

An evaluation of current commercial acoustic FEA software for modelling small complex muffler geometries: prediction vs experiment

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ABSTRACT

Mufflers are incorporated into continuous positive airway pressure (CPAP) devices to reduce noise in the air paths to and from the flow generating fan. The mufflers are very small, irregularly shaped, and must attenuate noise up to high frequencies. It is important that the acoustic performance of these mufflers is reliably predicted and optimised, in order to improve the quality of the well-being of the user. In this study, finite element acoustical results for three reactive muffler designs were obtained using four commercial software packages – ANSYS, COMSOL, LMS Virtual.Lab Acoustics and VA-One. Experimental results have been obtained using the two-microphone acoustic pulse method. Results of the transmission loss of each muffler obtained from the finite element models are presented and validation of the computational results is discussed. The finite element modelling packages were consistent in their prediction of the resonant frequencies and magnitude of the transmission loss at each resonance. Validation of the models showed good agreement with the experimental results over the lower half of the frequency range however the models appear to over-predict the magnitude of the transmission loss at resonant frequencies.

INTRODUCTION

Obstructive sleep apnoea (OSA) is a medical condition that can be successfully managed through the application of a positive pressure to the airway. This elevated airway pressure is produced by a flow generating fan within a continuous positive airway pressure (CPAP) device. Noise from the flow generator is controlled using mufflers situated in the flow path at fan inlet and flow generator outlet. The mufflers are very small and must attenuate noise up to high frequencies. Often these mufflers are irregularly shaped and consist of a number of interconnected volumes. They are predominantly reactive though absorptive materials may be considered.

A comparison of the various numerical methods used to predict the performance of mufflers has been given by Bilawchuk and Fyfe (2003). The most common type of linear acoustic model applies classical electrical filter theory and is most widely known as the transfer matrix method, though it is also referred to as the two port approach or 4-pole parameter method (Jones 1984, Munjal 1987, Davies 1988). Kim and Soedel (1989) and Wu *et al.* (1988) present an improved method for calculation of the 4-pole parameters which offers several advantages over the original method when applied with the finite element method to evaluate transmission loss (Barbieri *et al.* 2004a).

Computational approaches used to predict the performance of mufflers include the finite element method (FEM) (Young and Crocker 1975, Barbieri *et al.* 2004b, Barbieri and Barbieri 2006), the boundary element method (BEM) (Cheng and Seybert 1991, Selamat and Ji 1999), and computational fluid dynamics (CFD) (Middelberg *et al.* 2004).

Barbieri *et al.* (2004b, 2006) implemented the transfer matrix method using the FEM approach to predict the acoustic performance of expansion chambers using both the original parameter formulation and the improved method.

This study builds on a previous assessment of methodologies for analysing simple expansion chamber mufflers (Jones and Kessissoglou 2008) and applies the finite element acoustic techniques to three reactive muffler designs similar to those found in CPAP devices. Each of the muffler designs were modelled using four commercially available finite element (FE) acoustic software packages – ANSYS, COMSOL, LMS Virtual.Lab Acoustics (VL.Acoustics) and VA-One. Results of the transmission loss of each muffler obtained from the FE models are compared. The computational results are also compared with data obtained experimentally using the two-microphone acoustic pulse method.

MUFFLER DESIGNS

Three reactive muffler designs having dimensions and geometric complexity similar to those used in CPAP devices were selected for analysis.

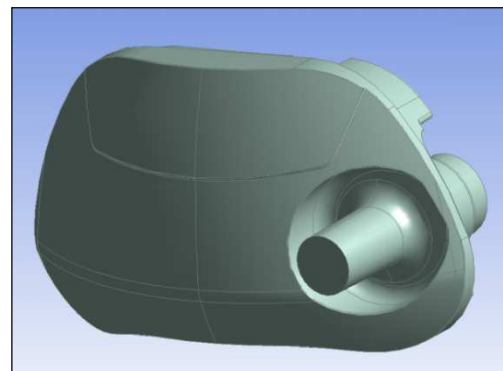


Figure 1: Single chamber muffler design.

