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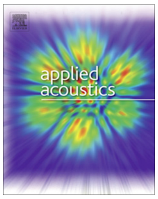

Technical Note

Noise reduction using a quarter wave tube with different orifice geometries

Carl Q. Howard *, Richard A. Craig

School of Mechanical Engineering, The University of Adelaide, South Australia 5005, Australia

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ABSTRACT

It is well known that the acoustic performance of silencing elements decreases with an increase in exhaust gas flow. Tests were conducted on three orifice geometries of side-branches on an adaptive quarter-wave tube to determine which was the least compromised by the high-speed exhaust gas passing over the side-branch. The side-branch geometries that were tested were a sharp edge, a backward inclined branch, and a bell mouth. The experimental results show that the side-branch with a bell-mouth geometry resulted in the greatest noise reduction by an adaptive quarter-wave tube.

1. Introduction

Quarter-wave tubes are a reactive acoustic device used to attenuate tonal noise at a fundamental and odd-harmonic frequencies. The quarter-wave tube is attached to the main duct with a side-branch. When gas flows past the side-branch at low speeds (less than Mach 0.1), it has been found that the acoustic attenuation performance is not degraded significantly [1]. However, when flow speeds increase above Mach > 0.1, then the acoustic attenuation caused by the quarter-wave tube can be degraded, because the resistive part of the side branch resonator impedance will usually increase [2].

In the work presented here, the noise reduction of an Adaptive Quarter Wave Tube (AQWT) with three configurations of side-branch geometries is described, namely a 90° branch with square edges, a side branch that is backwards inclined at 45° to the flow in the main exhaust duct, and a side branch with a bell-mouth.

2. Previous work

Lambert [3] showed theoretically that the insertion loss of a side-branch resonator attached to a main duct is very sensitive to the Mach flow number, especially at frequencies near resonance. In a companion paper [4], he experimentally showed that for flows greater than Mach > 0.1, the insertion loss "for all practical purposes had been destroyed by the flow".

Knotts and Selamet [5] conducted experimental investigations of several geometries of the neck of a side-branch to determine which configuration was least likely to generate unwanted pure tone whistle noise. The shapes included those with square edges, ramps, bevelled edges, and curved (radius) edges. Their experimental tests indicated that beveled and radius edges was effective at reducing unwanted tonal noise generation. Their work focused on the minimization of flow induced noise, whereas the focus of the work in this paper is providing the greatest noise reduction of tonal noise by an adaptive quarter wave tube.

Most of the previous work on suppression of flow noise from cavities has examined shallow cavities where the cavity length to width ratio (L/d) is less than one. Rockwell and Naudascher [6] provide an extensive review on the subject of noise generated by flow over cavities.

Singhal [7] examined the influence of air-flow past a quarter wave tube, that was backwards and forward inclined at 45°, on the noise generation. He found that when the device was installed so that the quarter-wave tube was forwards inclined to the flow, it whistled. He also examined the non-linear coupling between cavity modes for a quarter-wave tube attached at 90° to the main duct. The noise spectrum comprised combinations of sum and differences of the cavity modes.

Anderson [8] examined the effect of air flow past a Helmholtz resonator and found that the effect of increasing air flow was to increase the resonance frequency of the resonator, which implies

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* Corresponding author.
E-mail address: carl.howard@adelaide.edu.au (C.Q. Howard).

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that the reactive impedance of the neck decreased. Ingard and Ising [9] found that the effect of high intensity sound levels (above 110 dB) impinging on an orifice caused the resistive impedance to increase. The work reported here is similar to that by Knotts and Selamet [5], however, the emphasis is on examination of the noise reduction caused by a quarter wave tube with various side-branch geometries in a realistic application.

There is a large body of research literature on the acoustic performance of perforated plates and single small diameter (less than 5 mm) circular orifices, which indicates there is a linear relationship between flow speed and the resistance (real part of the impedance). However, references that describe the variation in throat impedance with flow speed for large diameters (>50 mm) are not available. This is consistent with comments in the research literature. Holmberg and Karlsson [10] commented that, “In side branch orifices the general trend for a grazing flow configuration is an increase in the resistance and a decrease in the reactance with an increasing Mach number. Analytical modeling of single orifices have been attempted, but it have been shown that the correlation with experimental data is generally unsatisfactory.” These comments are also consistent with Karlsson and Abom [11], “Attempts have been made to describe the acoustic impedance of single orifices with analytical models. However, Jing et al. and Peat et al. have shown that the correlation to measured data is generally unsatisfactory.”.

