Development of an Adaptive Quarter-Wave Tube Attached to a Large Diesel Engine

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ABSTRACT

An adaptive quarter-wave tube was developed for a large diesel engine that has the capability of tuning to variations in engine speed, exhaust gas temperature, and load on the engine. The system is robust to the diesel exhaust gas that reaches temperatures of over 450°C and contains soot. This paper describes the components of the system and some experimental results that demonstrate the effectiveness of the device, where noise reductions greater than 20dB were achieved.

INTRODUCTION

The noise emissions from the exhaust of a reciprocating internal combustion engine make a significant contribution to environmental noise pollution. Passive attenuation methods (such as a muffler) provide broadband noise reduction of such emissions. To provide narrow-band attenuation at select frequencies, Helmholtz resonators or quarter-wave tubes can be used. It is difficult to maintain the desired attenuation using passive resonators and tubes due to variations in engine speed and exhaust gas temperature, as these devices are designed to have a high quality factor to provide high levels of noise attenuation. Adaptive-passive attenuation devices have the capability of altering their geometry and thus resonance frequency, so that they can be tuned to attenuate tonal noise in an exhaust pipe. The main advantages of using adaptive-passive devices over active noise control systems are that additional energy is not required to achieve noise reduction, the measurement and control systems are less complex and lastly, there is less design complexity to make them robust to hot and dirty exhaust gases.

A number of researchers have investigated the use of Helmholtz resonators and quarter-wave tubes, both passive and adaptive-passive, to attenuate noise in the exhaust of reciprocating engines.

The resonant frequency of a Helmholtz resonators can be varied by adjusting the cavity volume or the neck volume (cross-sectional area or length). Several authors have developed Helmholtz resonators with a variable cavity volume (Lamancusa 1987; de Bedout et al. 1997; Kostek & Franchek 2000). A volume-variable resonator may be the easiest to implement in practice, but becomes unnecessarily bulky at low frequencies. Helmholtz resonators with adjustable neck areas have also been considered, although not widely implemented (Izumi et al. 1991; Nagaya et al. 2001; Esteve and Johnson 2004; Cheng et al. 1999; Kotsun et al. 2004; Ciray 2005).

There is less published research literature about adaptive quarter-wave tubes. This may be because of the analytical complexities of control and technical difficulties in implementing adjustment methods. Although Neise & Koopman (1980) and Koopman & Neise (1982) investigated the use of manually adjustable quarter-wave tubes to attenuate the noise from centrifugal fans and were able to attenuate the noise at the blade-passage-frequency by more than 25dB.

Howard and Craig (2011) experimentally tested an adaptive-passive quarter-wave tube for use on the exhaust gas stream of a spark ignition petrol engine. A manually adjusted quarter-wave tube system was used which attained approximately 15dB attenuation. The work presented here is an extension to this initial research and introduces a fully automatic adaptive quarter-wave tube control system that has been applied to attenuate the noise at the engine firing frequency on a compression ignition (diesel) engine.

This paper describes the development of an adaptive quarter-wave tube side-branch resonator system that is able to adapt to changes in engine speed, engine load, and exhaust gas temperature. The adaptive quarter-wave tube attenuates tonal noise at the odd harmonics of the fundamental firing frequency of a diesel engine. The fundamental equations used for adaptive quarter-wave tube design are presented, followed by a detailed presentation of the experimental apparatus and some experimental results. These measurements provide insight into the interrelationships between sound pressure level, exhaust gas flow rate, exhaust gas temperature, and the complexities of achieving noise attenuation in such a challenging environment.

BACKGROUND

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

$$f_{QWT} = c/(4L) \quad \text{Hz} \tag{1}$$

where c is the speed of sound, and L is the length of the quarter-wave tube. The speed of sound in air is given by (Bies & Hansen 2009):

$$c = \sqrt{\frac{\gamma RT}{M}} \ \text{m/s} \tag{2}$$

where $\gamma = 1.4$ is the ratio of specific heat that is applicable both for diatomic molecules and air, $R = 8.314 \text{J mol}^{-1} \text{K}^{-1}$ is the molar Universal gas constant, $M = 0.02857 \text{kg mol}^{-1}$ is the average molar mass, and T is the gas temperature in Kelvin.

