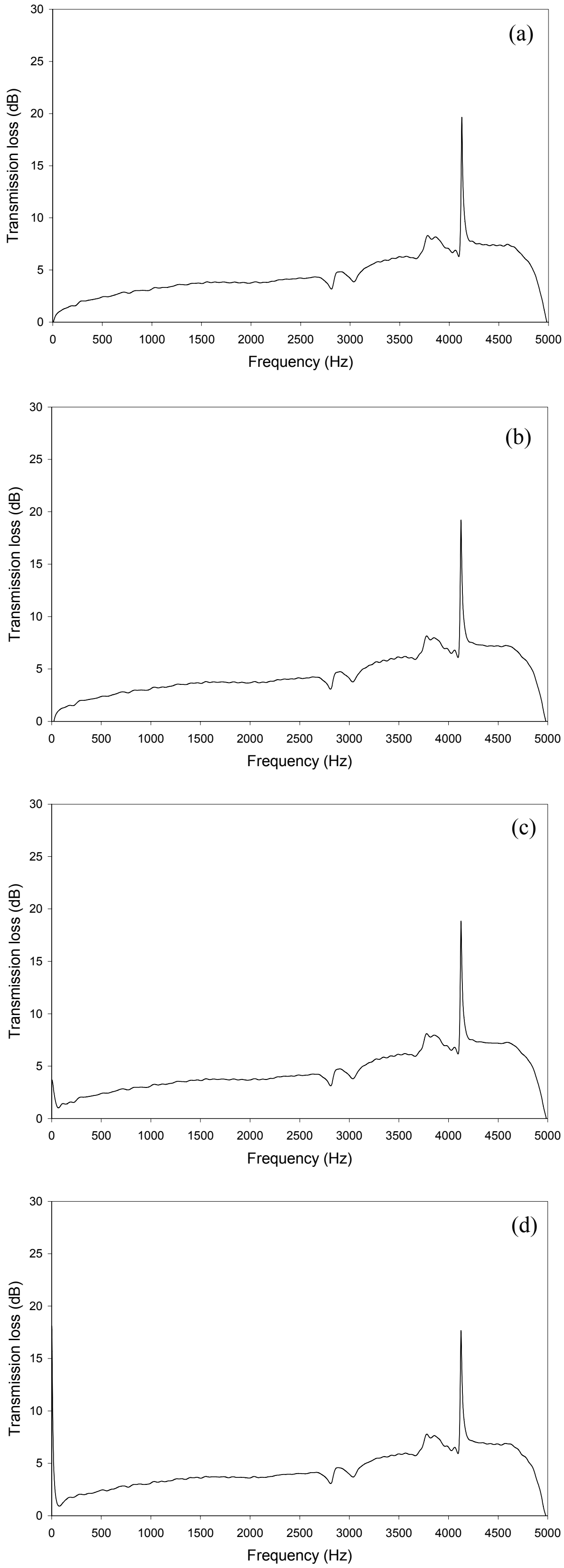

Figure 11. Conduit attenuation between the inlet (or upstream) microphone and the outlet (or downstream) microphone at mean flow velocities of 1 m/s (a), 3 m/s (b), 5 m/s (c) and 10 m/s (d).

A piece of conduit was then replaced with the aluminium expansion chamber muffler to give the same length from the upsteam to downstream microphones. Figure 12 shows the as measured sound transmission losses of the muffler at the same mean flow velocities and Figure 13 shows the corrected sound transmission loss curves. The conclusion which can be drawn from Figure 13 is that, in line with conventional wisdom, these relatively low mean flow velocities have very little effect on the sound transmission loss.