The first design (Figure 1) consists of a single chamber having coaxial inlet and outlet ports. While the level of geometric detail in the design is high, the proximity between the inlet and outlet ports suggests that this muffler will essentially perform as a Helmholtz resonator. The second design (Figure 2) consists of three interconnected chambers having orthogonal inlet and outlet ports. A 43mm length of cylindrical pipe having 18mm internal diameter connects each chamber to isolate through-wall transmission between adjacent chambers. It is anticipated that this design will be used in further modelling at a later date to incorporate fluid-structure coupling and to assess the acoustic interaction between adjacent chambers. The third design (Figure 3) consists of two integrated chambers and provides a more complex flow path between the inlet and outlet ports.

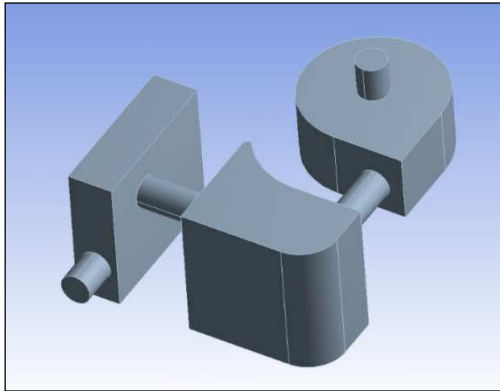


Figure 2: Interconnected chamber muffler design.

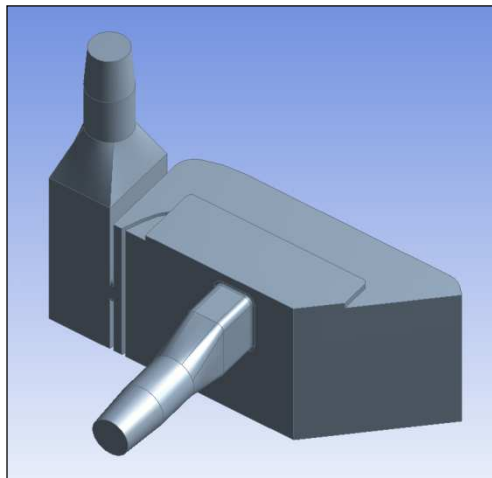


Figure 3: Integrated multi-chamber muffler design.

FINITE ELEMENT MODELS

Computational models of each muffler design were developed using four commercially available FE packages – ANSYS, COMSOL, VL.Acoustics and VA-One. Replication of the analyses using the various packages was undertaken to explore the relative capabilities of each of these products for small reactive type mufflers. Software costs and licencing terms are influenced by many commercial factors and were outside the scope of the evaluation. Two different boundary conditions were chosen in order to compare the different approaches (transfer matrix method and acoustic power).

ANSYS

Transmission loss is calculated in ANSYS by applying the transfer matrix methodology. Each element of an acoustic system can be characterised by a transfer matrix whose parameters describe only that element (Young and Crocker 1975). Adopting acoustic pressure p and volume velocity u as the two state variables, the following general transfer matrix may be written to relate the state variables on either side of an acoustic system.

$$\begin{bmatrix} p_n \\ u_n \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} p_{n+1} \\ u_{n+1} \end{bmatrix} \quad (1)$$

For the case of a non-reflecting (anechoic) termination of a system having inlet and outlet pipes of equal cross-sectional area, the corresponding form of the transmission loss equation incorporating the transfer matrix constants can be shown to be (Munjal 1987)

$$TL = 20 \log_{10} \left[\frac{1}{2} \left(\left| A + B \left(\frac{S_o}{\rho c} \right) + C \left(\frac{\rho c}{S_o} \right) + D \right| \right) \right] \quad (2)$$

S_o is the cross-sectional area of the inlet and outlet pipes, while ρ and c are the density and speed of sound of the fluid. A finite element acoustic model of each muffler design has been developed using ANSYS. The transfer matrices for each design are readily obtained by applying two different load cases to each model and obtaining pressure and velocity data at inlet and outlet over the desired frequency range. Each model was meshed using tetrahedral FLUID30 elements with mesh controls applied to adequately resolve the fine details and tight radii in the muffler geometries. The resulting mesh size produced 25 elements per acoustic wavelength at the upper bound of the frequency range being analysed (limiting case). This is very high compared a widely accepted reasonable minimum mesh density of 5 or 6 elements per wavelength. A harmonic input velocity equal to unity was specified for each case using an equivalent displacement boundary condition. The fluid (air) was assumed to be non-flowing and inviscid and acoustic damping was not applied at the fluid-structure interface (ie. the walls were treated as acoustically hard boundaries).

