FORUM ACUSTICUM

The effect of higher order modes on the performance of large diameter dissipative silencers

Paul T. Williams
AAF Limited, Cramlington, Northumberland, NE23 8AF, UK

Ray Kirby
Brunel University, Uxbridge, UB8 3PH, UK

Summary
Within the gas turbine industry dissipative silencers are regularly used to reduce broadband noise within duct systems. Silencer performance is normally quantified using the insertion loss due to a plane wave incident sound field. However for larger silencers the widths of these duct systems are large enough to allow higher order modes to propagate over much of the frequency range of interest (31 - 8kHz octave bands), which may have a significant effect upon silencer performance that is not normally accounted for.

The performance of dissipative parallel baffle and bar silencers in the presence of different types of incident sound field is investigated through a numerical model which uses the finite element method and point collocation to predict insertion loss. Excitation of the silencer using an equal modal energy density sound field is found to have a large effect upon performance compared to plane wave excitation. Increases to insertion loss are predicted at high frequencies for the geometries modelled and it is found that plane wave predictions do not necessarily give the worst case performance.

1. Introduction

Noise from industrial equipment is regularly channelled through ducts, such as HVAC systems and gas turbine exhausts. Within these systems the noise is predominantly broadband and noise control is required between 22Hz and 11220Hz. The large duct dimensions, high temperatures of up to $650°C$ and complex sound sources make it likely that higher order modes will propagate through the system. Although modern prediction software can include the propagation of multiple modes within the silencer and outlet duct, performance is regularly calculated using either the least attenuated mode or with plane wave excitation at the inlet. This is primarily due to experimental data commonly being measured with plane wave excitation in accordance with BS EN ISO 7235:2009 [1] and no data being available on the modal composition of gas turbines sources.

The performance of dissipative silencers is dependant upon frequency, with high efficiency at mid frequencies and lower efficiency at both upper and lower ends of the spectrum. Further to this it is difficult to increase the performance at low frequencies and high frequencies simultaneously, as each requires conflicting silencer properties, leading to over-engineered silencer designs. If it can be shown that a higher performance can be gained at high frequencies through the use of incident higher order modes in prediction models and that this is valid in reality, it is probable that silencer length and hence cost and weight could be reduced.

Mechel's [2, 3] development of a FEM numerical model for the prediction of insertion loss for parallel baffle silencers included methods of calculating the modal amplitudes of an incident sound field given assumptions upon its form, however the presented results were constrained to analysis of the effect of single mode excitations. Cummings and Sormaz's [4] study of a system consisting of multiple baffles showed that if a model was simplified to predictions of a single baffle, as in the plane wave simplification of Mechel, then modes may be omitted from the analysis. Although symmetry shows that this is unimportant under plane wave excitation, when higher order modes are present in the silencer inlet it is possible that these omitted modes include the current least attenuated mode, which may lead to overpredicted silencer performance if incorrectly modelled. The insertion loss of a parallel baffle silencer with multi-modal excitation was investigated by Kirby [5] and Kirby and Lawrie [6] finding both increased and decreased performance compared to plane wave predictions upwards from 350Hz. These differences in behaviour are likely due to differences in the chosen geometry of the silencers but is not explored further by the authors.

Within this paper the effect of a multi-modal sound field upon parallel baffle and bar silencers will be explored, along with the importance of modelling the full system and the possible problems with neglecting higher order modes within the inlet.

2. Theory

The dissipative parallel baffle and bar silencers studied within this paper all have rectangular cross-sections which are uniform along their length. Porous material is held within baffles separated from the fluid within the airway by a perforated plate with a high open area. A diagram of these silencers are shown within Figure 1. The duct system consists of an inlet and outlet duct of infinite length attached to a silencer of length l where the outlet is terminated anechoically. The numerical methods used within this paper has been described previously [5] and will not be discussed in detail.

