Hybrid dissipative/reactive silencer predictions with comparison to measurement

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Summary

Gas Turbines predominantly generate a broadband noise spectrum, although strong tones are also present, often at relatively low frequencies. Dissipative silencers are commonly used to reduce noise within gas turbine systems, however they are inefficient at removing low-frequency noise and this leads to silencers which are overdesigned at higher frequencies. Therefore, by introducing new techniques designed specifically to target low-frequencies it is proposed that the overall cost of the silencer may be decreased. Reactive elements are known to be successful at targeting low frequencies but their non-acoustic limitations, including problems with minimising pressure drop, have traditionally prevented their use in gas turbine systems.

This paper presents a new silencer design that combines dissipative and reactive silencer elements in order to target low frequency tonal noise, whilst at the same time minimising static pressure drop. This so-called hybrid silencer is designed using advanced finite element modelling techniques and new prototype designs are then tested in the laboratory. Comparison between prediction and experiment shows good agreement over a wide frequency range, which demonstrates the validity of the modelling approach. Results demonstrate that attaching a carefully designed reactive element to a dissipative element can deliver a large increase in performance at low frequencies. This improvement in performance is characterised by a resonance "peak", which is designed to cover an octave band surrounding the target frequency in order to accommodate uncertainties in the noise output of a gas turbine. Furthermore, it is found that additional peaks occur at higher harmonics, which further increases the performance of the hybrid silencer beyond the plane wave region of the inlet duct, delivering additional and significant improvements in performance for the new hybrid silencer when compared to traditional dissipative designs and previous hybrid silencer concepts.

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1. Introduction

The noise produced by a gas turbine generally has a broadband spectrum which requires attenuation in the 31Hz to 8kHz octave bands. Dissipative silencers are commonly used for this purpose due to their broadband insertion loss spectrum. These silencers provide a large insertion loss at mid-frequencies (250Hz-2kHz) but are less effective in the low frequency range (<250Hz) with an IL which tends to zero with decreasing frequency [1]. In practice this can lead to difficulties when controlling noise with more power in the low frequency octave bands and causes silencers to be overdesigned at mid-frequencies in order to meet the low frequency noise limits. Reactive silencers, such as the quarter wave and Helmholtz resonator, are able to provide a large insertion loss over a small range of frequencies, however this narrow bandwidth makes the reactive silencer unsuitable for use on its own. By combining these two silencer types it is possible to design a silencer with both broadband and tuned low frequency narrow band components of IL.

This concept is applied within the automotive industry by combining expansion chambers, helmholtz resonators and dissipative lined ducts in exhaust muf-

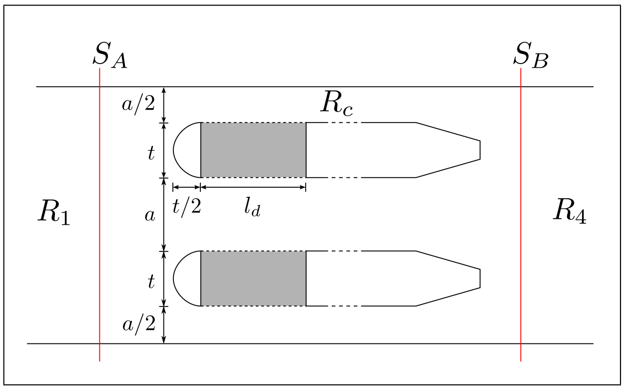

Figure 1. Diagram showing a plane along the duct system. Dashed lines represent perforated plates while shaded areas represent porous material.

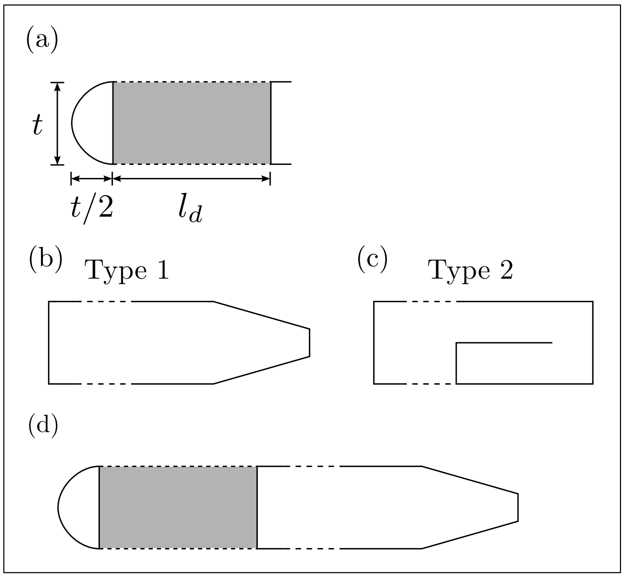

Figure 2. Diagram showing the shape along the axis of (a) a dissipative element, (b) a type 1 reactive element, (c) a type 2 reactive element and (d) a hybrid silencer baffle formed by combining a reactive and dissipative element.