COMSOL

Transmission loss is calculated directly in COMSOL using the acoustic power at the inlet and outlet of the acoustic system. Sound power is defined as (Reynolds 1981)

$$W = \int_s \frac{p_{rms}^2}{\rho c} dS \quad (3)$$

p_{rms} is the root mean square pressure and S is the area of the surface through which the sound power is passing. The corresponding form of the transmission loss equation is

$$TL = 10 \log_{10} (W_i / W_t) \quad (4)$$

W_i , W_t are the incident and transmitted sound power, respectively. This method of calculating transmission loss offers a significant advantage over the ANSYS method in that only one load case is required, thereby reducing computational effort. The corresponding disadvantage with this method is that it does not allow the transfer matrix for the acoustic system to be obtained. While the solve time per

frequency is significantly less as a percentage, due to the small size of the mufflers and the corresponding FE models, the absolute differences in computational run times were not significant.

A model of each muffler design has been developed using COMSOL. Each model was meshed using the default (Lagrange-quadratic) elements with controls applied to achieve a mesh consistent in size and distribution to that generated in the corresponding ANSYS model. A harmonic pressure of 1 Pa was specified at the inlet and a radiation condition applied at inlet and outlet. Acoustic damping was not applied at the fluid-structure boundaries for comparison with the ANSYS results.

LMS Virtual.Lab Acoustics (VL.Acoustics)

Transmission loss is calculated directly in VL.Acoustics by using the acoustic pressures associated with the positive travelling acoustic waves at the inlet and outlet of the acoustic system. The corresponding form of the transmission loss equation is

$$TL = 20 \log_{10}(p_i^+ / p_t^+) \tag{5}$$

p_i^+ , p_t^+ are the positive travelling components of the incident and transmitted acoustic pressure, respectively. Similar to the method of calculating transmission loss that was used in COMSOL, this method also only requires one load case and also does not allow the transfer matrix for the acoustic system to be obtained.

A model of each muffler design has been developed using VL.Acoustics. In each instance, the mesh generated in the corresponding ANSYS model was exported from ANSYS and imported directly into VL.Acoustics, ensuring that the size and distribution was identical. A harmonic normal velocity of 1 ms⁻¹ was specified at the inlet and an anechoic condition ($Z = \rho c$) was applied at the outlet boundary. The application of an anechoic condition eliminates any reflection at the outlet, ensuring that the positive travelling component of the transmitted pressure is identical to the total pressure at the outlet plane.

In order to obtain the positive travelling component of the incident pressure it is necessary to separate it from the negative travelling component total pressure using the incident velocity

$$p_i^+ = \frac{1}{2}(p_i + \rho c \cdot u_i) \tag{6}$$

p_i , u_i are the incident total pressure and particle velocity, respectively. Acoustic damping was not applied at the fluid-structure boundaries for comparison with the ANSYS results.

VA-One

Transmission loss is calculated in VA-One by applying the transfer matrix methodology implemented in the ANSYS models but using a different matrix arrangement. Again, acoustic pressure p and volume velocity u are adopted as the two state variables, however the transfer matrix is expressed in the form adopted by Wu *et al.* (1998) and Barbieri *et al.* (2004b)

$$\begin{bmatrix} p_n \\ p_{n+1} \end{bmatrix} = \begin{bmatrix} A^* & B^* \\ C^* & D^* \end{bmatrix} \begin{bmatrix} u_n / S_n \\ u_{n+1} / S_{n+1} \end{bmatrix} \tag{7}$$

Obtaining the transfer matrices for each design still requires the application of two different load cases to each model but

it is now only necessary to obtain pressure data at inlet and outlet over the desired frequency range (ie. velocity data is no longer required). The association between the matrix parameters used in Eq. (1) and those in Eq. (7) can be shown to be:

$$A = \frac{A^*}{C^*}, \quad B = B^* - \frac{A^* D^*}{C^*} \tag{8, 9}$$

$$C = \frac{1}{C^*}, \quad D = \frac{-D^*}{C^*} \tag{10, 11}$$

For the case of a non-reflecting (anechoic) termination of the system, the transmission loss is calculated using Eq. (2).

A model of each muffler design has been developed using the finite element capabilities of VA-One (FE subsystems and FE cavities). Each model was meshed using the default tetrahedral elements with controls applied to achieve a mesh consistent in size and distribution to that generated in the corresponding ANSYS model. A 0.1% (default) damping loss factor was applied to the FE subsystems which were modelled as uniform shells. A harmonic input velocity of unity amplitude was specified for each case using a constraint boundary condition. The fluid (air) was assumed to be non-flowing. For the results presented here, junctions between the FE faces and the FE cavity were omitted in order for the wall treatment to remain consistent with that used in ANSYS (ie. acoustically hard).