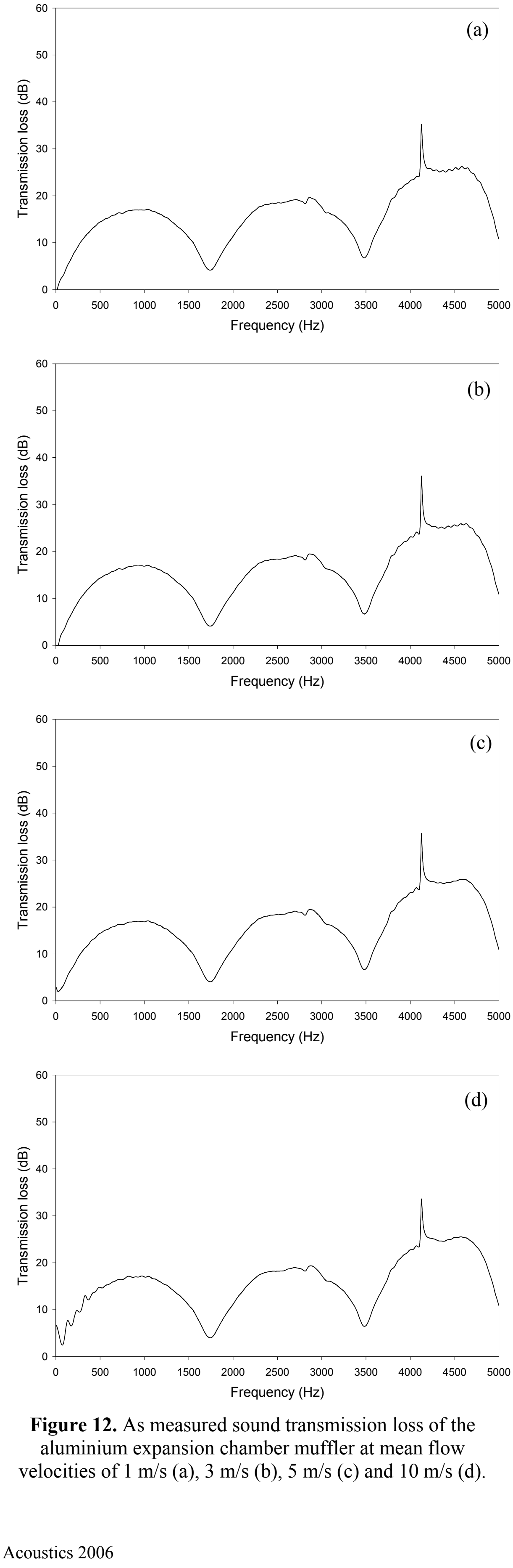

Figure 12. As measured sound transmission loss of the aluminium expansion chamber muffler at mean flow velocities of 1 m/s (a), 3 m/s (b), 5 m/s (c) and 10 m/s (d).