flers [2, 3]. Quarter wave resonators housed in the duct walls and dissipative lined duct sections are used similarly to reduce the noise of diesel engines on board ships [4]. Although these techniques are effective, they are unsuitable for the large duct widths (>1m) present within gas turbine exhaust systems due to the much lower cut-on frequency of higher order modes in the duct. This paper will explore a new hybrid silencer designed to target low frequency tonal noise while effectively attenuating broadband emissions.

2. Silencer Design

The hybrid silencer concept explored within this paper combines innovative silencing techniques allowing for one silencer to produce a broad band attenuation spectra whilst also targeting low frequency tonal content. This is achieved through the use of dissipative and reactive silencer elements, see Figure 1. By containing these elements within the same baffle, traditional silencer cross-sections may be used whilst maintaining the external baffle profile seen by the air flow. The air flow profile and static pressure drop are therefore unchanged avoiding the problems that would be seen by other reactive solutions.

Within this paper the hybrid silencer is modular and follows the common design for parallel baffle silencers. The silencer therefore consists of a number of baffles of thickness t placed across the duct separated by an airway with width a, see Figure 1. Here the dissipative element forms the front of the baffle and the reactive element forms the rear. The dissipative elements contain a volume of porous material, separated from the airway by perforated plates with a length l_d and terminated at the front and rear by solid fairings.

The reactive elements operate using the principles of quarter wave resonators where the fundamental resonant frequency, f_r, can be found through

$$f_r = \frac{c}{2L}, \tag{1}$$

where c is speed of sound and L is the effective length of the silencer chamber. Unlike the dissipative element which is perforated along its entire length, the reactive element is sealed, isolating it from the airway, except for an opening on either side of the baffle. These openings are perforated to reduce airflow into the chamber, add damping to the system and minimise the effects of flow. Two types of reactive element are investigated, here named type 1 and type 2. The type 1 reactive element shows a silencer which begins to taper along its length allowing it to be fitted inside the evase of a baffle, see Figure 2. In dissipative silencers this evase is usually unperforated and is used only to improve the aerodynamic qualities of the baffle. By extending the reactive chamber into the evase this can be used to increase acoustic efficiency. The resonating chamber of the type 2 reactive element is bent back upon itself to increase the effective length of the chamber, see Figure 2. This increase in effective length comes at the expense of halving the width of the chamber and therefore decreasing the potential insertion loss.

3. Theory

A hybrid numerical model was developed combining finite element methods and mode matching techniques to predict the transmission loss of silencers with arbitrary geometry, see Figure 1. The system will be split into the inlet duct, region R_1, silencer, region R_c, and outlet duct, region R_4. The inlet and outlet regions are axially uniform, empty ducts with rigid walls. The pressure fields in regions R_1 and R_4 will be represented using a modal expansion of incident and reflected waves,

$$p_1 = \sum_{m=1}^{\infty} A_m \Phi_m(y,z) e^{-i\phi_m k_0 x} + \sum_{m=1}^{\infty} B_m \Phi_m(y,z) e^{i\phi_m k_0 x} \qquad (2)$$

$$p_4 = \sum_{m=1}^{\infty} C_m \Phi_m(y,z) e^{-i\phi_m k_0 x}, \qquad (3)$$

where p_1 is the pressure field in the inlet, p_4 is the pressure field in the outlet, A_m, B_m and C_m are modal amplitudes, Φ_m is the transverse eigenvector of the ducts, ϕ_m is the axial wavenumber of the ducts, $k_0 = 2\pi f/c$, f is the frequency and m is the mode number. Eigenvectors and eigenvalues are found by carrying out a finite element eigenvalue analysis of the cross-sections [1].