EXPERIMENTAL METHOD

Methods that use a short duration acoustic pulse or acoustic impulse to determine the transmission loss of mufflers are well described in literature (Seybert and Ross 1977).

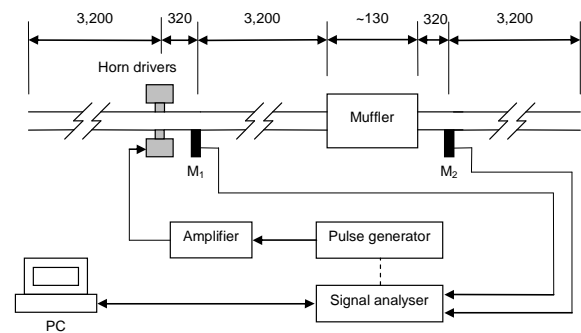


Figure 4: Schematic diagram of the two-microphone acoustic pulse experimental set-up.



Figure 5: Photograph of the experimental test rig.

Figure 4 shows a schematic diagram of the two-microphone experimental set-up used in the current work. Figure 5 shows a photograph of the experimental test rig.

A transient acoustic pulse was generated from the Brüel & Kjær Pulse front end and fed to two horn drivers via a power amplifier. The pulse propagated down the 18mm diameter conduit where it was measured by the upstream microphone, M_1 , before continuing to the muffler inlet. The pressure of the corresponding pulse transmitted from the outlet of the muffler was measured by the downstream microphone, M_2 .

The resulting time histories recorded by the two microphones were processed in order to extract only those time periods that captured the initial positive travelling wave. Rectangular windowing with leading and trailing cosine tapers was applied to the time history measured by M_1 (Figure 6) and exponential windowing with a leading cosine taper and 5ms decay constant (τ) was applied to the time history measured by M_2 (Figure 7).

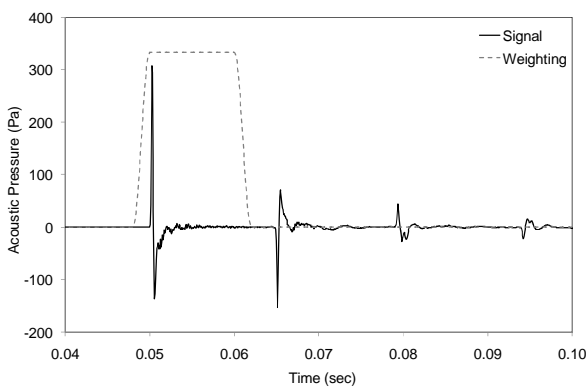


Figure 6: Inlet microphone (M_1) time history showing FFT windowing

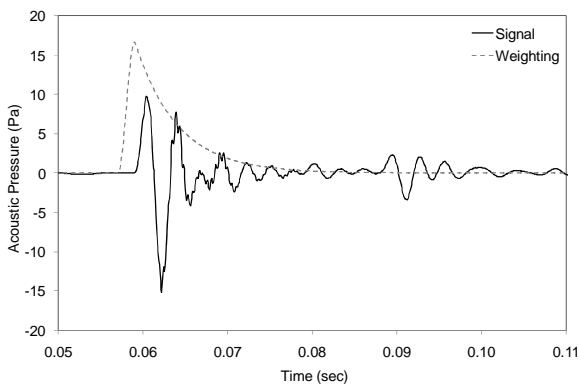


Figure 7: Outlet microphone (M_2) time history for single chamber muffler design showing FFT windowing

To facilitate the data extraction, long lengths of pipe were used which provided sufficient time delay (approximately 15ms) between the arrival of the initial pulse and the reflected waves generated at the muffler and pipe ends. These extracted time histories were captured for 100 individual pulses, Fourier Transformed, and the results averaged in the frequency domain. The transmission loss for the muffler was then obtained by

$$TL = 10 \log_{10} \left(\frac{FFT_1}{FFT_2} \right) \quad (12)$$

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

RESULTS

Results of the transmission loss of each muffler obtained using the two-microphone acoustic pulse method are presented and validation of the computational results obtained using the three software packages is discussed.