For an engine rotating at constant speed and under constant load, the resonant frequency (f_{QWT}) will remain constant and thus the length (L) of 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 Equation 2. Therefore, a fixed length quarter-wave tube will only provide attenuation for limited combinations of engine speeds and exhaust gas temperatures. 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 piston firing frequency f_{engine} occurs at half the crankshaft speed multiplied by the number of pistons in the engine, hence

$$f_{\mathrm{engine}} = \frac{RPM}{60} \times \frac{\text{pistons}}{2} \quad \text{Hertz} \qquad (3)$$

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

COMPONENTS OF THE EXPERIMENTAL APPARATUS

The diesel engine used for the adaptive QWT experiments is a four stroke, direct injection, Mercedes Benz OM502LA, PP1066 Power drive Unit. The engine configuration is a 90°-V8, with air to air intercooled twin turbochargers and displacement of 15.93 litres. The maximum rated power is 420 kW (563 hp (mechanical)) at 1800 rpm (intermittent) but reduces to 350 kW (469 hp) for continuous operation. The maximum torque at the flywheel is specified to be 2700 Nm at 1200 rpm.

The diesel engine was installed within a 20-foot acoustically lined modified shipping container. This reduced the noise level in the laboratory when testing and provided safe fuel storage. The exhaust system was routed through a hole in the roof of the enclosure. The adaptive quarter-wave tube system and passive muffler were mounted on the roof of the enclosure.

The engine is loaded using an engine-mounted water brake dynamometer, shown in Figure 1. This type of dynamometer works by converting the power from the engine into water shear and momentum exchange (water turbulence) which subsequently becomes heated. This braking action, or load, is developed by a rotor that directs the water against a stator. The stator in turn redirects the water back against the rotor, thereby opposing the motion of the rotor. The greater the flow of the water through the dynamometer, the greater the braking action or load on the engine. A Taylor Dynamometers model TD-3100 was attached to the engine flywheel, and has a maximum accepted load of 746 kW (1000 hp (mechanical)) and peak torque of 4068 Nm at 1250rpm. The water flow was controlled using a PC based Taylor Dynamometers DynPro data acquisition and control system which operates on a Microsoft Windows™ operating system.

To provide a continuous load capability, a water recirculation system was installed. A schematic of the system is shown in Figure 2. The system consisted of two 22,500 litre water reservoir tanks, a constant pressure multistage supply pump, a 450 litre return water reservoir, a high volume centrifugal return water pump, and a two stream, cross flow, single pass, unmixed heat exchanger (automotive radiator). The system was designed for water flow rates up to 6.5 L/s.

Figure 1. Photograph of the Taylor water brake dynamometer (red) mounted to the flywheel of the diesel engine. Also shown are the heat shielded (white) exhaust pipes.

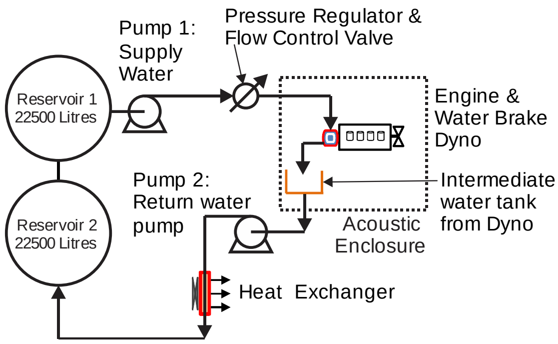

Figure 2. Water storage and recirculation system for dynamometer and water-cooled microphone mounts.

The diameter of the exhaust pipe from each turbo on the engine is 76.3mm (3 inch). These two exhaust pipes then joined at a Y - junction to form a 114.3mm (4.5″) diameter pipe, as can be seen in Figure 3, before passing through the roof of the enclosure. The exhaust pipe then turns 90° and extends horizontally along the roof of the acoustic enclosure where the side-branch Adaptive Quarter-Wave Tube (AQWT) is installed. Further downstream is the passive muffler. To allow entry into the passive muffler the exhaust pipe splits into two 3″ diameter pipes again (not shown in Figure 3).

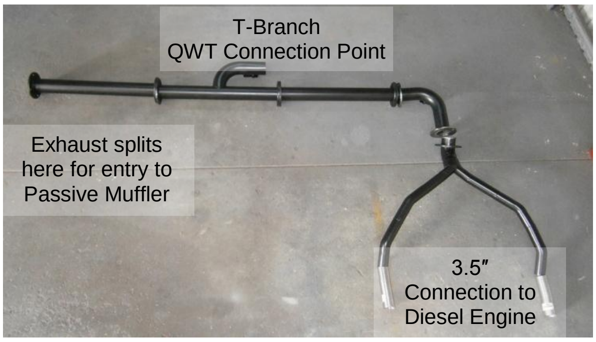

Figure 3. Photograph of the 4.5" (114mm) exhaust system before fitting to the Mercedes Benz diesel engine.