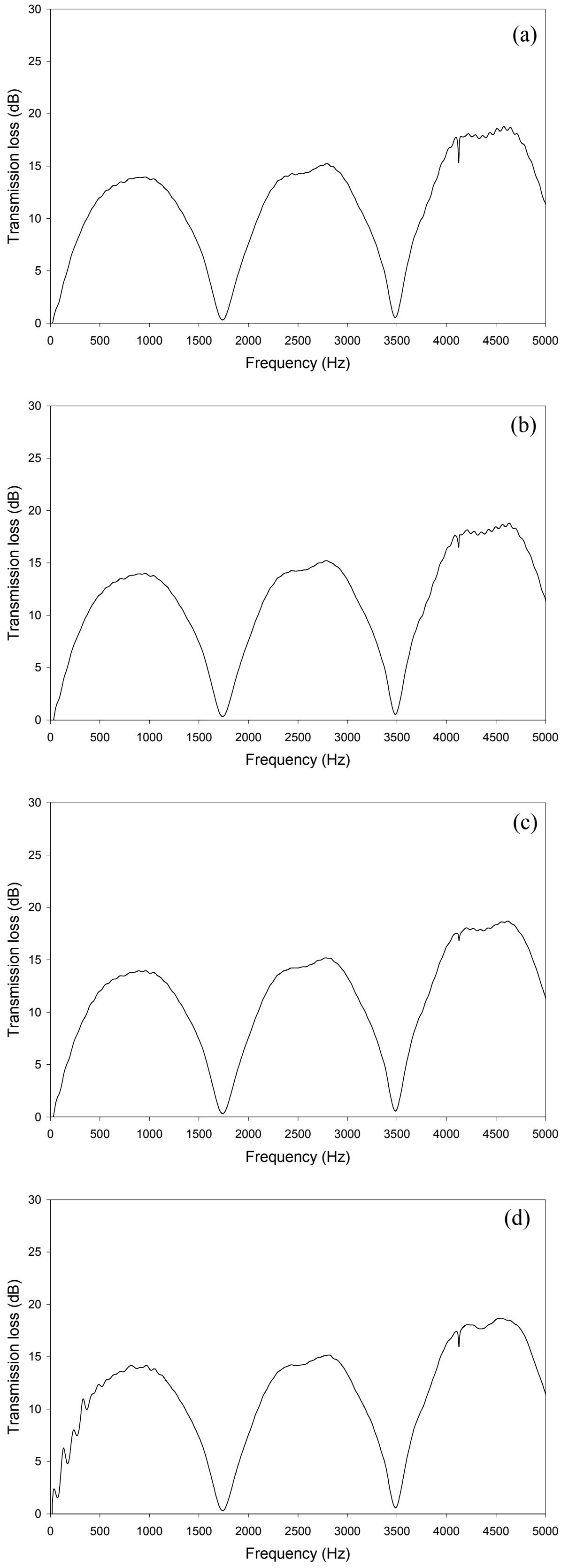

Figure 13. Corrected sound transmission loss of the aluminium expansion chamber muffler at mean flow velocities of 1 m/s (a), 3 m/s (b), 5 m/s (c) and 10 m/s (d).

THE EFFECT OF COMPLIANT BOUNDARIES ON THE SOUND TRANSMISSION LOSS

The expansion chamber muffler was constructed such that one of its walls could be removed and replaced by walls of different materials. In this preliminary study, plastic, paper and cardboard walls were used and these were characterised by their mass per unit area. Figure 14 shows a photograph of the expansion chamber muffler with its aluminium lid replaced by plastic. In view of the previously noted findings that mean flow has little effect, the measurements were conducted with no flow. However, the time history averaging technique was still used and the conduit sound attenuation was subtracted from the as measured sound transmission loss of the muffler.

The results for walls with different mass per unit areas are shown in Figures 15 to 17. The high transmisssion loss at low frequencies in these results is mainly attributed to sound leaking from the muffler rather than being attenuated. However, wall compliance does have a significant effect on the sound transmission loss of the muffler and must be considered in the modelling of actual mufflers.

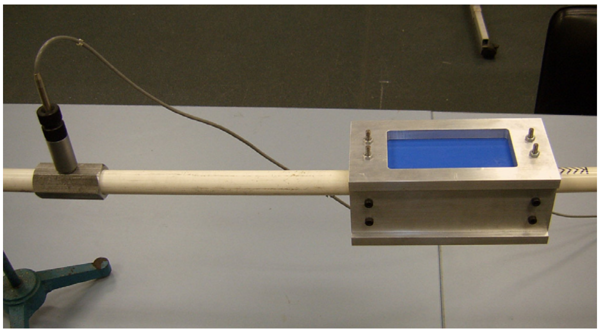

Figure 14. Photograph of the aluminium expansion chamber muffler with a plastic wall, and the downstream microphone.

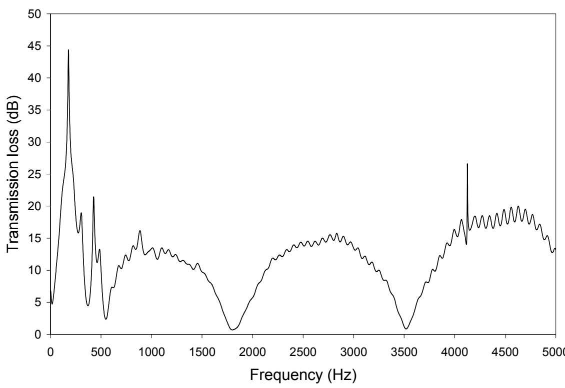

Figure 15. Corrected sound transmission loss of the expansion chamber muffler using cardboard.

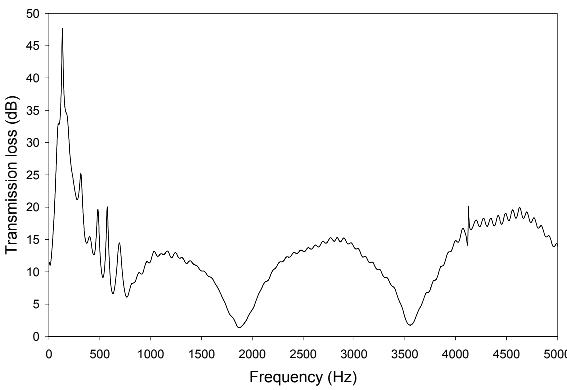

Figure 16. Corrected sound transmission loss of the expansion chamber muffler using paper.