The pressure within the inlet, silencer and outlet are represented as a sum of the modal pressures,

$$p_I(x,y,z) = \sum_{m=1}^{\infty} F_m \Psi_m(y,z) e^{-i\nu_m k_0 x} + \sum_{m=1}^{\infty} A_m \Psi_m(y,z) e^{i\nu_m k_0 x} \quad (1)$$

$$p_S(x,y,z) = \sum_{m=1}^{\infty} B_m \Phi_m(y,z) e^{-i\mu_m k_0 x} + \sum_{m=1}^{\infty} C_m \Phi_m(y,z) e^{i\mu_m k_0 x} \quad (2)$$

$$p_O(x,y,z) = \sum_{m=1}^{\infty} D_m \Psi_m(y,z) e^{-i\nu_m k_0 x}, \quad (3)$$

where k_0 is the wavenumber, F_m are the known modal amplitudes of the incident sound field, and A_m, B_m, C_m and D_m are modal amplitudes, Ψ_m and Φ_m are the transverse duct eigenvectors, and ν_m and μ_m the normalised axial wavenumbers. The modal amplitudes of the incident sound field used in this report are calculated using

$$\left|\frac{F_m}{p_0}\right| = \begin{cases} 1, & m = 1 \\ 0, & m > 1 \end{cases} \quad (4)$$

Table I. Dimensions of the parallel baffle silencer module

t(m)	a(m)	h(m)	l(m)
0.2	0.2	0.3	1.0

Table II. Dimensions of the bar silencer module.

t_z(m)	t_y(m)	a_z(m)	a_y(m)	l(m)
0.28	0.21	0.12	0.19	1.0

and

$$\left|\frac{F_m}{p_0}\right| = \sqrt{\frac{I_0}{\sum_n \mathbb{R}[\nu_n] I_m}} \quad (5)$$

for plane wave and equal modal energy density (EMED) respectively, where $I_m = \iint |\Psi_m(y,z)|^2 dydz$ and p_0 is a reference pressure.

By carrying out a FEM eigenvalue analysis of the cross-sections of the system and using the point collocation technique to match over axial discontinuities the modal amplitudes and axial wavenumbers are found and used to calculate the insertion loss using

$$IL = -10\log_{10}\left[\frac{\sum_m I_m |D_m|^2 \mathbb{R}[\nu_m]}{\sum_m I_m |F_m|^2 \mathbb{R}[\nu_m]}\right]. \quad (6)$$

3. Results

The two silencer cross-sections investigated in this paper are the parallel baffle and bar silencer, see Figure 1. The parallel baffle silencer is a common silencer design but has previously been shown to be less efficient than bar silencers at mid and high frequencies [7, 8]. To reduce computational requirements for the prediction only a single baffle or bar of each silencer has been simulated, unless stated otherwise. Both silencers have an open area of 50% and use the same material properties and perforated plate with dimensions shown in Tables I-II.

Figure 2 shows the insertion loss for a parallel baffle silencer calculated using a 1-dimensional mesh of the duct cross-section, reducing the geometry to a horizontal line across the silencer cross-section passing through the baffles and airway. By reducing the problem from 2-dimensions to 1-dimension the modes in the vertical direction, z-axis, are removed from the analysis leaving only the horizontal, y-axis, modes. This simplification is acceptable for predictions of silencers with negligible heights and for exploring the effect of higher order modes without the influence of the vertical duct modes and will be justified later. The performance under a plane wave sound field shows the expected behaviour for a parallel baffle silencer with a decreased performance at high frequencies compared to mid frequencies. When excited by an EMED sound field the spectrum changes showing a much greater performance to that of the plane