The adaptive quarter-wave tube (AQWT) is attached to the 4.5″ diameter section of the exhaust pipe via a side-branch, which is positioned approximately C_L = 0.775m from the downstream flange of the 90° elbow at the enclosure penetration. The side-branch was manufactured by welding a114.3mm diameter tube 90° to the main exhaust duct. Due to the limited area available on the roof of the acoustic enclosure, a 90° bend was used on the side-branch resulting in the AQWT running parallel to the exhaust pipe, as shown in Figure 4.

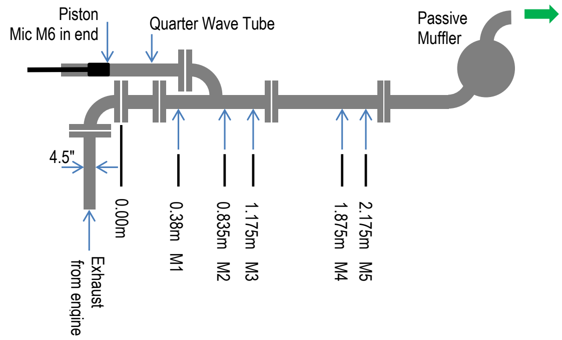

Figure 4. Schematic showing location of microphones mounted in the exhaust system.

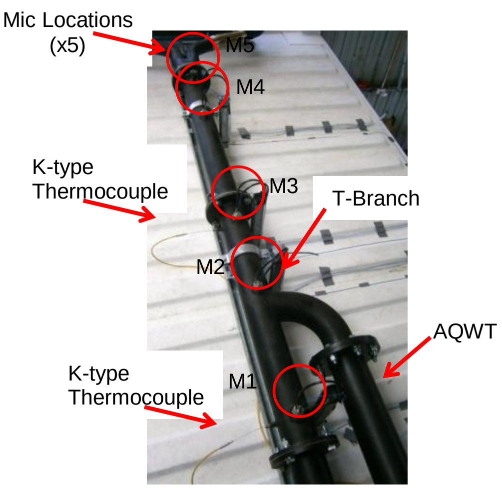

Figure 5: Photograph of the installed exhaust showing the side-branch and QWT, and positions of the water cooled microphones.

The sound in the exhaust was measured using PCB 106B microphones housed within custom-made water-cooled jackets and mounted at the locations shown in Figure 4 and Figure 5. The microphone was mounted such that its face was flush with the internal diameter of the exhaust pipe. The water-cooled adapters maintain the microphones at about 40°C, when exposed to exhaust gas temperatures of around 450°C. The cooling water was supplied from the two large water reservoirs.

Thermocouples (K type) were used for measuring the temperatures of the exhaust gas, microphones, cooling water and pipe surface.

COMPONENTS OF THE ADAPTIVE QUARTER-WAVE TUBE

The adaptive quarter-wave tube system consists of five main components, as shown in Figure 6:

1). The AQWT bore is a cold drawn steel tube.

2). The piston slides through the bore and is positioned to achieve the desired length. It also houses a microphone, M6.

3). The bearing housing helps alignment of the piston in the bore. It houses the bearing through which the piston rod slides.

4). The piston rod is a smooth tube that connects the piston to the rod coupling.

5). The rod coupling is used to connect the push rod to the linear actuator that positions the piston.

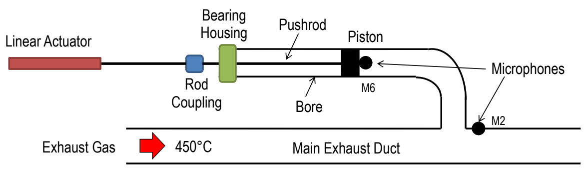

Figure 6. Components of the Adaptive Quarter-Wave Tube.

CONTROL SYSTEM AND ACTUATOR

The control system comprises a desktop computer running Matlab-Simulink software, a Dspace 1104 controller, two pressure sensor microphones (PCB106B) and cables, an ICP signal conditioning amplifier that provide a voltage signal from the PCB pressure sensors to the DSpace 1104 controller, and an engine tachometer signal. In brief, the microcontroller in the DSpace 1104 accepts analog inputs from the microphone and tachometer signals, and generates appropriate control signals to send to a motor-controller, that in turn is connected to a linear actuator that moves the piston in the adaptive quarter-wave tube. The cost-function evaluation for this system is based on the phase angle of the frequency response function between a microphone in the face of the piston of the AQWT and a microphone installed in the main duct. The adaptive algorithm adjusts the position of the piston until the phase angle is -90 degrees. A process and control system diagram is shown as Figure 7. A full discussion about the operation of the control system and the governing cost function algorithm is explained in Howard (2012).