Single Chamber Design

Figure 8 compares the transmission loss results obtained for the single chamber muffler using three of the FE software packages and show excellent agreement over the frequency range assessed. Results using VA-One were not obtained as it was not possible to generate a computational mesh for the single chamber muffler design using either the default mesh parameter settings or following limited adjustment of these parameters. Without restricted time constraints, it may be possible to generate a suitable mesh for this geometry by further refinement of the mesh parameter settings.

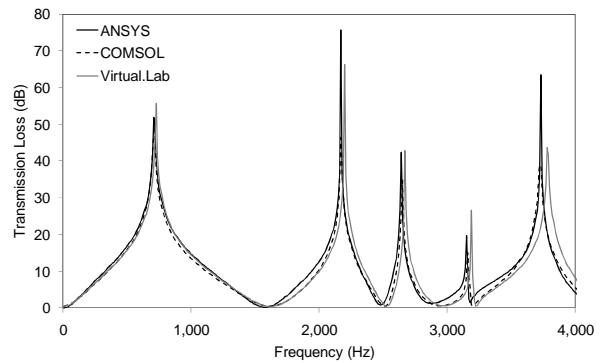


Figure 8: Single chamber muffler comparing results from each FE software package (except VA-One).

Figure 9 compares the transmission loss results obtained for the single chamber muffler using the ANSYS FE model with results obtained experimentally. The results show excellent agreement over the frequency range assessed with the exception that the magnitudes at resonant frequencies are over-predicted by the FE model. This is attributed to the model assuming inviscid fluid and rigid walls.

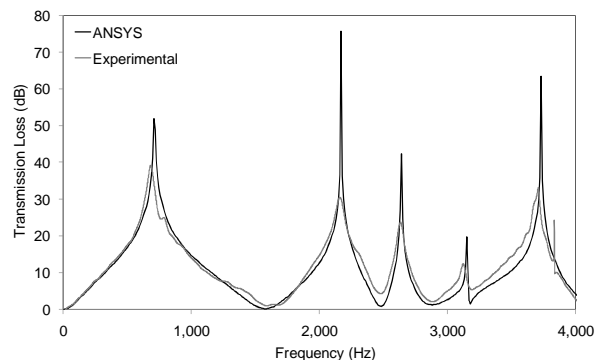


Figure 9: Single chamber muffler comparing ANSYS results (black line) and experimental results (grey line).

Interconnected Chamber Design

Figure 10 compares the transmission loss results obtained for the interconnected chamber muffler using all four FE modelling software packages. Figure 11 compares the transmission loss results obtained using the ANSYS FE model to results obtained experimentally. All software packages are consistent in their prediction of the resonant frequencies, particularly in the lower half of the frequency range assessed. The packages also show consistent prediction of the magnitude of the transmission loss at each resonance.

Differences in results obtained from the software packages may be attributed to (a) the finite element formulation used in each package (ie. linear elements in ANSYS versus quadratic Lagrange elements in COMSOL), (b) the different approaches used to obtain transmission loss (sound power reduction versus the use of transfer matrices), and (c) the different computational meshes (density and refinement) that inherently result from automated tetrahedral mesh generation tools.

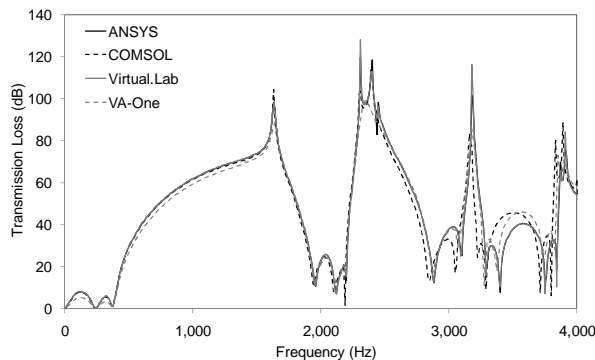


Figure 10: Interconnected chamber muffler comparing results from each FE software package.

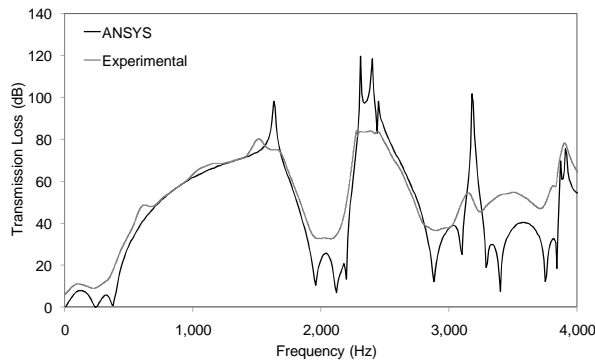


Figure 11: Interconnected chamber muffler comparing ANSYS results (black line) and experimental results (grey line).