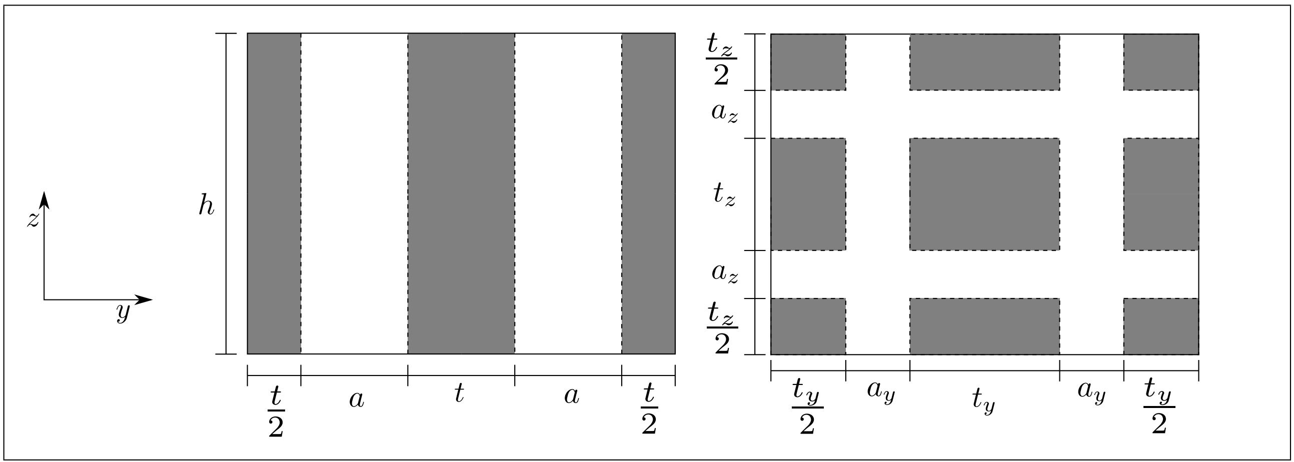

Figure 1. Diagram of the cross section of a parallel baffle silencer, left, and a bar silencer, right.

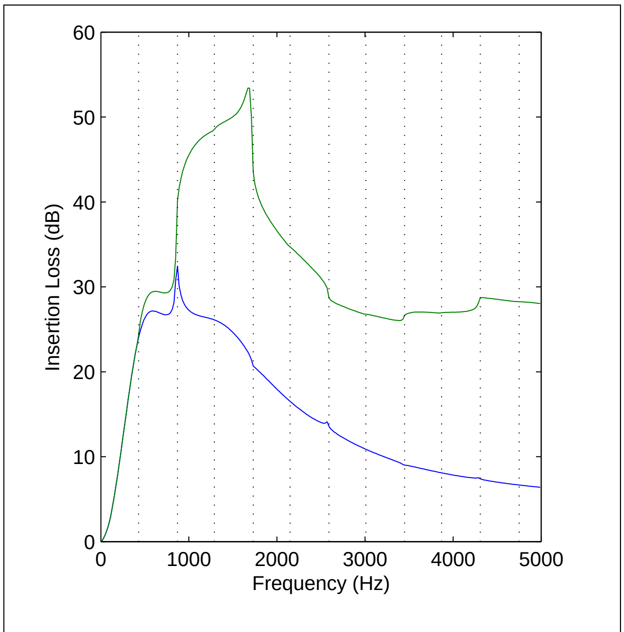

Figure 2. Insertion Loss of the parallel baffle silencer excited by a plane wave incident sound field (blue) and EMED incident sound field (green). Vertical lines denote the frequencies at which a new mode can propagate in the inlet duct.

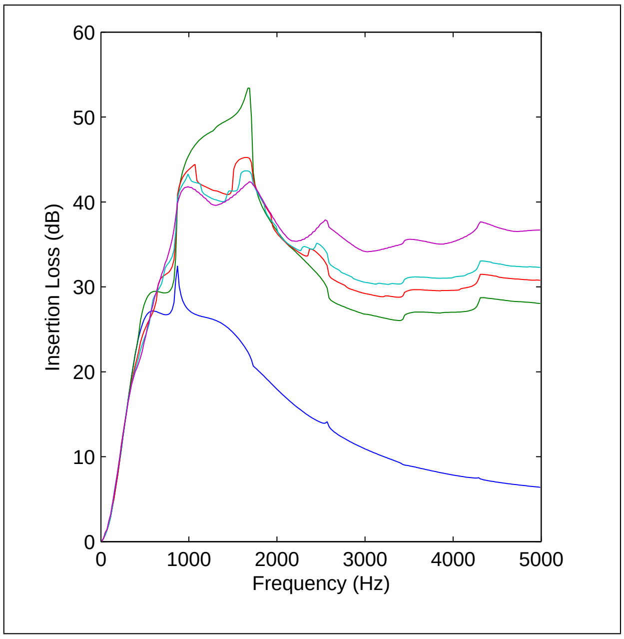

Figure 3. Insertion Loss for silencer 1 excited by a plane wave incident sound field (blue) and EMED incident sound field for 1 baffle (green), 2 baffles (red), 3 baffles (cyan) and 10 baffles (purple).