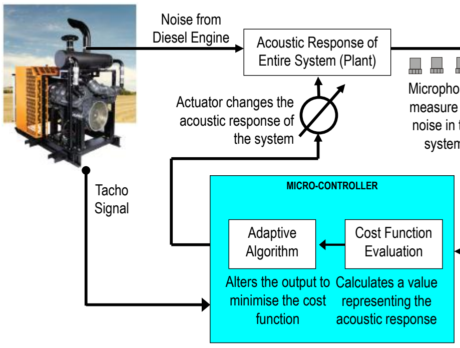

Figure 7. Process and control diagram for the adaptive quarter-wave tube.

The linear actuator that positioned the piston within the bore was a Parker Model ET050B02LA000. It is a ball screw drive with an inline direct drive motor that can traverse at up to 300mm/s for the 1000mm stroke. The actuator can deliver up to 3300N of force.

RESULTS

Experimental results are presented to show the effectiveness of the AQWT at adapting to changes in engine speed, exhaust gas temperature, and engine load. The first two cases were used to ascertain if the quarter-wave tube was located at an acoustic pressure node, which would reduce its effectiveness. The third case shows the attenuation for steady state engine and load conditions. The fourth case shows the adaptability of the system to maintain attenuation when there is a step change in engine speed. Similarly, the final case shows the attenuation during a change in applied load and the adaptability of the system to maintain good attenuation as the Exhaust Gas Temperature (EGT) increases.

The position of the adaptive quarter-wave tube along the exhaust pipe was chosen after measuring the sound pressure levels along the exhaust pipe. There was also consideration for having fully developed gas flow through the main exhaust to reduce the likelihood of unwanted turbulence and vortex shedding at the side-branch.

The results for the first case shown in Figure 8, are measurements of Sound Pressure Levels (SPLs) at locations along the exhaust pipe when the AQWT is fully retracted (maximum acoustic path length hence lowest frequency), fully extended (minimum acoustic path length, hence highest frequency) and when tuned using the automatic control system. A constant engine speed of 1500rpm (100Hz) was maintained and there was no load applied to the engine. The steady state EGT measured at the side-branch was approximately 163°C. The plot shows that the AQWT is not located at a minimum SPL region (acoustic node). It also shows that an average noise reduction of 26dB was achieved by comparing the results when the AQWT was tuned to the maximum frequency (shortest acoustic path length) and when it was tuned to 100Hz. Note also that the noise reduction commences at the location of the side-branch and continues downstream, and hence is not just attenuated at the 'error' microphone. The results also indicate that the SPL upstream of the AQWT has increased when it is tuned, which is to be expected as the incident noise is reflected upstream.

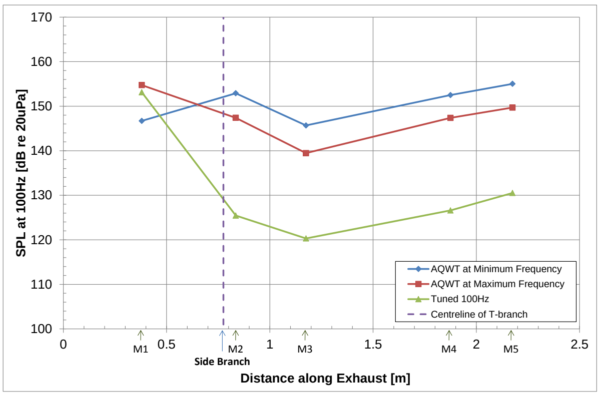

Figure 8. Sound Pressure Level at 100Hz for distances along the exhaust pipe for a constant engine speed of 1500rpm. No load was applied to the engine and the EGT at the side-branch was 163°C.