The propagation of sound through R_c is found through the wave equation using the Galerkin method starting from

$$\int_{R_c} N(\nabla p_c + k^2 p_c) dR_c = 0, \qquad (4)$$

where p_c is the pressure field within R_c. Region R_c is separated further into volumes of porous material, R_2, and air, R_3. Eq. (4) is expanded and boundary conditions applied for pressure and axial displacement across the perforated plate and normal acoustic velocity at the solid walls [1]. The properties of the porous material is calculated using the coefficients derived by Williams et al. [5] and the method of Kirby and Cummings [6]. At the planes separating R_1 and R_4 from R_c, continuity of acoustic velocity is enforced through eq. (4) while continuity of pressure is enforced separately.

Transmission loss is then calculated using

$$TL = -10 \log_{10} \left(\frac{\sum_m I_m |C_m|^2 \mathbb{R}[\phi_m]}{\sum_m I_m |A_m|^2 \mathbb{R}[\phi_m]} \right), \qquad (5)$$

where $I_m = \int |\Phi_m|^2 dS_A$.

The silencers in this paper are uniform along the z-axis, varying only along the x- and y-axis. This allows the z-axis to be removed from the analysis allowing for the silencer to be represented in only 2-dimensions, simplifying the model and reducing the size of the problem.

4. Experiment

Silencer performance was measured using the setup shown within Figure 3. The system consisted of an anechoic chamber connected to a reverberation room via a shared wall. This wall had a 0.8m by 0.6m hole cut through over which a 0.8m by 0.6m test duct was attached on the reverberation room side. Airflow

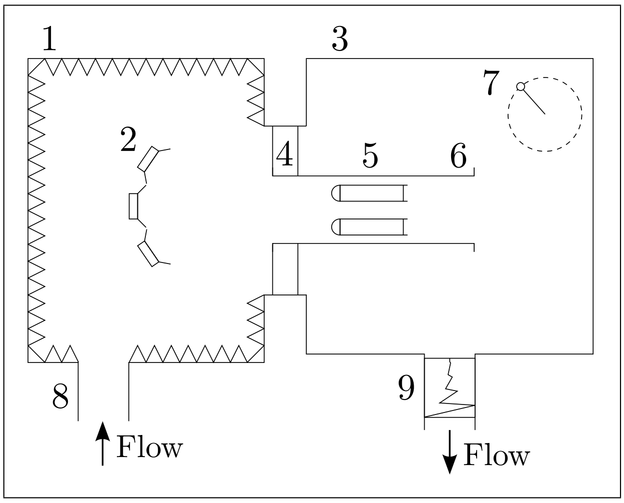

Figure 3. Diagram of equipment showing the anechoic chamber (1), the loudspeaker array (2), reverberation room(3), separating wall (4), silencer (5), test duct (6), microphone on rotating arm (7), flow inlet duct (8) and flow exit duct with silencer (9).

could therefore only travel between rooms by passing through this hole and along the test duct. The test duct was constructed from 1.5mm thick steel and had a length of 2.05m. Noise was generated in the anechoic chamber using five loudspeakers placed in an arc pointing at, and within 2m of, the entrance to the test duct. Sound pressure levels were measured in the reverberation room using a single microphone attached to a rotating arm.

Silencer insertion loss was measured using the common substitution method as detailed within ISO 7235 [7] using the equipment described above. The third octave band sound pressure levels were measured in the reverberation room, first with an empty duct and then with a silencer present within the duct. These measurements were then used to calculate the insertion loss of the silencer. Measurements were also taken in the presence of flow at speeds of 13ms^{-1} and 26ms^{-1}. In all cases the flow entered through the anechoic chamber and exited through the reverberation room so that sound propagated in the direction of flow.

For the purposes of this experiment the hybrid silencers had a modular design with the dissipative and reactive elements being separate objects. This allowed for the same set of dissipative elements to be used for all measurements giving a consistent baseline against which the performance of the reactive elements could be compared. Figure 2 illustrates the design of the dissipative baffle. At the entrance to the silencer the baffle fairings were semicircular to smooth airflow and reduce flow noise. A flange was placed at the opposite end of the baffle allowing for a reactive element to slot into position creating a hybrid silencer baffle. In this paper all silencers consist of two baffles placed across the duct.

Table I. Silencer parameters.