wave prediction at mid to high frequencies. The insertion loss of the EMED predictions diverge from the plane wave predictions at the first cut-on mode of the inlet duct at 430Hz. Analysis of the silencer's modal amplitudes, B_m and C_m, shows that the anti-symmetric silencer modes are not excited unless there is an anti-symmetric mode present in the inlet and therefore below the first cut-on mode and for plane wave analyses the anti-symmetric modes do not carry energy. The largest changes to insertion loss are shown at the cut-on frequencies of symmetric modes, such as at 859Hz and 1717Hz, with new anti-symmetric inlet duct modes providing only small changes.

From Mechel's discussion upon symmetry [3] it is expected that the number of baffle modules simulated makes no difference to the predicted insertion loss when using a plane wave sound field, but that when using a multimodal field the entire cross-section must be included to prevent modes being missed from the eigenanalysis. This is demonstrated within Figure 3 where, as the number of baffles is increased, high frequency attenuation is seen to improve. On the other hand an interesting result arises where the insertion loss around 450Hz of silencers excited with higher order modes decreases below that of the plane wave prediction. This occurs where the least attenuated mode corresponds to the first anti-symmetric mode showing

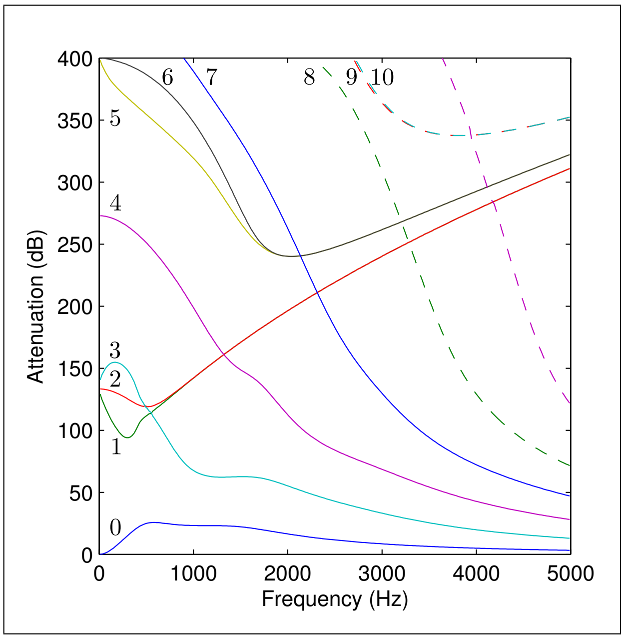

Figure 4. The least attenuated modes within the parallel baffle silencer.

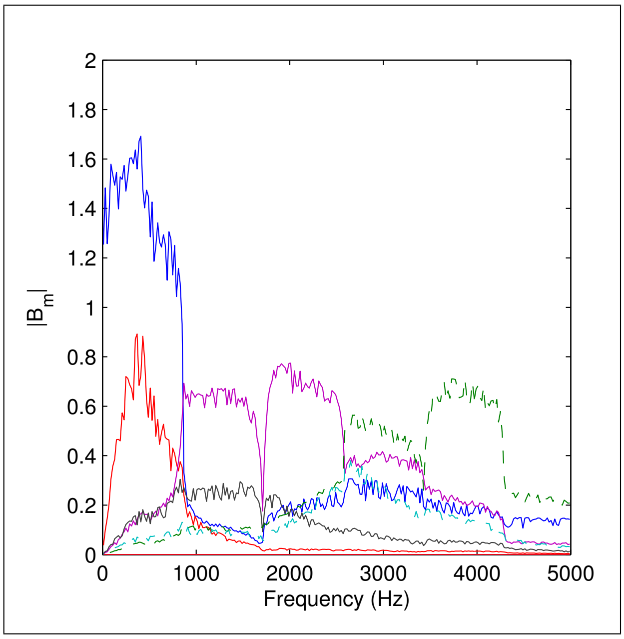

Figure 6. $|B_m|$ of the first six symmetric modes for the parallel baffle silencer under EMED excitation.