The second case, is the similar to Case 1 but for a constant engine speed of 1800rpm (120Hz) and an applied load of 125kW, which results in an EGT of 447°C at the side-branch. Figure 9 shows the SPL in the exhaust versus axial distance along the exhaust pipe. Again, the plot shows that the AQWT is not located at a minimum SPL region (acoustic node). The average noise reduction is 10dB, by comparing the results when the AQWT was tuned to the maximum frequency (shortest acoustic path length) and when it was tuned to 120Hz. Although not displayed here, it should be noted that an average noise reduction of 29dB was measured for the 1800rpm, no load case (EGT = 183°C). The reduction in acoustic attenuation performance with the applied load is because of the increase in gas flow past the side branch, which increases the resistive part of the side branch resonator impedance (Bies & Hansen 2009).

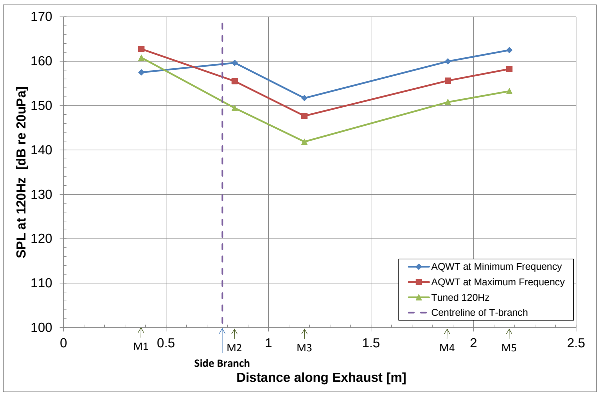

Figure 9. Sound Pressure Level at 120Hz for distances along the exhaust pipe for a constant engine speed of 1800rpm. A load of 125kW was applied to the engine and the EGT at the side-branch was 447°C.

Comparing Figure 8 and Figure 9, it can be seen that the same trends are followed but there is an overall increase in SPL for the 1800rpm, engine loaded case. This is the result of higher cylinder pressures generated to produce the higher output power from the engine.

The third case is a comparison of the SPL at the firing frequency (fourth order) both with and without an AQWT installed in the exhaust system. A straight exhaust pipe section was put in the location of the AQWT when it was removed. When the AQWT was installed, it was initially tuned to the maximum frequency (minimum acoustic path length) for 10 seconds; the control system was then engaged. A comparison of the SPL at the same microphone position in this initial period shows that having the AQWT tuned to the maximum frequency with the shortest acoustic path length, is very similar to having no AQWT installed. Hence, the noise reduction can be easily measured by comparing the difference in SPLs when the AQWT has the shortest acoustic path, and when it is tuned to attenuate the noise at engine firing frequency.

Figure 10 shows these comparisons for the case when the engine was at a constant speed of 1500rpm and was loaded with 106kW, which resulted in an EGT of 450°C at the side-branch. The plot shows that the system takes approximately 15 seconds to tune and can obtain approximately 8dB to 10dB of noise reduction. It should be noted that the speed of the linear actuator can operate at more than 20 times greater than used in these tests, and hence the speed of adaptation can be improved significantly.

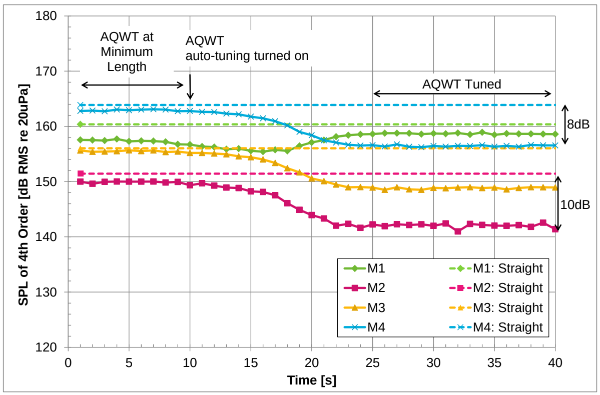

Figure 10. Comparison of with (solid line) and without (broken line) a AQWT installed in the exhaust system. When AQWT installed: variation from minimum length to tuned. Engine at 1500rpm, 106kW load, EGT 450°C

The adaptability of the system to maintain noise reduction when there is a step change in engine speed was then examined. This fourth case investigated a step change of 300rpm (1500rpm to 1800rpm) over approximately 4 seconds for both with and without the AQWT installed in the exhaust pipe. An increase of applied load by the dynamometer was used to maintain a constant exhaust gas temperature of 450°C at the side-branch during the change in engine speed. This was achieved by maintaining a constant applied torque (i.e. constant flow rate of water through the dynamometer).