The experimental results also predict similar resonant frequencies for the majority of the frequency range. Departure between the experimental and computational results may be attributed to poor coherence above 2 kHz due to very low pressures (less than 20×10^{-6} Pa) observed in the FFT spectrum of the downstream microphone (M_2). These observations highlight the importance of producing an acoustic pulse of short duration which still has sufficient energy at high frequencies to provide an acceptable signal-to-noise ratio. The periodic oscillations in the experimental results are attributed to leakage errors associated with the FFT of the downstream microphone results and a possible resonance condition in the system. Both the acoustic pulse

generation and periodic oscillations will be investigated in future experiments.

Integrated Chamber Design

Figure 12 compares the transmission loss results obtained for the integrated chamber muffler using all four FE modelling software packages. Figure 13 compares the transmission loss results obtained using ANSYS to results obtained experimentally. All of the FE results show excellent agreement for the whole frequency range with the exception of the results using COMSOL that slightly deviate at high frequencies. All resonant frequencies predicted computationally are in excellent agreement as are the magnitudes of the corresponding transmission loss predictions. The FE results validate well against the experimental data over the lower half of the frequency range before the correlation starts to break down.

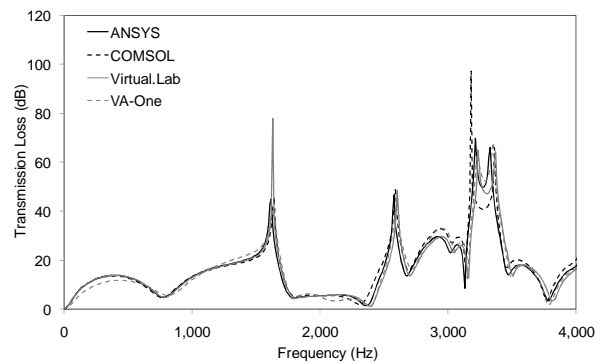


Figure 12: Integrated chamber muffler comparing results from each FE software package.

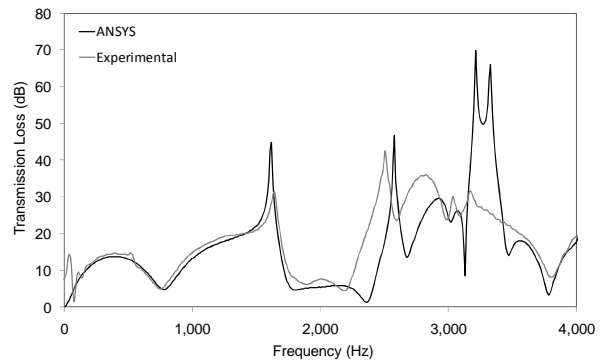


Figure 13: Integrated chamber muffler comparing ANSYS results (black line) and experimental results (grey line).

CONCLUSIONS

In this study, the acoustic performance of three reactive muffler designs used in CPAP devices has been computationally and experimentally compared. Finite element acoustical results were obtained using four commercial software packages – ANSYS, COMSOL, Virtual.Lab and VA-One. Experimental results have been obtained using the two-microphone acoustic pulse method.

The finite element modelling packages were consistent in their prediction of resonant frequencies and magnitude of the transmission loss at each resonance. Differences in the computational predictions were generally observed only at higher frequencies and this is attributed to differences in the computational meshes, different finite element formulations and different approaches used to obtain the transmission loss.

Comparison of the computational results with those obtained experimentally showed that the models over-predicted the magnitude of the transmission loss at resonant frequencies. This is attributed to the fact that no damping is included in the computational models, which all assume inviscid fluid and rigid walls.

Improved results from the FE models may be achieved by introducing fluid damping and realistic fluid-structure interaction while improved experimental results may be achieved by improving the high frequency acoustic energy of the transient pulse and addressing the resonance condition in the system.

Evaluation of the four software packages found that, for the reactive type mufflers investigated in this work, the results did not identify any one package that performed significantly better than the others (compared to the experimental results). Further exploration of the capabilities of the FE packages is required to assess their relative suitability for future work which will include modelling resistive muffler elements (foam) and implementation of optimisation techniques.

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