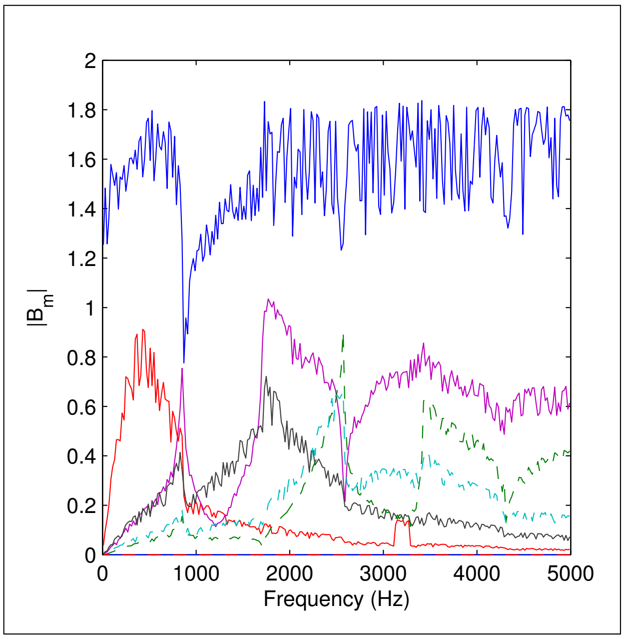

Figure 5. $|B_m|$ of the first six symmetric modes for the parallel baffle silencer under plane wave excitation.

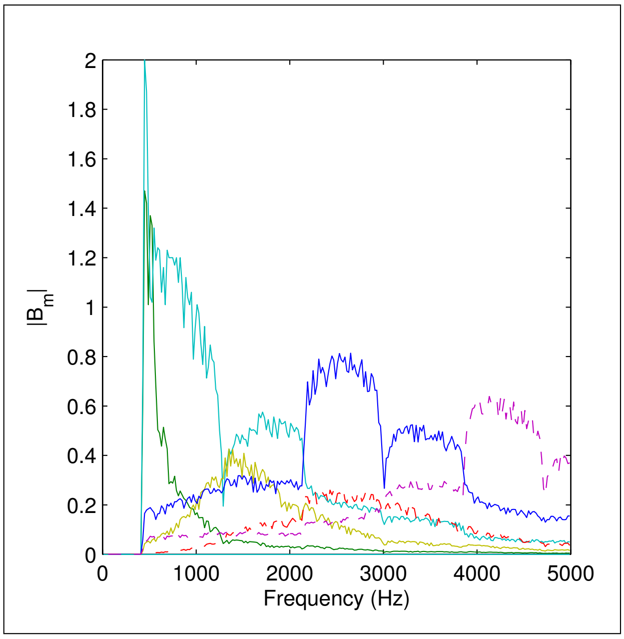

Figure 7. $|B_m|$ of the first six anti-symmetric modes for the parallel baffle silencer under EMED excitation.

that a plane wave sound field does not necessarily excite the least attenuated mode or provide the worst case silencer performance at all frequencies, [4]. While the number of higher order modes within the duct is increased by widening the inlet duct and the divergence between plane wave and EMED predictions continues to occur at the frequency of the first cut-on mode, the characteristic large changes to insertion loss remain close to the same frequencies, suggesting that the addition of higher order modes is not the only influence upon insertion loss.

Figure 4 shows the attenuation of symmetric (even numbered) and anti-symmetric (odd numbered) modes in the silencer. Comparing these modal attenuations to the modal amplitudes for the parallel baffle silencer under plane wave excitation, see Figure 5, shows that the least attenuated mode dominates the performance by coupling the most strongly to the inlet sound field across all frequencies consistent with the lower silencer performance. The behaviour of the silencer under EMED excitation is seen to be much more complex with different modes coupling