The results from this fourth investigation are displayed in Figure 11. The plot shows an initial noise reduction of approximately 10dB was obtained by using the AQWT system before the engine speed change. During the change in engine speed the control system responds quickly and tracks the change in phase, and determines the new optimum position for the piston in the AQWT. There is a slight decrease in the noise reduction of about 2dB until the adaptive system has finished moving after about 10 seconds, after which the noise reduction is 10dB.

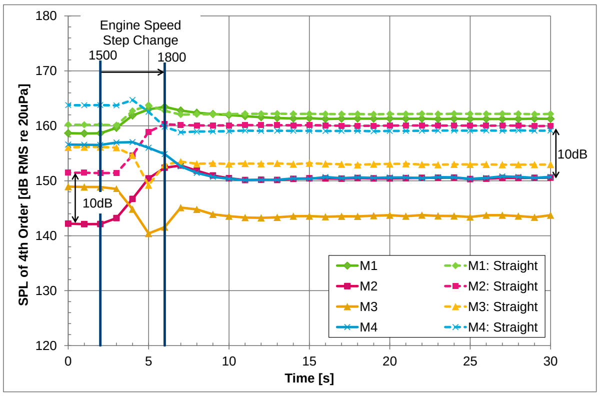

Figure 11. Measurement of SPL over time for a step change in engine speed from 1500rpm to 1800rpm, for both with an AQWT (solid line) and with a straight pipe (broken line) installed in the exhaust system. A constant EGT of approx. 450°C was maintained.

The fifth case demonstrations the adaptability of the AQWT system to maintain noise reduction during a change in applied load to the engine over time. With an increase in applied load, the EGT increases.

The engine was run at a constant speed of 1800rpm with a load of 60kW. After steady state conditions were achieved, the load was slowly increased at a rate of 7kW/min over 10 minutes. The increase in load caused the EGT to rise from 344°C to around 452°C. These temperatures were measured slightly downstream of the side branch (i.e. at M3). Figure 12 shows the increase in EGT over time due to the applied load to the engine for both with and without (straight pipe) the AQWT installed in the exhaust pipe. A temperature difference of between 0°C and 12°C was recorded between the two experiments (with and without an AQWT), indicating the same experimental conditions were being achieved.

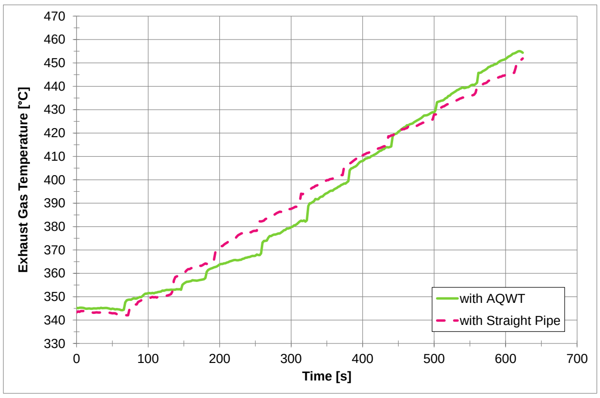

Figure 12. Measured EGT at M3 over time for a constant engine speed of 1800rpm, for both with an AQWT (solid line) and with a straight pipe (broken line) installed in the exhaust system. Beginning with 60kW applied to the engine, a continual load of 7kW/min was applied over 10 minutes (resulting in 130kW).

The variation in sound pressure level at the engine firing frequency (120Hz) over time is shown in Figure 13. The results show that an average noise reduction of 15dB was measured at the commencement of the test, when compared with the straight pipe data. At the end of the test, an average noise reduction of around 10dB was measured. The decrease in attenuation is due to the greater gas flow past the side branch due to the higher output power from the engine (higher cylinder pressures), which increases the resistive part of the side-branch resonator impedance.

Figure 13 also shows an increase in SPL with applied load. Microphones M2 and M4 show an increase of approximately 10dB (from 141dB to 151dB) over the 10 minutes. The change in SPL is caused by the increase in amplitude of the exhaust pressure pulsations which is caused by maintaining a constant engine speed when increasing the load.

The AQWT was switched off after 10 minutes and manually adjusted to the highest frequency (shortest acoustic path length). It can be seen that the measured SPL for the straight pipe is the same as when the AQWT is at a minimum length.