Dequand et al. [12] investigated flow induced acoustic resonances of side-branches. In their experimental and theoretical studies they were concerned with two side-branches that were coaxially aligned. They considered side-branches that had junctions to the main duct with sharp edges and rounded edges. From their experimental testing they found that there was a significant variation in the sound pressure response both in amplitude, and flow velocity depending on whether the cross-junction geometry had a sharp-edged or bell-mouth opening, which is consistent with our experimental results presented in this paper.

3. Theory

A Quarter-Wave Tube (QWT) side-branch resonator has an (uncoupled) resonance frequency f_{QWT} given by

$$f_{\mathrm{QWT}} = \frac{c}{4L_{\mathrm{eff}}} \tag{1}$$

where c is the speed of sound and L_{eff} is the effective length of the quarter-wave tube that includes end effects. The speed of sound in air is given by [13]

$$c = \sqrt{\frac{\gamma R T}{M}} \tag{2}$$

where $\gamma = 1.4$ is the ratio of specific heat that is applicable both for diatomic molecules and air, $R = 8.314\ \mathrm{J\ mol^{-1}\ K^{-1}}$ is the molar Universal gas constant, $M = 0.02857\ \mathrm{kg\ mol^{-1}}$ is the average molar mass, and T is the gas temperature in Kelvin. These values will be slightly different for exhaust gases that typically have high concentrations of gaseous carbon dioxide and water.

These quarter wave tube resonators provide acoustic attenuation at odd multiples (i.e. $1\times$, $3\times$, $5\times$, ...) of the fundamental ($1\times$) resonance frequency, when they are attached to an acoustic duct.

For an engine rotating at constant speed and under constant load, the excitation frequency will remain constant and thus the required length (L_{eff}) of a QWT remains constant. However, if the engine conditions were to change, such as by altering the load on the engine, the temperature of the exhaust gas (T) changes, which alters the speed of sound in the exhaust gas as indicated in Eq. (2). Therefore, a fixed length quarter-wave tube will only provide attenuation for a fixed engine speed and exhaust gas temperature. Whereas the adaptive quarter-wave tube presented here can be tuned to attenuate tonal noise over a range of engine speeds and exhaust gas temperatures.

For a four-stroke reciprocating engine, the cylinder firing frequency f_{cylinder} occurs at half the crankshaft speed multiplied by the number of cylinders in the engine, hence

$$f_{\mathrm{cylinder}} = \frac{\mathrm{RPM}}{60} \times \frac{\mathrm{cylinders}}{2} \tag{3}$$

For example, in the case of a V8 engine, rotating at 1500 rpm the cylinder firing frequency is 100 Hz.

4. Numerical predictions

There are several metrics to describe the acoustic performance of mufflers that include Insertion Loss (IL), Transmission Loss (TL), and Noise Reduction (NR) [14].

The *transmission loss* is the ($10\log_{10}$) ratio of the incident to transmitted acoustic power on the muffler. This is simple to calculate using numerical simulations, but extremely difficult to measure in practice [15,16], as it requires the measurement of the incident sound power on the muffler in a system with flowing high-temperature (500 °C) exhaust gas that is turbulent.

The *noise reduction* is the difference in the upstream and downstream sound pressure levels across the exhaust muffler. This is relatively easy to measure in practice, but difficult to estimate theoretically as the source and termination impedances must be known. Several methods have been proposed to measure the source impedance of an engine [15–18]. Munjal [18] noted that Prasad and Crocker [19] found that the source impedance of the multi-cylinder inline engine they investigated could be approximated as an anechoic termination. The method suggested by Boonen and Sas [17], which involves the use of two microphones and varying the impedance of the exhaust system, was attempted in this work but gave unsatisfactory results. In their work they found that the averaged source impedance at high frequencies could be approximated as anechoic. In the modeling work conducted here, the source impedance was simulated as an anechoic termination.

The *insertion loss* is the reduction (in decibels) in sound power transmitted through a duct compared to that transmitted with no muffler in place. The insertion loss can be measured by using a single microphone located outside the exhaust system, with and without the muffler installed. This measurement technique was attempted however, it was found that the acoustic enclosure around the diesel engine did not provide adequate isolation and the measurements of exhaust noise were contaminated by noise radiating from the engine and hence the insertion loss results were inaccurate. In order to make theoretical predictions of insertion loss, the source and termination impedance must be known, which as described above, is very difficult to measure in practice.

Numerical predictions of the expected transmission loss of a quarter-wave tube were made using a lumped-parameter model and Finite Element Analysis (FEA). One benefit of using FEA is that the transmission loss can be predicted for the three side-branch geometries considered here.

4.1. Lumped parameter model

A lumped parameter model was created to aid in the verification of the predictions made with finite element analysis. The insertion loss caused by the installation of a quarter-wave-tube on an infinite duct is given by [20]

$$\mathrm{IL} = 20\log_{10}\left|1 + \frac{Z_d}{Z_r}\right| \tag{4}$$