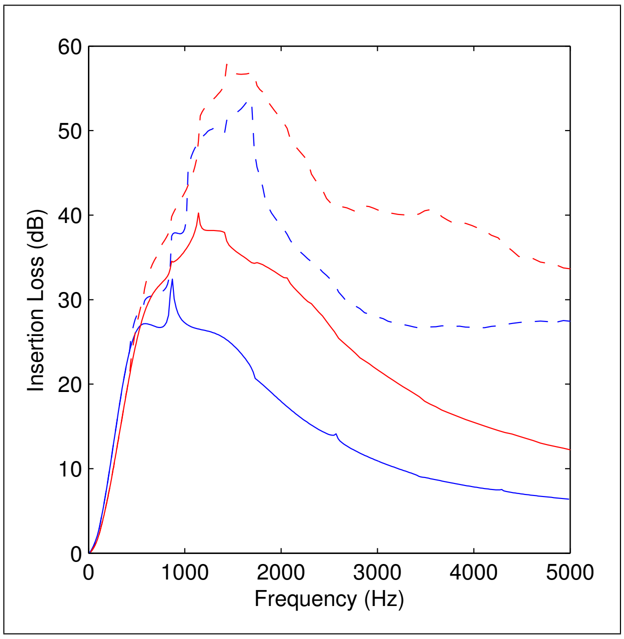

Figure 8. Insertion loss of the parallel baffle silencer (blue) and bar silencer (red) excited by plane wave (solid) and EMED (dashed) excitation.

most strongly to the incident field within different frequency ranges, see Figures 6-7. These features correspond to those found in Figure 2, such as the cut-on of the first anti-symmetric mode at 430Hz. In this case, and also with two baffles within the duct, the amplitude of the first silencer mode is seen to decrease sharply at 860Hz combined with an increase in the amplitude of the much more highly attenuated third symmetric mode following the increased insertion loss in Figure 2.

Figure 8 compares predictions for a parallel baffle and bar silencer, both computed using a 2-dimensional mesh, excited by plane wave and EMED sound fields. The inlet duct cross-section is the same for both silencers and therefore the incident fields are the same. As the parallel baffle silencer has now been simulated using a 2-dimensional mesh of the cross-section, vertical duct modes are now included in the analysis. The inclusion of vertical modes has made only small changes to the insertion loss of the parallel baffle silencer across the peak when compared to the previous results which were calculated using a 1-dimensional mesh, justifying the simplified calculations above. Previous comparisons between parallel baffle silencers and bar silencers have shown the bar silencer to be the more efficient design and this is shown here with both silencers under EMED excitation. The bar silencer maintains its higher insertion loss although the difference between both silencers is smaller at the peak.

4. Conclusions

Using finite element methods the effect of exciting the inlet to a silencer with a equal modal energy density sound field has been explored. It has been shown that with this particular excitation a much greater performance is predicted at mid to high frequencies for both parallel baffle and bar type dissipative silencers. If higher order modes within the inlet were to be accounted for during the design process for dissipative silencers it is probable that silencer lengths could be reduced by altering designs so as to target low frequency performance where in the past high frequency noise was a limiting factor. As with previous studies the increased efficiency of bar silencers over parallel baffle silencers has again been noted.

Predictions also showed that, due to the presence of both symmetric and anti-symmetric modes within real systems, the plane wave excitation may not necessarily give the most conservative insertion loss for all geometries.

It is clear that in order to have a better understanding of how silencers perform in the exhaust of a gas turbine the modal composition of the noise must be known.

References

[1] BSENISO7235:2009: Acoustics - Laboratory measurement procedures for ducted silencers and air-terminal units - Insertion loss, flow noise and total pressure loss (2009).

[2] F.P. Mechel: Theory of baffle-type silencers. Acustica **70** (1990) 93-111.

[3] F.P. Mechel: Numerical results to the theory of baffle-type silencers. Acustica **72** (1990) 7-20.

[4] A. Cummings and N. Sormaz: Acoustic attenuation in dissipative splitter silencers containing mean fluid flow. Journal of Sound and Vibration **168** (1993) 209-227.

[5] R. Kirby: The influence of baffle fairings on the acoustic performance of rectangular splitter silencers. J. Acoust. Soc. Am. **118** (2005) 2302-2312.

[6] R. Kirby and J. B. Lawrie: A point collocation approach to modelling large dissipative silencers. Journal of Sound and Vibration **286** (2005) 313-339.

[7] R. Kirby, P.T. Williams and J. Hill: A comparison between the performance of different silencer designs for gas turbine exhaust systems. Proc. Acoustics 2012, 3199-3204.

[8] R. Kirby, P.T. Williams and J. Hill: A three dimensional investigation into the acoustic performance of dissipative silencers. J. Acoust. Soc. Am. **135** (2014) 2727-2737.