Silencer	t(m)	a(m)	l_d(m)	Reactive Type	f_r(Hz)
S1	0.2	0.2	0.9	-	-
S2	0.2	0.2	0.9	1	125
S3	0.2	0.2	0.9	1	500
S4	0.2	0.2	0.9	2	220
S5	0.2	0.2	0.9	2	255

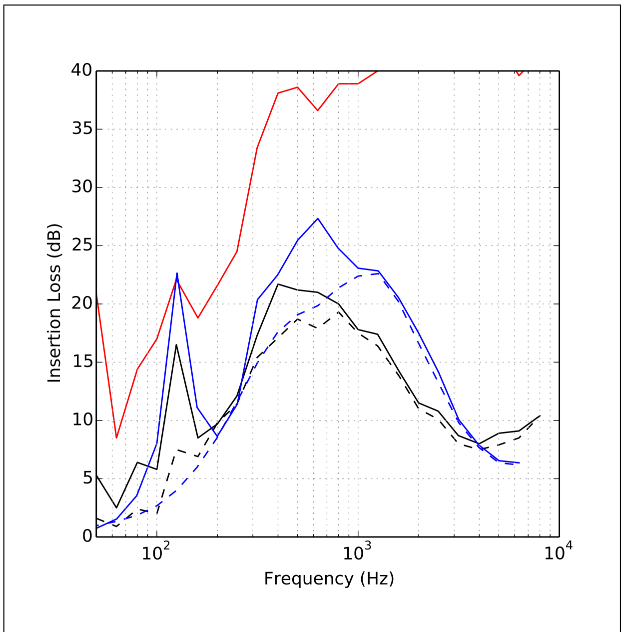

Figure 4. Insertion loss of S1 (Measured, – – –; Predicted, – – –) and S2 (Measured, ———; Predicted, ———). Limiting insertion loss, ———.

The dissipative elements were constructed entirely from 1.2mm thick steel which was perforated along the sides. Slabs of rock wool were held within the baffles but was separated from the perforated plate by a thin glass cloth to prevent fibres from leaving the baffle when subject to flow. These baffles were bolted to the duct floor and ceiling to strengthen the duct and to maintain the same position for the dissipative baffles through all tests. Reactive elements were constructed from 1.2mm thick steel with dimensions designed to target specific frequencies. Parameters for the hybrid silencers presented in this paper can be found within Table I.

5. Results and Discussion

The third octave band transmission loss predictions presented within this section have been calculated by averaging the transmission loss of 15 frequencies per third octave band.

Figure 4 illustrates the ability of the numerical model to predict the TL of the dissipative silencer, S1. Measurement and prediction show a good match up to the 800Hz third octave band with over prediction above the measured peak IL. Above 4000Hz the IL of S1 is expected to decrease, however here it is found to increase indicating an error in the experiment at high frequencies. It will be shown that this does not affect the measurement of the reactive silencers, as their contribution to the IL will not extend to this region.

Figure 4 shows a 8.8dB increase in the measured IL of the 125Hz third octave band of S2 compared to S1. There is also an increase in IL in the neighbouring third octave bands with the whole 125Hz octave band having an average IL increase of 4.2dB. In all tests the increase in insertion loss was gained without decreasing the broadband dissipative insertion loss in any third octave band. This demonstrates that the design of the hybrid silencer is capable of significantly increasing the performance of traditional dissipative silencers at a range of frequencies surrounding the target. The fact that the bandwidth of this silencer is greater than one third octave means that the silencer will continue to operate effectively despite small changes to the frequency of the tonal noise.

Above this fundamental peak there is a second region with increased insertion loss between 315Hz and 800Hz caused by harmonic resonance peaks. These peaks have similar bandwidths and amplitudes as the fundamental and occur at frequencies of approximately $f = (2n + 1)f_r$, where n are integers. However there is a clear upper frequency limit above which these resonances do not appear and the reactive silencer does not contribute to the insertion loss. For silencer S2 this is observed in the 800Hz third octave band, see Figure 4. This upper frequency limit is related to the cut-on frequency of a single silencer module rather than the whole duct width, allowing for the reactive components to operate well above the plane wave region of the duct.

Comparing the predicted transmission loss to the measured insertion loss shows that the model is capable of accurately calculating the frequency of the resonance peaks, see Figure 4. However the numerical model overpredicts the amplitude of the resonances. This difference is partly due to the damping effects present in the system such as structural interactions. The model assumes an ideal silencer with completely rigid walls, neglecting interaction between airborne sound and wall vibrations, whereas it was found during the experiment that there was some flex in the silencer walls that would act to decrease performance.

Similar observations are made for S3, where the IL of the hybrid silencer is 7.6 dB greater than the dissipative silencer in the 500Hz third octave band, see Figure 5. The numerical model again shows that it is able to accurately calculate the frequency of the peaks, but significantly overpredicts the amplitude of the peak with an 18.9dB increase in transmission loss.