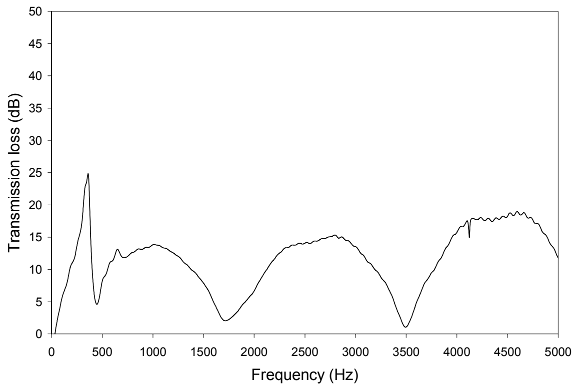

Figure 17. Corrected sound transmission loss of the expansion chamber muffler using plastic.

CONCLUSIONS

A method for measuring the acoustic performance of small mufflers is described in this paper. The measurement method is based on attaching long pipes to the inlet and outlet of the muffler. Horn drivers located along the pipe attached to the inlet were used to generate an acoustic pulse which propagated through the muffler and attached pipes. Microphones in the inlet and outlet pipes were used to sense the acoustic pressures of these pulses. The attenuation in the straight-through plastic pipe was subtraced from the total attenuation measured with the muffler in place to give the attenuation due to the muffler alone. The measured result was compared with an analytical result in which the validity of the measurement technique was demonstrated.

To measure the effect of mean flow on the acoustic performance of the muffler, a fan was used to force a flow through these pipes and the muffler. The horn drivers were repeatedly pulsed and the pressure time histories sensed by the microphones were recorded at precise times relative to the horn driver pulses, and then averaged to reduce the effect of the flow induced pressures. Low mean flow velocities were shown to have very little effect on the sound transmission loss of the muffler. However, by replacing one of the walls of the muffler with walls of different materials, wall compliance was shown to have a significant effect on the sound transmission loss of the muffler at low frequencies, though there was little effect on the high frequency transmission loss characteristics. Future work will investigate the acoustic performance of small, irregularly shaped mufflers used with sleep apnea assisted breathing devices.

REFERENCES

Craggs, A. 1977 “A finite element method for modelling dissipative mufflers with a locally reactive lining”, *Journal of Sound and Vibration* **54**, 285-296.

Ji Zhenlin, Ma Qiang and Zhang Zhihua 2003 “Application of the boundary element method to predicting acoustic performance of expansion chamber mufflers with mean flow”, *Journal of Sound and Vibration* **173**, 57-71.

Lee, C.-M. and Wang, Y. S. 2006 “A prediction method of the acoustic properties of multilayered noise control materials in standing wave-duct systems”, *Journal of Sound and Vibration* **298**, 350-365.

Munjal, M.L. 1997 “Plane wave analysis of side inlet/outlet chamber mufflers with mean flow”, *Applied Acoustics* (technical note) **52**,165-175.

Peat, K.S. 1988 “The acoustic impedance at discontinuities of ducts in the presence of a mean flow”, *Journal of Sound and Vibration* **127**, 123-132.

Song, B.H. and Bolton, J.S. 2000 “A transfer-matrix approach for estimating the characteristic impedance and wave numbers of limp and rigid porous materials", *Journal of the Acoustical Society of America* **107**, 1131-1152.

APPENDIX A

Sound transmission loss for a simple expansion chamber reactive muffler

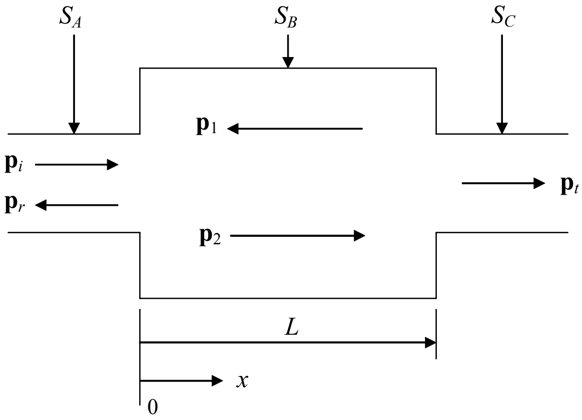

Figure A1. A simple expansion chamber reactive muffler.