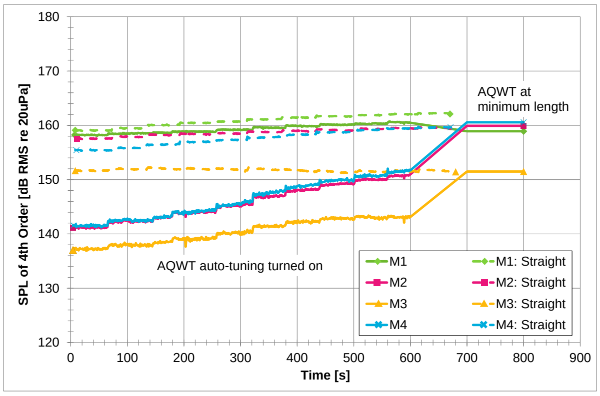

Figure 13. Measured SPLs over time at microphones M1 to M4 for a constant engine speed of 1800rpm. Beginning with 60kW applied load, a continual load of 7kW/min was applied over 10 minutes (resulting in 130kW). The EGT increased from 344°C to 452°C.

RESPONSE OF MICROPHONES WITH AND WITHOUT SOOT

Tests were conducted to measure the transfer function between microphones with and without soot on the face of the microphones. The tests involved removing two microphones from the exhaust system that had been installed for approximately 10 hours of tests. The faces of the microphones were covered in soot, as shown in Figure 14(A).

The microphones were inserted into a custom made calibrator made from standard exhaust pipe with a nominal bore diameter of 65mm. A loudspeaker was installed into a sealed chamber at the top of the calibrator. The loudspeaker was connected to a 200W Playmaster power amplifier, which was connected to a signal generator (NI 9263). The loudspeaker was excited with broadband noise and the transfer function between the two microphones was recorded using a National Instruments cDAQ NI-9234 data acquisition module and processed using SignalExpress software.

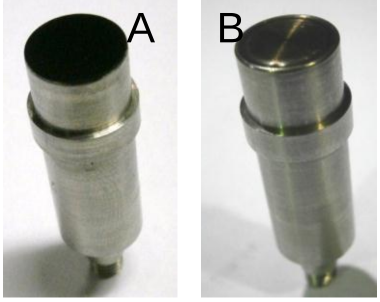

Figure 14. Photo of a microphone with soot on the face (A) and after cleaning (B).

Figure 15 and Figure 16 show the amplitude and phase angle of the transfer function between microphones M5 and M3 with and without soot on their faces, respectively. The results show that there are no significant differences when the microphones have soot on their faces and when they are clean.

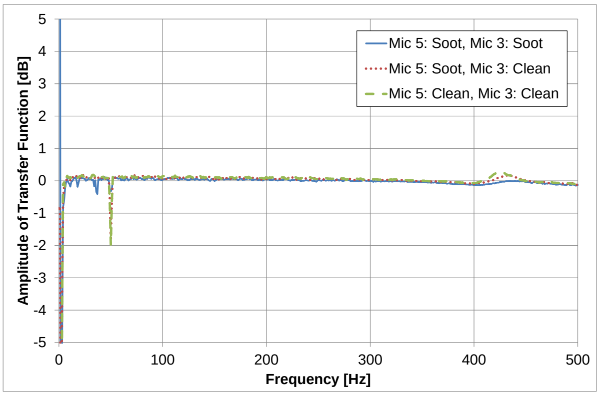

Figure 15. Amplitude of the transfer function between microphones with and without soot on the face.

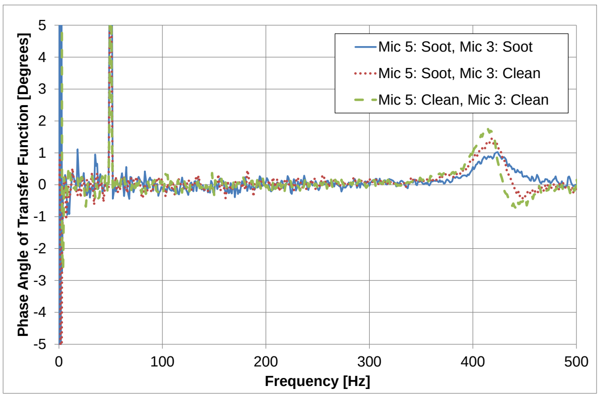

Figure 16. Phase angle of the transfer function between microphones with and without soot.

SUMMARY

This paper described the development of an adaptive quarter-wave tube that was used to attenuate the variable engine firing frequency in the exhaust of a large diesel engine. The adaptive system has the capability of tuning to variations in engine speed, applied engine load and exhaust gas temperature. The system is robust to the diesel exhaust gas that reaches temperatures of over 450°C and contains soot. Tests were conducted to ensure that the side-branch was not installed at an acoustic pressure node, which would limit the noise reduction that could be achieved.