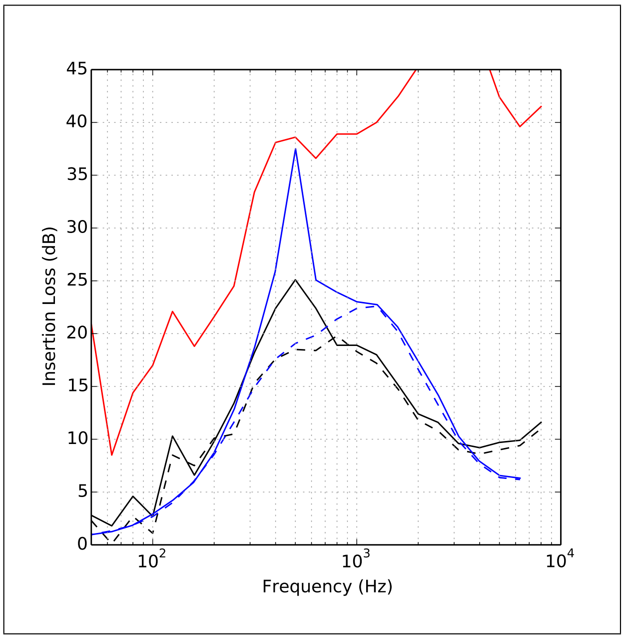

Figure 5. Insertion loss of S1 (Measured, – – –; Predicted, – – –) and S3 (Measured, ———; Predicted, ———). Limiting insertion loss, ———.

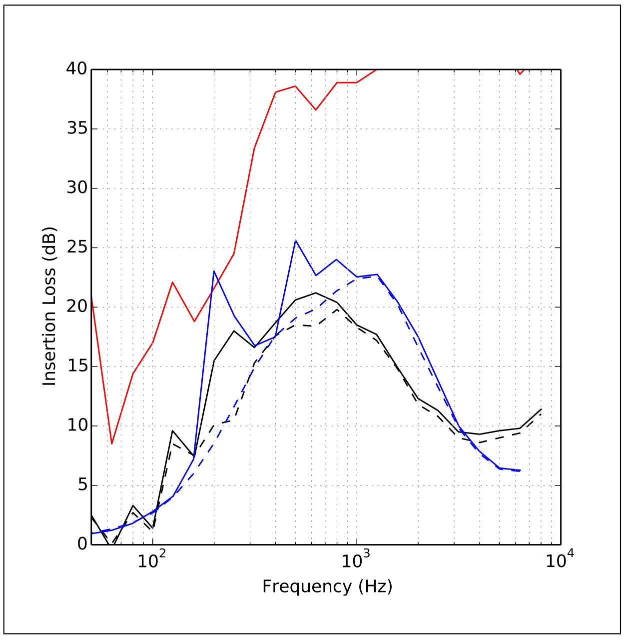

Figure 6. Insertion loss of S1 (Measured, – – –; Predicted, – – –) and S4 (Measured, ———; Predicted, ———). Limiting insertion loss, ———.

As the target frequency is so close to the silencers upper frequency limit no harmonics are found in the spectrum of S3.

Figure 6 presents the IL for S4, demonstrating that the type 2 model designed to operate at very low frequencies is able to increase silencer performance of the 125Hz third octave band by 7.5dB. Theoretically the type 2 design should have a lower performance than the type 1 design as the cross-sectional area of the

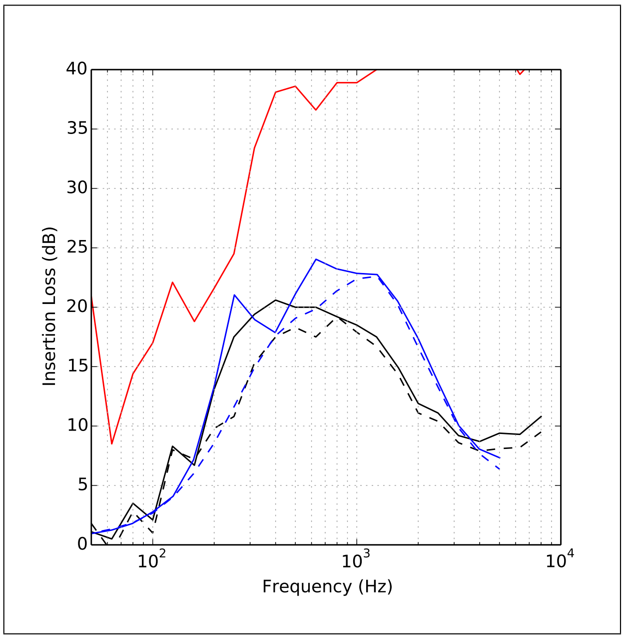

Figure 7. Insertion loss of S1 (Measured, – – –; Predicted, – – –) and a silencer using 1 baffle of S4 and S5 (Measured, ———; Predicted, ———). Limiting insertion loss, ———.