An expression for the sound transmission loss, *TL*, of the simple expansion chamber reactive muffler shown in Figure A1 is derived. The complex representation of the acoustic pressures associated with the positive and negative travelling one dimensional harmonic plane acoustic waves as shown in the diagram, of angular frequency ω and wave number $k = \omega / c$, are given by the following equations:

$$\mathbf{p}_i = \mathbf{P}_i e^{j(\omega t - kx)} \quad \text{(A1)}$$

$$\mathbf{p}_r = \mathbf{P}_r e^{j(\omega t + kx)} \quad \text{(A2)}$$

$$\mathbf{p}_1 = \mathbf{P}_1 e^{j(\omega t - kx)} \quad \text{(A3)}$$

$$\mathbf{p}_2 = \mathbf{P}_2 e^{j(\omega t + kx)} \quad \text{(A4)}$$

$$\mathbf{p}_t = \mathbf{P}_t e^{j(\omega t - kx)} \quad \text{(A5)}$$

where the amplitude and phase information has been grouped as $\mathbf{P} = Pe^{j\phi}$. At the change of section at $x = L$, equality of acoustic pressure and equality of volume velocity lead to:

$$\mathbf{P}_1 e^{-jkL} + \mathbf{P}_2 e^{jkL} = \mathbf{P}_t e^{-jkL} \quad \text{(A6)}$$

$$S_B \mathbf{P}_1 e^{-jkL} - S_B \mathbf{P}_2 e^{jkL} = S_C \mathbf{P}_t e^{-jkL} \quad \text{(A7)}$$

These two equations can be used to determine $\mathbf{P}_1$ and $\mathbf{P}_2$ in terms of $\mathbf{P}_t$.

$$\mathbf{P}_1 = \frac{1}{2}\left(1 + \frac{S_C}{S_B}\right)\mathbf{P}_t \quad \text{(A8)}$$

$$\mathbf{P}_2 = \frac{1}{2}\left(1 - \frac{S_C}{S_B}\right)\mathbf{P}_t e^{-j2kL} \quad \text{(A9)}$$

At the change of section at $x = 0$, equality of pressure and equality of volume velocity lead to:

$$\mathbf{P}_i + \mathbf{P}_r = \mathbf{P}_1 + \mathbf{P}_2 \quad \text{(A10)}$$

$$S_A \mathbf{P}_i - S_A \mathbf{P}_r = S_B \mathbf{P}_1 - S_B \mathbf{P}_2 \quad \text{(A11)}$$

Equations (A10) and (A11) can be used to express $\mathbf{P}_i$ in terms of $\mathbf{P}_1$ and $\mathbf{P}_2$.

$$\mathbf{P}_i = \frac{1}{2}\left(1 + \frac{S_B}{S_A}\right)\mathbf{P}_1 + \frac{1}{2}\left(1 - \frac{S_B}{S_A}\right)\mathbf{P}_2 \quad \text{(A12)}$$

Substitution of the expressions for $\mathbf{P}_1$ and $\mathbf{P}_2$ given by equations (A8) and (A9) into (A12) and rearranging allows $\mathbf{P}_i$ and $\mathbf{P}_t$ to be related by:

$$\frac{\mathbf{P}_t}{\mathbf{P}_i} = \frac{4e^{jkL}}{\left(1 + \frac{S_C}{S_A} + \frac{S_C}{S_B} + \frac{S_B}{S_A}\right)e^{jkL} + \left(1 + \frac{S_C}{S_A} - \frac{S_C}{S_B} - \frac{S_B}{S_A}\right)e^{-jkL}} \quad \text{(A13)}$$

The sound transmission coefficient α_t is defined as the ratio of the intensity of the transmitted wave to the intensity of the incident wave. Since the only medium is air, it is given by:

$$\alpha_t = \frac{I_t}{I_i} = \frac{\left|\mathbf{P}_t^2\right|}{\left|\mathbf{P}_i^2\right|} \quad \text{(A14)}$$

The sound power transmission coefficient T_t is defined as the ratio of the power of the transmitted wave to the power of the incident wave. In this study, since the areas associated with the incident and transmitted waves are equal (that is, $S_C = S_A$), the sound power transmission coefficient T_t is equal to the sound transmission coefficient α_t . Hence:

$$T_t = \alpha_t = \frac{4}{A^2 \cos^2 kL + B^2 \sin^2 kL} \quad \text{(A15)}$$

where $A = 1 + \frac{S_C}{S_A}$ and $B = \frac{S_C}{S_B} + \frac{S_B}{S_A}$. The sound transmission loss *TL* can then be found from:

$$TL = -10\log_{10} T_t \quad \text{(A16)}$$

It should be noted that the previous analysis is only valid if the cross-sectional dimensions associated with S_A and S_B are less than about one half a wavelength, so that the plane wave assumption is reasonably valid.