The results from five test cases were presented and a noise reduction of up to 26dB was measured. It was found that by tuning the AQWT to the highest frequency, where the acoustic path length in the AQWT is the shortest, gave results that were very similar to having a straight piece of exhaust pipe installed instead of the side-branch and AQWT. This provided a quick method of ascertaining the noise reduction, which significantly reduced experimental complexity and time, by not having to repeat every test condition with a straight piece of exhaust pipe. In addition, by conducting this simple test means that the engine operating conditions are constant, rather than having to shutdown the engine, install the straight piece of exhaust pipe, and restart the test configuration. This finding was confirmed for more than 30 engine configurations of engine speed, exhaust gas temperature and load. However, this finding cannot be generalised to other muffler installations where there is the potential for vortex shedding and the generation of self-noise to occur due to the installation of a side-branch.

Measurements were made of the amplitude and phase angle of the transfer function between microphones with and without soot on the face of the microphones. It was found that soot made no difference to the calibration of the microphone.

In summary, it has been demonstrated that an adaptive quarter-wave tube can be used to attenuate noise at the engine firing frequency for a range of engine speeds, exhaust gas temperatures and applied loads.

In a companion paper (Howard & Craig 2012), it is shown that noise reductions of up to 40dB can be achieved by altering the geometry of the side-branch.

REFERENCES

Bies, DA and Hansen, CH 2009, *Engineering Noise Control*, Fourth Edition, Spon Press.

Ciray, MS 2005, 'Exhaust processor with variable tuning system', United States Patent Number 6,915,876.

Cheng, CR, McIntosh, JD, Zuroski, MT and Eriksson, LJ 1999, 'Tunable acoustic system', United States Patent Number 5,930,371.

de Bedout, JM, Franchek, MA, Bernhard, RJ, Mongeau, L 1997, 'Adaptive-passive noise control with self-tuning helmholtz resonators', *Journal of Sound and Vi-bration*, vol 202, no. 1, pp. 109-123, DOI: 10.1006/jsvi.1996.0796.

Estève, SJ and Johnson, ME 2004, 'Development of an adaptive Helmholtz resonator for broadband noise control', *Proceedings of IMECE 2004*, Anaheim, CA, November. ASME International Mechanical Engineering Congress.

Howard, CQ 2012, 'A Sliding Goertzel Algorithm for Adaptive Passive Neutralizers', *Journal of Sound and Vibration*, vol. 331, no. 9, April, pp. 1985-1993, DOI: 10.1016/j.jsv.2012.01.007.

Howard, CQ and Craig, RA 2012, 'Acoustic Performance of Three Orifice Geometries of Side-Branches in a Duct with Flowing Gas', *Proceedings of Acoustics 2012*, Western Australia, Australia, 21-23 November, Paper 80.

Howard, CQ and Craig, RA 2011, 'Adaptive-Passive Quarter-Wave Tube Resonator Silencer', Proceedings of Acoustics 2011, Gold Coast, Australia, 2-4 November, Paper 64.

Izumi T, Takami H, Narikiyo T 1991, 'Muffler system controlling an aperture neck of a resonator', *Journal of the Acoustical Society of Japan*, vol. 47, no. 9, pp. 647-652.

Koopmann, GH, Neise W 1982, 'The use of resonators to silence centrifugal blowers', *Journal of Sound and Vibration*, vol. 82, no. 1, pp. 17-27, DOI: 10.1016/0022-460X(82)90539-9.

Kostek, TM and Franchek, MA 2000, 'Hybrid noise control in ducts', *Journal of Sound and Vibration*, vol. 237, no. 1, pp. 81-100, DOI: 10.1006/jsvi.2000.3056.

Kotsun, JD, Goenka, LN, Moenssen, DJ, and Shaw, CE 2004, 'Helmholtz resonator', United States Patent Number 6,792,907.

Lamancusa, JS 1987, 'An actively tuned, passive muffler system for engine silencing', *Proceedings of Noise-Con 87*, pp. 313–318.

Nagaya K, Hano Y, and Suda A 2001, 'Silencer consisting of a two stage Helmholtz resonator with auto tuning control', *Journal of the Acoustical Society of America*, vol. 110, no. 1, pp. 289-295.

Neise W, Koopmann, GH 1980, 'Reduction of centrifugal fan noise by use of resonators', *Journal of Sound and Vibration*, vol. 73, no. 2, pp. 297-308, DOI: 10.1016/0022-460X(80)90697-5.