chamber has been halved. The fact that this is not observed in the measurements signifies that the maximum insertion loss is being limited by losses in the experimental setup. Measured values should therefore be taken as being a lower bound on the performance at the resonance peaks.

To extend the range of problems that the hybrid silencer is capable of solving, designs were explored which increased the number of consecutive third octaves affected by the reactive element. Figure 7 illustrates the effect of tuning both baffles to different frequencies by using one baffle from silencer S4 and one baffle from S5 to form the hybrid silencer. The resulting spectrum shows an increase in performance across 6 consecutive third octave bands demonstrating that by careful choice of the fundamental resonance frequencies it is possible to achieve a pseudo-broadband performance using reactive silencer elements.

As hybrid silencers are designed to operate in the presence of flow it is important that this does not have an adverse affect upon the insertion loss. All silencers tested were measured under the three flow conditions of static flow, flow at 13ms^{-1} and flow at 26ms^{-1}. Figure 8 shows the result of these tests for S3 with the whole spectrum showing a small decrease in performance as flow speed is increased. This decrease is comparable at third octaves close to and away from resonance peaks indicating that the change in performance is due to the dissipative element's insertion loss decreasing and is not due to the reactive elements.

In practical applications these noise control solutions are required to operate in high temperature conditions. The elevated temperature will act to increase

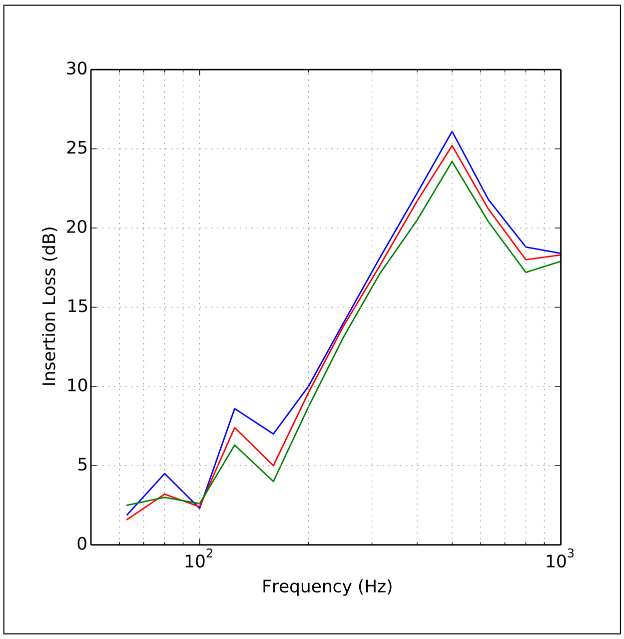

Figure 8. Measured insertion loss of S3 in flow with speeds of $0ms^{-1}$ (———), $13ms^{-1}$ (———) and $26ms^{-1}$ (———).

wavelengths and therefore reactive silencer length, but will also act to decrease the attenuation of dissipative silencers. The numerical model predicts that a reactive silencer tuned to target the 31.5Hz octave band in air with a temperature of 500°C would require a length of under 4.5m, providing a minimum insertion loss of 6.5dB for a type 1 design. A dissipative silencer under the same conditions would require a length of 24m to provide the same level of insertion loss in the 31.5Hz octave band. Clearly novel silencing solutions, such as the presented hybrid silencer, become more desirable when low frequency tonal noise exists in the source noise spectrum.

6. CONCLUSIONS

A novel hybrid in-duct silencer has been presented for controlling broadband noise with tonal content in large diameter ducts, using current silencer design techniques to minimise static pressure drop. Measurements were taken to demonstrate that significant performance gains could be achieved in targeted octave bands while maintaining the insertion loss of the dissipative element. An advanced numerical model was developed and shown to be capable of accurately predicting the insertion loss of these hybrid silencers. Discrepancies between measurements and the predictions were attributed to damping effects present in the experimental system.

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